Heating Ventilating Air Conditioning Guide 1939

HEATING VENTILATING AIR CONDITIONING GUIDE

1939

AN INSTRUMENT OF SERVICE PREPARED FOR THE PROFESSION—CONTAINING A

Technical Data Section

OF REFERENCE MATERIAL ON THE DESIGN AND SPECIFICATION OF HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS—BASED ON THE TRANSACTIONS—THE INVESTIGATIONS OF THE RESEARCH LABORATORY AND CO-OPERATING INSTITUTIONS—AND THE PRACTICE OF THE MEMBERS AND FRIENDS OF THE SOCIETY

TOGETHER WITH A

Manufacturers' Catalog Data Section

CONTAINING ESSENTIAL AND RELIABLE INFORMATION CONCERNING MODERN EQUIPMENT

ALSO

The Roll of Membership of the Society

WITH

Complete Indexes

TO TECHNICAL AND CATALOG DATA SECTIONS

Vol. 17

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AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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PREFACE TO THE 17th EDITION

THE 1939 edition of the HEATING, VENTILATING, AIR CONDITIONING GUIDE contains the largest Technical Data Section of any of the annual reference volumes published by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS to serve the profession and industry. With 856 pages of text material this 17th edition of The GUIDE reflects the outstanding research and engineering advancements which have taken place in this field of engineering since the first GUIDE made its appearance in 1922.

The Research Laboratory of the Society and cooperating universities have contributed much toward the advancement of the science of heating, ventilating and air conditioning in 1938. All of these developments have been summarized and added to this volume in order to maintain the original conception of The GUIDE, whereby only scientific and current investigative results are presented which may be of practical value to design engineers.

Basic and fundamental data have been retained from previous editions but in this volume a concentrated effort has been made by the Committee to condense and combine material throughout the text. Those chapters dealing with central systems for heating, humidifying, cooling and dehumidifying have been carefully reviewed and a completely new chapter has been prepared which incorporates all of the essential material from other chapters The title of this chapter is, Central Systems for Comfort Air Conditioning, and the material is presented in such a manner as to apply directly to the requirements set forth in the application Code of Minimum Requirements for Comfort Air Conditioning which was adopted by the Society in January, 1938. References are given in this material to other chapters which deal more specifically with the design of particular equipment and consequently this new chapter may serve as a basic guide for the design of all comfort air conditioning installations.

A subject which has not heretofore received special attention in previous editions of The GUIDE is information dealing with heat transfer surface coils for summer and winter applications. The Committee presents in this edition a chapter which is intended to give the latest information which is available on this type of equipment. Details are given describing the conventional construction and arrangement of heating and cooling coils with suggestions for typical flow arrangements of the circulating media. Due to the wide variety of methods now in use for the selection of heat transfer coils it is impossible to give specific recommendations for the proper choice of coils until such time as additional research findings are reported and uniform standardization is adopted by the industry. Charts are given which indicate the performance of coils operating under varying design load and equipment conditions.

The problem of fuels and their combustion was recognized by the Committee this year as an important factor to the heating engineer and as a consequence a completely revised chapter on Combustion and Fuels

is presented in the 1939 edition. Fundamental principles of combustion together with related factors dealing with flame temperature and proper air requirements are given special consideration. The latest current information on classification of all types of fuels is given along with suggested methods of firing to obtain optimum results. Another chapter closely related to this subject has been rewritten this year on Heat and Fuel Utilization. Methods are outlined for estimating fuel consumption based on calculated heat losses and degree-day factors. Charts are included for coal, oil and gas whereby fuel burning rates may be estimated for various design loads and operating efficiencies.

Other chapters which have been rewritten for the 17th edition deal with Air, Water and Steam, Cooling Load, Radiators and Gravity Convectors, Steam Heating Systems, Piping for Steam Heating Systems, Mechanical Warm Air Furnace Systems, Unit Heaters, Ventilators, Air Conditioning and Cooling Units, Natural Ventilation and District Heating.

Minor changes were made in the chapters on Heat Transmission Coefficients and Tables, Air Leakage, Automatic Fuel Burning Equipment, Air Distribution and Automatic Control.

The Bulkeley Psychrometric Chart will again be found in a convenient envelope attached to the inside back cover thus making it easily accessible for performing psychrometric calculations. The Problems in Practice have been continued with new and practical solutions to expand the text material given in each of the 45 chapters. A revised table giving the weights of saturated and partially saturated air for various psychrometric pressures has been added this year to bring the data in conformity with government standards. The degree-day table appearing in the chapter on Heat and Fuel Utilization has been corrected and brought up-to-date based on more recent Weather Bureau data.

The increase in usage of The GUIDE as a text book in many universities, engineering and technical schools for student instruction indicates its wide acceptance as a standard authority in the field of heating, ventilating and air conditioning.

The manufacturers have cooperated fully to make their information useful in the Catalog Data Section so that the reader can apply it effectively to the selection of materials or equipment to be used in general design. The Committee appreciates the aid of those progressive manufacturers who have assisted in this cooperative enterprise to advance the art of heating, ventilating and air conditioning which so vitally affects the comfort and health of everyone. In 1939 it will be possible to distribute 13,000 copies of The GUIDE to various types of readers and it is extremely gratifying that a large number of manufacturers recognize the effective service which The GUIDE accomplishes as an advertising medium.

For the user who desires to obtain reliable data in convenient form it is felt that a careful perusal of both the text and catalog data pages of this volume will emphasize its brevity and simplicity of presentation.

The Committee releases this 17th edition of 13,000 copies with the sincere hope that it will receive the same enthusiastic reception that was accorded to its predecessors.

Albert Buenger, *Chairman*

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EDITORIAL ACKNOWLEDGMENT

EXTENDING over 17 years the HEATING, VENTILATING, AIR CONDITIONING GUIDE, published by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, has maintained an enviable record as the authoritative engineering reference volume of the profession. This recognition of world-wide acceptance has been possible because of the willingness of hundreds of technically trained engineers to contribute freely from their knowledge and practical experience for the benefit of the entire profession and allied industries.

It is with deep feeling of appreciation that the Guide Publication Committee acknowledges the work of the following individuals who have assisted in the preparation of the 1939 edition and it also commends those contributors who have previously prepared a firm foundation for the addition of new information.

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Individuals who have adopted the GUIDE as their standard reference volume and members of the Society are deeply indebted to those engineers who have assisted in the preparation of this 17th edition and the Guide Publication Committee wishes to pay tribute for this loyal cooperation and devotion to public service in advancing the art of heating, ventilating and air conditioning engineering.

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ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics·

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

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7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

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Chapter 1

AIR, WATER AND STEAM

Dalton's Law, Temperatures, Air Properties, Humidity, Relative Humidity, Specific Humidity, Relation of Dew Point to Relative Humidity, Adiabatic Saturation of Air, Total Heat and Enthalpy, Psychrometric Chart, Properties of Water, Properties of Steam

AIR conditioning has for its objective the supplying and maintaining, in a room or other enclosure, of an atmosphere having a composition, temperature, humidity, and motion which will produce desired effects upon the occupants of the room or upon materials stored or handled in it.

Dry air is a mechanical mixture of gases composed, in percentage of volume, as follows[1]: nitrogen 78.03, oxygen 20.99, argon 0.94, carbon dioxide 0.03, and small amounts of hydrogen and other gases.

Atmospheric air at sea level is given in percentage by volume as: N_2 77.08, O_2 20.75, water vapor 1.2, A 0.93, CO_2 0.03 and H_2 0.01. The amount of water vapor varies greatly under different conditions and is frequently one of the most important constituents since it affects bodily comfort and greatly affects all kinds of hygroscopic materials.

DALTON'S LAW

A mixture of dry gases and water vapor, such as atmospheric air, obeys Dalton's Law of Partial Pressures; each gas or vapor in a mixture, at a given temperature, contributes to the observed pressure the same amount that it would have exerted by itself at the same temperature had no other gas or vapor been present. If p = the observed pressure of the mixture and p_1, p_2, p_3, etc. = the pressure of the gases or vapors corresponding to the observed temperature, then

$$p = p_1 + p_2 + p_3, \text{ etc.} \tag{1}$$

TEMPERATURES

Air is said to be saturated at a given temperature when the water vapor mixed with the air is in the dry saturated condition or, what is the equivalent, when the space occupied by the mixture holds the maximum possible weight of water *vapor* at that temperature. If the water vapor

[1] *International Critical Tables*

mixed with the dry air is superheated, *i.e.*, if its temperature is above the temperature of saturation for the actual water vapor partial pressure, the air is not saturated.

The starting point of most applications of thermodynamic principles to air conditioning problems is the experimental determination of the dry-bulb and wet-bulb temperatures, and sometimes the barometric pressure.

The *dry-bulb temperature* of the air is the temperature indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air. The *wet-bulb temperature* is determined by a thermometer with its bulb encased in a fine mesh fabric bag moistened with clean water and whirled through the air until the thermometer assumes a steady temperature. According to the theory of W. H. Carrier[2], this steady temperature is the result of a dynamic equilibrium between the rate at which heat is transferred from the air to the water on the bulb and the rate at which this heat is utilized in evaporating moisture from the bulb. The rate at which heat is transferred from the air to the water is substantially proportional to the wet-bulb depression $(t - t')$, while the rate of heat utilization in evaporation is proportional to the difference between the saturation pressure of the water at the wet-bulb temperature and the actual partial pressure of the water vapor in the air $(e' - e)$. Carrier's equation for this dynamic equilibrium is.

$$\frac{e' - e}{t - t'} = \frac{B - e'}{2800 - 1.3t'} \tag{2a}$$

In the form commonly used,

$$e = e' - \frac{(B - e')(t - t')}{2800 - 1.3t'} \tag{2b}$$

where

e = actual partial pressure of water vapor in the air, inches of mercury

e' = saturation pressure at wet-bulb temperature, inches of mercury.

B = barometric pressure, inches of mercury.

t = dry-bulb temperature, degrees Fahrenheit

t' = wet-bulb temperature, degrees Fahrenheit.

The derivation of Equation 2a was based upon the theory, supported by extensive experiments with atmospheric air, that the wet-bulb temperature and the temperature of adiabatic saturation (see page 11) are identical. Subsequent study and experiment[3,4,5] have shown that these temperatures are very nearly the same for air and water-vapor mixtures in the proportions and temperature range of normal atmospheric air, but that they differ widely for mixtures of dry air and vapors other than water, and for air-water mixtures at high temperature or vapor content.

It is now recognized that the wet-bulb temperature is influenced, not

[2]Rational Psychrometric Formulae, by W H Carrier (*A.S M E Transactions*, Vol 33, 1911, p 1005).

[3]The Evaporation of a Liquid into a Gas—a Correction, by W K. Lewis (*Mechanical Engineering*, September, 1933).

[4]The Theory of the Psychrometer, by J H Arnold (*Physics*, July, September, 1933)

[5]The Deviation of the Actual Wet-Bulb Temperature from the Temperature of Adiabatic Saturation, by David Dropkin (Cornell University *Engineering Experiment Station Bulletin*, No 23, July, 1936)

only by the rate of heat transfer by convection from air to wet-bulb, but also by the rate of heat conduction through the thin film of stagnant air that clings to the wet-bulb, and by the rates of outward diffusion of vapor through the air film and of convection of vapor away from the film. Thus, there is no theoretical foundation for the equality of wet-bulb and adiabatic-saturation temperatures; rather, it is by mere chance that in atmospheric air the ratio of the heat transfer and vapor transfer coefficients is such as to make these temperatures substantially equal. In accordance with current air conditioning practice, they are assumed to be equal in the psychrometric equations and chart presented in the 1939 Guide.

Formula 2b may be used to determine the actual partial pressure of the water vapor in a dry air-water vapor mixture. Then, from Dalton's Law of Partial Pressures, Equation 1, it follows that the partial pressure of the dry air is $(B - e)$.

If a mixture of dry air and water vapor, initially unsaturated, be cooled at constant pressure, the temperature at which condensation of the water vapor begins is called the *dew-point temperature*. Clearly the dew-point is the saturation temperature corresponding to the actual partial pressure, e, of the water vapor in the mixture.

AIR PROPERTIES

Density is variously defined as the mass per unit of volume, the weight per unit of volume, or the ratio of the mass, or weight, of a given volume of a substance to the mass, or weight, of an equal volume of some other substance such as water or air under standard conditions of temperature and pressure. The term *specific gravity* is more commonly used to express the latter relation but, when the gram is taken as the unit of mass and the cubic centimeter as the unit of volume, density and specific gravity have the same meaning. The term *specific density* is sometimes used to distinguish the weight in pounds per cubic foot; and as here used, *density* is the weight in pounds of one cubic foot of a substance.

The density of air decreases with increase in temperature when under constant pressure. The density of dry air at 70 F and under standard atmospheric pressure (29.921 in. of Hg.) is approximately 0.075 lb (see Table 1), while that of a mixture of air and saturated water vapor at the same temperature and barometric pressure is only about 0.0742 lb. In the mixture the density of the dry air is 0.07307 and that of the vapor is 0.00115 lb (see Table 2).

In order to make comparisons of air volumes or velocities it is necessary to reduce the observations to a common pressure and temperature basis. The basic pressure is usually taken as 29.921 in. of Hg., but no basic temperature is universally recognized. Common temperatures for this purpose are 32 F, 60 F, 68 F, and 70 F. Since 70 F is the most commonly specified temperature to which rooms for human occupancy must be heated, it is usually understood, when no other temperature is specified, that 70 F is the basic temperature for measuring the volume or the velocity of air in heating and ventilating work.

The *specific volume* of air is the volume in cubic feet occupied by one pound of the air. Under constant pressure the specific volume varies inversely as the density and directly as the absolute temperature.

TABLE 1. PROPERTIES OF DRY AIR[a]

Barometric Pressure 29 921 In of Hg.

Temperature Deg F	Weight Pounds per Cu Ft	Ratio of Volume to Volume at 70 F	Btu Absorbed by One Cu Ft Dry Air per Deg F	Cu Ft Dry Air Warmed One Deg per Btu
0	0 08633	0.8678	0 02077	48 15
10	0 08449	0 8867	0 02030	49.26
20	0 08273	0 9056	0 01986	50 35
30	0 08104	0 9245	0 01944	51.44
40	0.07942	0 9433	0 01905	52 49
50	0.07785	0 9624	0 01868	53.36
60	0 07636	0.9811	0.01832	54.44
70	0 07492	1.0000	0 01798	55.62
80	0 07353	1.0189	0.01765	56.66
90	0.07219	1.0378	0 01733	57.70
100	0 07090	1 0567	0 01702	58 75
110	0 06966	1 0755	0 01672	59.81
120	0 06845	1 0946	0 01643	60.86
130	0 06729	1 1133	0.01616	61 88
140	0 06617	1.1322	0 01589	62.93
150	0 06509	1 1510	0 01563	63 98
160	0 06403	1 1701	0 01538	65.02
180	0 06203	1.2078	0 01490	67.11
200	0 06015	1.2456	0 01446	69 24
220	0 05838	1 2832	0 01403	71.27
240	0 05671	1 3211	0 01364	73 31
260	0.05514	1.3587	0 01326	75.41
280	0 05365	1.3965	0 01291	77 46
300	0 05223	1 4344	0 01257	79 55
350	0.04901	1.5287	0 01181	84 67
400	0.04615	1.6234	0 01114	89 67
450	0 04362	1 7176	0.01054	94 87
500	0 04135	1.8119	0 01001	99.01
550	0 03930	1.9064	0 00953	104.93
600	0 03744	2 0011	0.00908	110.13
700	0.03422	2 1893	0 00833	120 05
800	0.03150	2.3784	0.00769	130.04
900	0 02911	2 5737	0 00713	140.25
1000	0.02718	2 7564	0 00668	149 70

[a]Compiled by W H Severns, based on the instantaneous specific heats of air The values for the heats and for the cubic feet warmed one degree are for the temperatures stated and are not true over a temperature range of more than one degree above or below the temperatures stated

TABLE 2. PROPERTIES OF SATURATED AIR[a]

Weights of Air, Vapor and Saturated Mixture of Air and Vapor at 29 921 In of Hg.

TEMP DEG F	WEIGHT IN A CUBIC FOOT OF MIXTURE: Weight of Dry Air Pounds	Weight of Vapor Pounds	Total Weight of Mixture Pounds	BTU ABSORBED BY ONE CUBIC FOOT SAT AIR PER DEG F	CUBIC FEET SAT AIR WARMED ONE DEG PER BTU	SPECIFIC HEAT BTU PER POUND OF MIXTURE
0	0 08622	0 000068	0 08629	0.02078	48.12	0 2408
10	0 08431	0 000111	0 08442	0.02031	49.24	0.2406
20	0 08244	0 000177	0 08262	0.01987	50 33	0.2405
30	0.08060	0 000278	0.08088	0 01946	51.39	0.2406
40	0.07876	0 000409	0 07917	0 01908	52.41	0 2410
50	0 07692	0.000587	0 07751	0.01872	53 42	0 2415
60	0.07503	0 000828	0 07586	0 01838	54 41	0 2423
70	0.07307	0.001151	0 07422	0.01805	55.40	0 2432
80	0.07099	0 001578	0 07257	0 01775	56 34	0.2446
90	0.06877	0.002134	0 07090	0 01747	57 24	0 2464
100	0 06634	0 002851	0 06919	0 01721	58.11	0 2487
110	0 06361	0 003762	0 06737	0 01696	58.96	0 2517
120	0 06057	0 004912	0 06548	0 01675	59.70	0 2558
130	0 05712	0 006344	0 06346	0 01657	60 35	0 2611
140	0 05317	0 008116	0 06129	0.01642	60 91	0.2679
150	0.04863	0.010284	0 05891	0 01630	61.35	0 2767
160	0.04339	0 012919	0 05631	0 01624	61 58	0 2884
170	0 03733	0 016092	0 05342	0 01621	61 69	0 3034
180	0.03033	0 019888	0 05022	0 01624	61 58	0 3234
190	0.02228	0 024384	0 04666	0 01633	61 24	0.3500
200	0 01298	0 029700	0 04268	0 01649	60 64	0 3864
210	0 00230	0 035932	0 03616	0 01672	59 81	0 4624
212	0 00000	0 037286	0 03729	0 01818	55 01	0 4875

[a]Compiled by W H Severns, based on the instantaneous specific heats of air

TABLE 3 SPECIFIC HEATS OF DRY AIR[a]

Constant Barometric Pressure of 29 921 In of Hg

TEMPERATURE DEG F	INSTANTANEOUS OR TRUE SPECIFIC HEAT	TEMPERATURE RANGE DEG F	MEAN SPECIFIC HEAT
−301.0	0 2520	32 to 212	0 2401
−108.4	0 2430	32 to 392	0 2411
32.0	0 2399	32 to 752	0 2420
212.0	0 2403	32 to 1112	0.2430
392.0	0.2413	....	....
752.0	0 2430	....	....
1112.0	0 2470	....	....

[a]Compiled by W H Severns, based on data given in the *International Critical Tables*

TABLE 4. WEIGHT OF SATURATED AND PARTLY SATURATED AIR[a]

DRY-BULB TEMP DEG F	WEIGHT OF SATURATED AIR FOR VARIOUS BAROMETRIC AND HYGROMETRIC CONDITIONS—POUNDS PER CUBIC FOOT							APPROX AVERAGE INCREASE IN WEIGHT PER DEG WET-BULB DEPRESSION
	Barometric Pressure Inches of Mercury						Increase In Weight Per 0.1 in Rise in Barometer	
	28.5	29.0	29.5	30.0	30.5	31.0		
30	0.07703	0.07839	0.07974	0.08110	0.08245	0.08381	0.00027	0.000017
32	0.07671	0.07806	0.07940	0.08075	0.08210	0.08345	0.00027	0.000017
34	0.07638	0.07772	0.07907	0.08041	0.08175	0.08310	0.00027	0.000018
36	0.07605	0.07739	0.07873	0.08007	0.08141	0.08274	0.00027	0.000018
38	0.07573	0.07706	0.07840	0.07973	0.08106	0.08239	0.00027	0.000019
40	0.07541	0.07674	0.07806	0.07939	0.08072	0.08205	0.00027	0.000019
42	0.07509	0.07641	0.07773	0.07905	0.08038	0.08170	0.00026	0.000020
44	0.07477	0.07609	0.07740	0.07872	0.08004	0.08135	0.00026	0.000020
46	0.07445	0.07576	0.07707	0.07838	0.07970	0.08101	0.00026	0.000021
48	0.07413	0.07544	0.07674	0.07805	0.07936	0.08066	0.00026	0.000021
50	0.07381	0.07512	0.07642	0.07772	0.07902	0.08032	0.00026	0.000022
52	0.07350	0.07479	0.07609	0.07739	0.07868	0.07998	0.00026	0.000023
54	0.07318	0.07447	0.07576	0.07706	0.07835	0.07964	0.00026	0.000023
56	0.07287	0.07415	0.07544	0.07673	0.07801	0.07930	0.00026	0.000024
58	0.07255	0.07383	0.07512	0.07640	0.07768	0.07896	0.00026	0.000025
60	0.07224	0.07352	0.07479	0.07607	0.07734	0.07862	0.00026	0.000026
62	0.07193	0.07320	0.07447	0.07574	0.07701	0.07828	0.00026	0.000027
64	0.07161	0.07288	0.07414	0.07541	0.07668	0.07794	0.00026	0.000028
66	0.07130	0.07256	0.07382	0.07508	0.07634	0.07760	0.00026	0.000029
68	0.07098	0.07224	0.07350	0.07475	0.07601	0.07727	0.00026	0.000030
70	0.07067	0.07192	0.07317	0.07442	0.07568	0.07693	0.00026	0.000031
72	0.07035	0.07160	0.07285	0.07410	0.07534	0.07659	0.00025	0.000032
74	0.07004	0.07128	0.07252	0.07377	0.07501	0.07625	0.00025	0.000033
76	0.06972	0.07096	0.07220	0.07343	0.07467	0.07591	0.00025	0.000034
78	0.06940	0.07064	0.07187	0.07310	0.07434	0.07557	0.00025	0.000036
80	0.06909	0.07032	0.07155	0.07277	0.07400	0.07523	0.00025	0.000037
82	0.06877	0.07000	0.07122	0.07244	0.07366	0.07489	0.00024	0.000039
84	0.06845	0.06967	0.07089	0.07211	0.07333	0.07454	0.00024	0.000040
86	0.06812	0.06934	0.07056	0.07177	0.07299	0.07420	0.00024	0.000042
88	0.06780	0.06901	0.07022	0.07143	0.07264	0.07385	0.00024	0.000043
90	0.06748	0.06868	0.06989	0.07109	0.07230	0.07351	0.00024	0.000045
92	0.06715	0.06835	0.06955	0.07075	0.07195	0.07316	0.00024	0.000047
94	0.06682	0.06801	0.06921	0.07041	0.07161	0.07280	0.00024	0.000049
96	0.06648	0.06768	0.06887	0.07006	0.07126	0.07245	0.00024	0.000051
98	0.06615	0.06734	0.06853	0.06972	0.07091	0.07209	0.00024	0.000053
100	0.06581	0.06700	0.06818	0.06937	0.07055	0.07174	0.00024	0.000055

[a]Approximate average decrease in weight per 0.1 F rise in dry-bulb temperature equals 0.000017 lb per cubic foot.

The *specific heat* of air is the number of Btu required to raise the temperature of 1 lb of air 1 F. Distinction should always be made between the instantaneous specific heat at any existent temperature and the mean specific heat, which is the average specific heat through a given temperature range. The mean specific heat is the value required in most calculations. The specific heats at constant pressure, C_p, and the specific heats, C_v, at constant volume are different. The specific heat at constant pressure is commonly used and it varies, under a pressure of one atmosphere, from a minimum at 32 F from which it increases with either increase or decrease of temperature. The value of 0.24, as the mean specific heat at constant pressure, is sufficiently accurate for use at ordinary temperatures. Values for instantaneous and mean specific heats are given n Table 3.

The *mean specific heat* of *water vapor* at constant pressure is taken as).45 for all general engineering computations.

Table 4 is intended to aid in determining the density of moist air, aking into account its temperature, pressure, and moisture content.

Example 1 To show the use of Table 4: Given air at 83 F dry-bulb and 68 F wet-ulb (or a depression of 15 deg) with a barometric pressure of 29.40 in of mercury. Vhat will be the weight of this air in pounds per cubic foot?

Solution. From Table 4 the weight of saturated air at 82 F and 29.00 in barometer is ound to be 0.07000 lb per cubic foot. There is a decrease of 0.00017 lb per degree dry-ulb temperature above 82 F. There is an increase of 0.00024 lb for each 0 1 in. above 9 00 in. From the last column of Table 4 it is found that there is an increase of approxi-lately 0 000039 lb per degree wet-bulb depression when the dry-bulb is 83 F. Tabu-iting the items:

0 07000	=	weight of saturated air at 82 F and 29 00 bar.
— 0.00017	=	decrement for 1 deg dry-bulb, 1 × 0 00017.
+ 0 00096	=	increment for 0.4 in. bar., 4 × 0.00024.
+ 0 00059	=	increment for 15 deg wet-bulb depression, 15 × 0 000039.
0 07138	=	weight in pounds per cubic foot of air at 83 F dry-bulb, 68 F wet-bulb, 29.40 in. bar.

It is usual to assume that dry air, moist air, and the water vapor in the ir follow the laws of perfect gases. This assumption while not absolutely ue, especially with saturated vapor at temperatures much above 140 F, sufficiently accurate for practical purposes and it greatly simplifies omputations.

Boyle's Law refers to the relation between the pressure and volume of a as, and may be stated as follows: *With temperature constant, the volume of given weight of gas varies inversely as its absolute pressure.* Hence, if $_1$ and P_2 represent the initial and final absolute pressures, and V_1 and $_2$ represent corresponding volumes of the same mass, say one pound of is, then $\frac{V_1}{V_2} = \frac{P_2}{P_1}$, or $P_1 V_1 = P_2 V_2$, but since $P_1 V_1$ for any given case is definite constant quantity, it follows that the product of the absolute essure and volume of a gas is a constant, or $PV = C$, when T is kept nstant. Any change in the pressure and volume of a gas at constant mperature is called an *isothermal change.*

Charles' Law refers to the relation among pressure, volume, and temperature of a gas and may be stated as follows: *The volume of a given*

weight of gas varies directly as the absolute temperature at constant pressure, and the pressure varies directly as the absolute temperature at constant volume. Hence, when heat is added at constant volume, V_c, the resulting equation is $\frac{P_2}{P_1} = \frac{T_2}{T_1}$, or, for the same temperature range at constant pressure, P_c, the relation is $\frac{V_2}{V_1} = \frac{T_2}{T_1}$.

In general, for any weight of gas, W, since volume is proportional to weight, the relation among P, V, and T is

$$PV = WRT \tag{3}$$

where

P = the absolute pressure of the gas, pounds per square foot
V = the volume of the weight W, cubic feet.
W = the weight of the gas, pounds.
R = a constant depending on the nature of the gas The average value of R for air is 53.34.
T = the absolute temperature, degrees Fahrenheit.

This is the characteristic equation for a perfect gas, and while no gases are perfect in this sense, they conform so nearly that Equation 3 will apply to most engineering computations.

HUMIDITY

Humidity is the water vapor mixed with dry air in the atmosphere. *Absolute humidity* has a multiplicity of meanings, but usually the term refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. With this meaning, absolute humidity is nothing but the actual density of the water vapor in the mixture and might better be so called A study of the Properties of Saturated Steam in Table 8 indicates that water vapor, either saturated or superheated, at partial pressures lower than 4 in. of mercury may be treated as a gas with a gas constant R of 1.21 (with partial pressure of vapor expressed in inches of Hg.) in the characteristic equation of the gas $pV = wR\ (t + 460)$. Within such limits, the density (d) of water vapor is

$$d = \frac{w}{V} = \frac{e}{1.21\ (t + 460)} \text{ (pounds per cubic foot)} \tag{4a}$$

$$= \frac{5785\ e}{t + 460} \text{ (grains per cubic foot)} \tag{4b}$$

where

e = actual partial pressure of vapor, inches of mercury
t = dry-bulb temperature, degrees Fahrenheit

Specific Humidity

It simplifies many problems which deal with mixtures of dry air and water vapor to express the weight or the mass of the vapor in terms of the weight or the mass of dry air. If the weight of the water vapor in a mixture be divided by the weight of the dry air, and the weight of dry air

be made unity, we have an expression of the weight of water vapor carried by a unit weight of dry air. This relation has no generally accepted name. It has been variously called: mixing ratio, proportionate humidity, mass or density ratio, absolute humidity, and specific humidity. Of all these terms *specific humidity* is the most suggestive of the meaning which it is desired to express and it has found considerable use in this sense even though it is defined in *International Critical Tables* as the ratio of the mass of vapor to the total mass. It will be understood here that *specific humidity* refers to the weight of water vapor carried by one pound of dry air.

The gas constant for dry air, when the partial pressure of the air is expressed in inches of Hg., is 0.753; so that the specific humidity, if represented by W, is

$$W = \frac{e}{1.21\ (t + 460)} \div \frac{B\text{-}e}{0.753\ (t + 460)}$$

$$= 0.622 \left(\frac{e}{B\text{-}e}\right) \text{ (pounds)} \tag{5a}$$

$$= 4354 \left(\frac{e}{B\text{-}e}\right) \text{ (grains)} \tag{5b}$$

where

e = actual partial pressure of vapor, inches of mercury.
B = total pressure of mixture (barometric pressure), inches of mercury.

Relative Humidity

Relative humidity (Φ) is either the ratio of the actual partial pressure, e, of the water vapor in the air to the saturation pressure, e_t, at the dry-bulb temperature, or the ratio of the actual density, d, of the vapor to the density of saturated vapor, d_t, at the dry-bulb temperature. That is:

$$\Phi = \frac{e}{e_t} = \frac{d}{d_t} \tag{6}$$

The relative humidity of a given mixture at a given temperature is not the same as the specific humidity, W, of the mixture divided by the specific humidity, W_t, of saturated vapor at the same temperature, for from Equations 5a and 6

$$\frac{W}{W_t} = 0.622 \left(\frac{\Phi\ e_t}{B - \Phi\ e_t}\right) \div 0.622 \left(\frac{e_t}{B\text{-}e_t}\right) = \frac{\Phi\ (B\text{-}e_t)}{B - \Phi\ e_t} \tag{7}$$

The specific humidity of an unsaturated air-vapor mixture cannot, therefore, be accurately found by multiplying the specific humidity of saturated air by its relative humidity; although the error is usually small especially when the relative humidity is high.

With a relative humidity of 100 per cent, the dry-bulb, wet-bulb, and dew-point temperatures are equal. With a relative humidity less than 100 per cent, the dry-bulb exceeds the wet-bulb, and the wet-bulb exceeds the dew-point temperature.

RELATION OF DEW-POINT TO RELATIVE HUMIDITY

A peculiar relationship exists between the dew-point and the relative humidity and this is found most useful in air conditioning work. This

relationship is, that for a fixed relative humidity there is substantially a constant difference between the dew-point and the dry-bulb temperature over a considerable temperature range. Table 5, giving the dry-bulb and dew-point temperatures and the dew-point differentials for 50 per cent relative humidity, illustrates this relationship clearly.

TABLE 5. TEMPERATURES FOR 50 PER CENT RELATIVE HUMIDITY

Dry-bulb temperature	65.0	70.0	75.0	80.0	85.0	90.0
Dew-point temperature	45.8	50.5	55.25	59.75	64.25	68.75
Difference between dew-point and dry-bulb temperature	19.2	19.5	19.75	20.25	20.75	21.25

It will be seen from an inspection of this table that the difference between the dew-point temperature and the room temperature is approximately 20 deg throughout this range of dry-bulb temperatures or, to be more exact, the differential increases only 10 per cent for a range of practically 25 deg.

This principle holds true for other humidities and is due to the fact that the pressure of the water vapor practically doubles for every 20 deg through this range.

The approximate relative humidity for any difference between dew-point and dry-bulb temperature may be expressed in per cent as:

$$\frac{100}{2^{\frac{t-t_1}{20}}} \tag{8}$$

where

t_1 = dew-point temperature.

This principle is very useful in determining the available cooling effect obtainable with saturated air when a desired relative humidity is to be maintained in a room, even though there may be a wide variation in room temperature. This problem is one which applies to certain industrial conditions, such as those in cotton mills and tobacco factories, where relatively high humidities are carried and where one of the principal problems is to remove the heat generated by the machinery. It also permits the use of a differential thermostat, responsive to both the room temperature and the dew-point temperature, to control the relative humidity in the room.

Table 6 gives, for different temperatures, the density of saturated vapor, d_t, the weight of saturated vapor mixed with 1 lb of dry air, W_t, (at a relative humidity of 100 per cent and a barometric pressure, B, of 29.92 in. of mercury), the specific volume of dry air, and the volume of an air-vapor mixture containing 1 lb of dry air (at a relative humidity of 100 per cent and a pressure of 29.92 in. of mercury). The preceding equations or the data from Table 6 may be conveniently used in solving the following typical problems:

Example 2. Humidifying and Heating. **Air is to be maintained at 70 F with a relative humidity of 40 per cent (Φ = 0.4) when the outside air is at 0 F and 70 per cent**

relative humidity ($\Phi = 0.7$) and a barometric pressure, B, of 29.92 in. of mercury. Find the weight of water vapor added to each pound of dry air and the dew-point temperature of the humidified air.

Solution. From Equation 5a and Table 6,

$$W_1 = 0.622 \left(\frac{0.7 \times 0.03773}{29.92 - 0.0264}\right) = 0.000548 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622 \left(\frac{0.4 \times 0.7386}{29.92 - 0.295}\right) = 0.00618 \text{ lb per pound of dry air.}$$

The water vapor added per pound of dry air must be $(W_2 - W_1)$ or 0.005632 lb. By inspection of Table 6, $W_t = 0.00618$ at 44.5 F, so this is the dew-point temperature of the humidified air.

An approximation of the same result from Table 6 is

$W_1 = 0.7 \times 0.0007852 = 0.00054964$ lb per pound of dry air.
$W_2 = 0.4 \times 0.01574 = 0.006296$ lb per pound of dry air.

The water vapor added per pound of dry air is approximately 0.00574636 lb and the dew-point temperature is approximately 45 F. The degree of approximation is evident.

Example 3. Dehumidifying and Cooling. Air with a dry-bulb temperature of 84 F, a wet-bulb of 70 F, or a relative humidity of 50 per cent ($\Phi = 0.5$), and a barometric pressure, B, of 29.92 in. of mercury is to be cooled to 54 F. Find the dew-point temperature of the entering air and the weight of vapor condensed per pound of dry air.

Solution. From Equation 5a and Table 6,

$$W_1 = 0.622 \left(\frac{0.5 \times 1.1752}{29.92 - 0.5876}\right) = 0.01248 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622 \left(\frac{0.42003}{29.92 - 0.42003}\right) = 0.00887 \text{ lb per pound of dry air.}$$

Since $W_1 = W_t$ when $t = 63.4$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is $(W_1 - W_2)$ or 0.00361 lb per pound of dry air.

An approximate result is

$W_1 = 0.5 \times 0.02543 = 0.012715$ lb per pound of dry air.
$W_2 = 1 \times 0.008856 = 0.008856$ lb per pound of dry air, since the exit air is saturated.

Since $W_1 = W_t$ at $t = 64$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is 0.003859 lb per pound of dry air. The degree of approximation is again evident.

Since Table 6 was prepared, the new steam tables, *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, have been published. For the last two years an A.S.H.V.E. Research Technical Advisory Committee on Psychrometry has been formulating standard psychrometric data, on a tentative basis. Included in this Committee's work is a revision of Table 6 to bring it into conformity with the Keenan and Keyes tables. Pending acceptance by the Society of the Committee's report, the tabular data and psychrometric chart published in earlier editions of THE GUIDE have been retained.

ADIABATIC SATURATION OF AIR

The process of adiabatic saturation, or evaporative cooling, is of considerable importance in air conditioning. Suppose that unsaturated air be made to pass at a steady rate through a tunnel which is perfectly insulated

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I)

TEMP, F	PRESSURE OF SATURATED VAPOR $\times 10^6$		WEIGHT OF SATURATED VAPOR				VOLUME IN CU FT BAROMETER, 29 92 IN. HG.		ENTHALPY PER LB		
	In. of Hg.	Lb per Sq In	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds $\times 10^6$	Grains	Pounds $\times 10^6$	Grains					
−130	0 276	0.1356	0.000693	0 000049	0.005738	0 00040	8.31	8.31	−31.71	1000 7	−31.71
−129	.306	.1503	.000766	.000054	.006362	.00045	8.33	8 33	−31.46	1001.2	−31.46
−128	.338	.1660	.000843	.000059	.007027	.00049	8.36	8 36	−31.21	1001.6	−31.21
−127	.373	.1832	.000928	.000065	.007755	.00054	8.38	8 38	−30.96	1002.1	−30.96
−126	.411	.2019	.001019	.000071	.008545	.00060	8.41	8.41	−30.71	1002.5	−30.71
−125	0 455	0.2235	0.001125	0 000079	0.009459	0.00066	8.43	8.43	−30.46	1003.0	−30.46
−124	.499	.2451	.001230	.000086	.01037	.00073	8 46	8 46	−30.21	1003.4	−30.21
−123	.542	.2662	.001332	.000093	.01127	.00079	8.48	8 48	−29 96	1003.9	−29.96
−122	.604	.2967	.001480	.00010	.01256	.00088	8 51	8 51	−29.72	1004.3	−29.72
−121	.669	.3286	.001635	.00012	.01391	.00097	8.53	8.53	−29.47	1004.8	−29.47
−120	0.735	0.3610	0.001791	0.00013	0.01528	0.00107	8.56	8.56	−29.22	1005.2	−29.22
−119	.805	.3954	.001956	.00014	.01674	.00117	8 58	8.58	−28.97	1005.7	−28.97
−118	.892	.4382	.002161	.00015	.01854	.00130	8 61	8.61	−28.72	1006.1	−28.72
−117	.989	.4858	.002388	.00017	.02056	.00144	8 63	8.63	−28 47	1006.6	−28.47
−116	1.098	.5393	.002644	.00019	.02283	.00160	8.66	8.66	−28.23	1007.0	−28.23
−115	1.208	0.5934	0.002900	0 00020	0 02511	0 00176	8.68	8.68	−27.98	1007.5	−27.98
−114	1.317	.6469	.003153	.00022	.02738	.00192	8 71	8.71	−27.73	1007.9	−27.73
−113	1.444	.7093	.003447	.00024	.03002	.00210	8.73	8.73	−27.48	1008.4	−27.48
−112	1.575	.7736	.003749	.00026	.03274	.00229	8.76	8.76	−27.23	1008.8	−27.23
−111	1.728	.8488	.004101	.00029	.03593	.00252	8.78	8.78	−26.99	1009.3	−26.99
−110	1 889	0 9279	0 004471	0 00031	0.03927	0 00275	8.81	8 81	−26.74	1009.7	−26.74
−109	2.087	1 0251	.004925	.00035	.04339	.00304	8.83	8.83	−26.49	1010 2	−26.49
−108	2.292	1.1258	.005393	.00038	.04765	.00334	8 86	8.86	−26.24	1010.6	−26.24
−107	2.511	1.2334	005892	.00041	.05220	.00365	8.89	8 89	−26.00	1011.1	−26.00
−106	2.742	1.3469	.006415	.00045	.05701	.00399	8.91	8.91	−25.75	1011.5	−25.75
−105	2.983	1.4652	0 006960	0.00049	0 06202	0 00434	8.94	8.94	−25.50	1012.0	−25.50
−104	3.258	1 6003	.007580	.00053	06773	.00474	8.96	8.96	−25.26	1012.4	−25.26
−103	3.543	1 7403	.008219	.00058	.07366	.00516	8.99	8 99	−25.01	1012.9	−25 01
−102	3 872	1 9019	.008958	.00063	.08050	.00564	9.01	9 01	−24.76	1013.3	−24.76
−101	4.213	2.0694	.009719	.00068	.08759	.00613	9.04	9 04	−24.51	1013.8	−24.51
−100	4.607	2.2630	0.010599	0 00074	0.09578	0.00705	9.06	9.06	−24.27	1014.2	−24.27
−99	5.018	2.4637	.011512	.00081	.1043	.00730	9.09	9.09	−24.02	1014.7	−24.02
−98	5.455	2.6784	.012480	.00087	.1134	.00794	9 11	9.11	−23.78	1015.1	−23.78
−97	5.946	2.9196	.013566	.00095	.1236	.00865	9.14	9.14	−23.53	1015.6	−23.53
−96	6.470	3.1781	.014721	.00103	.1345	.00942	9.16	9 16	−23.28	1016.0	−23.28

[a]Compiled by W. M Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I, CONTINUED)

Temp, F	Pressure of Saturated Vapor × 10⁵		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In Hg		Enthalpy per Lb		
	In. of Hg.	Lb per Sq In	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds × 10⁶	Grains	Pounds × 10⁵	Grains					
−95	7.047	3 4604	0 015990	0 00112	0 1465	0 01026	9 19	9.19	−23 04	1016.5	−23.04
−94	7.638	3 7507	.017284	.00121	.1588	.01112	9 21	9.21	−22 79	1016.9	−22.79
−93	8.316	4.0837	.018767	.00131	.1729	.01210	9 23	9.23	−22 55	1017.4	−22.55
−92	9.017	4 4281	020292	00142	.1875	.01312	9.26	9.26	−22 30	1017 8	−22.30
−91	9.806	4 8156	022009	.00154	.2039	.01427	9.29	9.29	−22.05	1018.3	−22.05
−90	10 64	5 2264	0 023817	0 00167	0 2212	0 01548	9 31	9 31	−21.81	1018 7	−21.81
−89	11.53	5.6635	025738	.00180	2397	.01678	9 34	9.34	−21 56	1019 2	−21.56
−88	12.51	6.1449	.027851	.00195	2601	.01821	9.36	9.36	−21 32	1019.6	−21.32
−87	13.53	6.6459	030041	.00210	2813	.01969	9.39	9.39	−21.07	1020.1	−21.07
−86	14.69	7.2157	.032530	.00228	.3054	.02138	9.41	9.41	−20 83	1020.5	−20.83
−85	15 87	7 7953	0 035049	0 00245	0 3299	0 02309	9.44	9 44	−20 58	1021 0	−20.58
−84	17.20	8 4486	.037885	.00265	3576	.02503	9 46	9.46	−20 34	1021 4	−20 34
−83	18.58	9 1265	040817	00286	3863	02704	9.49	9 49	−20.09	1021.9	−20 09
−82	20 10	9.8731	.044037	.00308	.4179	.02925	9 51	9 51	−19.84	1022.3	−19 84
−81	21.72	10.669	047463	.00332	4516	.03161	9.54	9.54	−19.60	1022 8	−19.60
−80	23 47	11.517	0.051151	0.00358	0 4879	0.03415	9 56	9.56	−19 36	1023.2	−19.36
−79	25 34	12.436	055082	.00386	5268	.03688	9 59	9 59	−19 11	1023 7	−19.10
−78	27 29	13.394	059165	.00414	.5674	03972	9 61	9 61	−18 87	1024.1	−18.86
−77	29.52	14.489	.063831	.00447	.6137	.04296	9 64	9.64	−18.62	1024.6	−18.61
−76	31.81	15.614	068605	.00480	.6613	04629	9.66	9 66	−18 38	1025 0	−18.37
−75	34.37	16 883	0 073933	0.00518	0 7146	0 05002	9.69	9 69	−18 13	1025 5	−18 12
−74	37.01	18.179	079405	.00556	.7694	.05386	9.72	9.72	−17 89	1025.9	−17.88
−73	39 96	19.628	085510	.00599	.8308	.05816	9.74	9.74	−17.64	1026.4	−17.63
−72	43.04	21 141	.091865	.00643	.8948	.06264	9.77	9.77	−17 40	1026.8	−17 39
−71	46.33	22.757	.098632	.00690	.9632	.06742	9.79	9.79	−17 16	1027.3	−17.15
−70	49.87	24.496	0 10590	0 00741	1 037	0.07259	9.82	9 82	−16 91	1027.7	−16.90
−69	53 59	26 323	11350	00795	1 114	.07798	9 84	9 84	−16 67	1028.2	−16.66
−68	57.65	28 318	.12179	.00853	1.199	.08393	9.87	9 87	−16 42	1028.6	−16 41
−67	61 81	30.361	13024	.00911	1 285	.08995	9 89	9 89	−16 18	1029.1	−16.17
−66	66 41	32.621	13959	.00977	1 381	.09667	9.92	9.92	−15 94	1029.5	−15.93
−65	71.17	34 959	0 14922	0 01044	1 480	0 10360	9 94	9.94	−15 69	1030 0	−15 67
−64	76 64	37 646	.16028	.01122	1.593	.11151	9.97	9.97	−15.45	1030 4	−15.43
−63	82 28	40.416	17164	.01201	1.711	.11977	9 99	9.99	−15 21	1030.9	−15.19
−62	88.19	43 319	18350	.01285	1.833	.12831	10 02	10.02	−14.96	1031.3	−14.94
−61	94.62	46 477	.19638	.01375	1.967	.13769	10 04	10.04	−14.72	1031.8	−14.70

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I, CONTINUED)

Temp. F	Pressure of Saturated Vapor × 10⁵		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In. Hg.		Enthalpy per Lb		
	In of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds × 10^5	Grains	Pounds × 10^5	Grains					
−60	101.4	49 808	0.20993	0 01470	2 108	0.14756	10 07	10.07	−14.48	1032.2	−14.46
−59	108 8	53 443	.22469	.01573	2 262	.15834	10 09	10 09	−14.23	1032.7	−14.21
−58	116.3	57 127	.23958	.01677	2.418	.16926	10 12	10 12	−13.99	1033.1	−13.97
−57	124.8	61.302	.25645	.01795	2 595	.18165	10 14	10 14	−13.75	1033.6	−13.72
−56	133.4	65.526	.27344	.01914	2.773	.19411	10.17	10.17	−13.50	1034.0	−13.47
−55	143 0	70 242	0 29239	0.02047	2 973	0 20811	10.19	10 19	−13.26	1034.5	−13.23
−54	153.0	75 154	.31207	02184	3 181	22267	10 22	10.22	−13 02	1034.9	−12 99
−53	163.5	80 311	.33267	.02329	3 399	.23793	10.24	10.24	−12.78	1035.4	−12.74
−52	174.9	85 911	.35499	.02485	3 636	.25452	10 27	10.27	−12.53	1035.8	−12.49
−51	187.0	91 854	37862	02650	3 888	.27216	10 29	10.29	−12.29	1036.3	−12.25
−50	199.9	98 191	0.40376	0 02826	4 156	0 29092	10.32	10.32	−12 05	1036.7	−12.01
−49	213.0	104 63	.42917	.03004	4 428	.30996	10 34	10.34	−11 81	1037.2	−11 76
−48	227.9	111 94	.45808	.03207	4 738	.33166	10 37	10 37	−11.57	1037.6	−11.52
−47	243.1	119 41	.48744	03412	5 054	35378	10 40	10 40	−11 32	1038.1	−11.27
−46	259.5	127 47	.51905	.03633	5 395	.37765	10 42	10.42	−11 08	1038.5	−11.02
−45	276 7	135 92	0 55213	0 03865	5.753	0 40271	10 45	10.45	−10 84	1039.0	−10.78
−44	295.0	144 90	.58722	.04111	6 133	.42931	10 47	10 47	−10 60	1039.4	−10 54
−43	314.7	154 58	.62493	.04375	6 543	.45801	10 50	10 50	−10 35	1039.9	−10.28
−42	335 3	164 70	.66426	.04650	6 971	.48797	10 52	10 52	−10.11	1040.3	−10.04
−41	357.6	175 65	.70672	.04947	7.435	.52045	10.55	10.55	−9.872	1040.8	−9.795
−40	380.3	186 80	0 7498	0 05249	7 907	0 55349	10 57	10 57	−9 629	1041.2	−9.547
−39	405.5	199.18	.7976	05583	8 431	.59017	10 60	10 60	−9 388	1041.7	−9.300
−38	431 2	211 81	.8461	05922	8 965	62755	10 62	10.62	−9.146	1042.1	−9.053
−37	459.2	225 56	.8989	06292	9 548	.66836	10 65	10.65	−8.905	1042.6	−8.805
−36	488.4	239.90	.9538	.06677	10 16	.71120	10 67	10.67	−8.663	1043.0	−8.557
−35	519 5	255.18	1.0122	0.07085	10 80	0.75600	10 69	10 69	−8.422	1043.5	−8.309
−34	552.4	271 34	1.0738	.07517	11.49	.80430	10.72	10.72	−8.180	1043 9	−8.060
−33	586.5	288 09	1 1374	.07962	12 20	.85400	10 75	10 75	−7.939	1044.4	−7.812
−32	623.7	306 36	1 2067	.08447	12 97	.90790	10 77	10 77	−7.698	1044.8	−7.562
−31	661.8	325.08	1 2774	08942	13 76	.96320	10 80	10 80	−7.457	1045.3	−7.313
−30	701 0	344 33	1.3499	0 09449	14 58	1.0206	10 82	10 82	−7.216	1045.7	−7.064
−29	742 2	364 57	1 4260	09982	15 43	1 0801	10 85	10 85	−6 975	1046.2	−6.814
−28	791.2	388 64	1.5166	.10616	16 45	1.1515	10 87	10 87	−6.734	1046 6	−6.562
−27	841.0	413.10	1 6083	.11258	17.49	1.2243	10 90	10.90	−6 493	1047.1	−6 310
−26	892.1	438 20	1.7021	.11914	18.55	1 2985	10 92	10 92	−6.251	1047.5	−6.057

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I, CONCLUDED)

Temp, F	Pressure of Saturated Vapor × 10⁵		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In Hg.		Enthalpy per Lb		
	In of Hg	Lb per Sq In	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds × 10⁵	Grains	Pounds × 10⁵	Grains					
−25	946 4	464.87	1 8016	0.12611	19 68	1.3776	10 95	10 95	−6 011	1048 0	−5.805
−24	1003.	492.67	1 9049	.13334	20 86	1 4602	10 97	10 97	−5 770	1048 4	−5.551
−23	1064.	522.64	2 0162	.14113	22 13	1.5491	11 00	11 00	−5.529	1048 9	−5.297
−22	1126.	553 09	2 1287	.14901	23 42	1.6394	11 02	11 02	−5 288	1049 3	−5.042
−21	1192.	585.51	2.2484	.15739	24.79	1 7353	11 05	11 05	−5 047	1049 8	−4.787
−20	1262.0	619.89	2 3750	0.16625	26 25	1 8375	11 07	11 07	−4 807	1050 2	−4.531
−19	1337.	656.73	2 5105	.17574	27 81	1.9467	11.10	11 10	−4 566	1050 7	−4.274
−18	1416.	695 54	2.6527	18569	29 45	2 0615	11.13	11 13	−4.325	1051 1	−4.015
−17	1496.	734 84	2.7963	.19574	31 12	2 1784	11 15	11 15	−4 085	1051 6	−3.758
−16	1584.	778 06	2.9542	.20679	32 95	2 3065	11 18	11 18	−3.844	1052 0	−3.497
−15	1675.0	822 76	3.1168	0.21818	34 84	2 4388	11.20	11 21	−3 604	1052 5	−3.237
−14	1772.	870 41	3.2899	.23029	36 86	2 5802	11.23	11 24	−3 363	1052 9	−2.975
−13	1874.	920 51	3 4714	.24300	38 98	2 7286	11 25	11 26	−3 123	1053 4	−2.712
−12	1980.	972 58	3 6596	.25617	41.19	2.8833	11 28	11 29	−2 883	1053 8	−2 449
−11	2093.	1028 1	3.8599	.27019	43.54	3.0478	11.30	11 31	−2.642	1054.3	−2.183
−10	2210.0	1085 6	4.0666	0 28466	45 98	3 2186	11 33	11 34	−2 402	1054 7	−1 917
−9	2335.	1147 0	4.2871	.30009	48 58	3 4006	11 35	11 36	−2 162	1055 2	−1.649
−8	2463.	1209 8	4.5120	.31584	51 25	3 5875	11 38	11 39	−1 921	1055 6	−1 380
−7	2502.	1229 0	4.5734	.32014	52 06	3 6442	11 40	11 41	−1 681	1056 1	−1.131
−6	2745.	1348.3	5.0066	.35046	57.12	3.9984	11 43	11 44	−1 441	1056 5	−0.8375
−5	2898 0	1423.5	5.2738	0.36917	60.30	4.2210	11.45	11 46	−1 201	1057.0	−0.5636
−4	3055.	1500 6	5 5473	.38831	63.57	4.4499	11 48	11.49	−0 9604	1057 4	−0 2882
−3	3222.	1582 6	5 8379	.40865	67 05	4.6935	11.50	11.51	−0 7203	1057 9	−0.01098
−2	3397.	1668 6	6 1414	.42990	70 69	4 9483	11.53	11 54	−0 4802	1058.3	+0.2679
−1	3580.	1758 5	6 4583	.45208	74 50	5.2150	11 55	11 57	−0 2401	1058.8	+0.5487
0	3773 0	1853.3	6 7914	0 47500	78 52	5 5000	11 58	11 59	0	1059 2	+0.8317

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II)

Temp, F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In Hg		Enthalpy per Lb		
	In of Hg	Lb per Sq In.	per Cu Ft, Pounds	per Cu Ft, Grains	per lb of Dry Air, Pounds	per lb of Dry Air, Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
0	0.03773	0.01853	0.000067914	0.475	0.0007852	5.50	11.58	11.59	0.0000	1059.2	0.8317
1	.03975	.01963	.000071395	.500	.0008275	5.79	11.60	11.62	.2401	1059.7	1.117
2	.04186	.02056	.000075021	.525	.0008714	6.10	11.63	11.64	.4801	1060.1	1.404
3	.04409	.02166	.000078851	.552	.0009179	6.43	11.65	11.67	.7201	1060.6	1.694
4	.04645	.02282	.000082890	.580	.0009671	6.77	11.68	11.70	.9601	1061.0	1.986
5	0.04886	0.02400	0.000087005	0.609	0.001017	7.12	11.70	11.72	1.200	1061.5	2.280
6	.05144	.02527	.000091399	.640	.001071	7.50	11.73	11.75	1.440	1061.9	2.577
7	.05412	.02658	.000095955	.672	.001127	7.89	11.75	11.77	1.680	1062.4	2.877
8	.05692	.02796	.00010070	.705	.001186	8.30	11.78	11.80	1.920	1062.8	3.180
9	.05988	.02941	.00010572	.740	.001247	8.73	11.80	11.83	2.160	1063.3	3.486
10	0.06295	0.03092	0.00011090	0.776	0.001311	9.18	11.83	11.85	2.400	1063.7	3.795
11	.06618	.03251	.00011634	.814	.001379	9.65	11.86	11.88	2.640	1064.2	4.108
12	.06958	.03418	.00012206	.854	.001450	10.15	11.88	11.91	2.880	1064.6	4.424
13	.07309	.03590	.00012794	.896	.001523	10.66	11.91	11.93	3.120	1065.1	4.742
14	.07677	.03771	.00013410	.939	.001600	11.20	11.93	11.96	3.359	1065.5	5.064
15	0.08067	0.03963	0.00014062	0.984	0.001682	11.77	11.96	11.99	3.599	1066.0	5.392
16	.08469	.04160	.00014732	1.031	.001766	12.36	11.98	12.01	3.839	1066.4	5.722
17	.08895	.04369	.00015440	1.081	.001855	12.99	12.00	12.04	4.079	1066.9	6.058
18	.09337	.04586	.00016174	1.132	.001947	13.63	12.03	12.07	4.319	1067.3	6.397
19	.09797	.04812	.00016935	1.185	.002043	14.30	12.06	12.09	4.559	1067.8	6.741
20	0.1028	0.05050	0.00017747	1.242	0.002144	15.01	12.08	12.12	4.798	1068.2	7.088
21	.1078	.05295	.00018564	1.299	.002250	15.75	12.11	12.15	5.038	1068.7	7.443
22	.1132	.05560	.00019439	1.361	.002361	16.53	12.13	12.18	5.278	1069.1	7.802
23	.1186	.05826	.00020335	1.423	.002476	17.33	12.16	12.20	5.518	1069.6	8.166
24	.1244	.06111	.00021276	1.489	.002596	18.17	12.18	12.23	5.758	1070.0	8.536
25	0.1304	0.06405	0.00022255	1.558	0.002722	19.05	12.21	12.26	5.998	1070.5	8.912
26	.1366	.06710	.00023278	1.629	.002853	19.97	12.23	12.29	6.237	1070.9	9.292
27	.1432	.07034	.00024342	1.704	.002991	20.94	12.26	12.32	6.477	1071.4	9.682
28	.1500	.07368	.00025445	1.781	.003133	21.93	12.28	12.34	6.717	1071.8	10.075
29	.1571	.07717	.00026597	1.862	.003283	22.99	12.31	12.37	6.957	1072.3	10.477
30	0.1645	0.08080	0.00027797	1.946	0.003439	24.07	12.33	12.40	7.197	1072.7	10.886
31	.1722	.08458	.00029043	2.033	.003601	25.21	12.36	12.43	7.437	1073.2	11.302
32	.1803	.08856	.00030343	2.124	.003771	26.40	12.38	12.46	7.677	1073.6	11.726
33	.1879	.09230	.00031471	2.203	.003931	27.52	12.41	12.49	7.917	1074.1	12.139
34	.1957	.09610	.00032690	2.288	.004094	28.66	12.43	12.51	8.157	1074.5	12.556

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp, F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In Hg		Enthalpy per Lb		
			per Cu Ft		per lb of Dry Air						
	In of Hg	Lb per Sq In	Pounds	Grains	Pounds	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
35	0 20360	0.1000	0 0003394	2.376	0 004262	29 83	12 46	12.54	8.397	1075.0	12 979
36	.21195	.1041	0003527	2 469	.004438	31.07	12 48	12.57	8 636	1075.4	13 409
37	.22050	.1083	0003662	2.563	.004618	32 33	12 51	12 60	8 876	1075.9	13.845
38	.22925	.1126	0003799	2 660	004803	33 62	12 53	12 63	9 116	1076 3	14.285
39	23842	.1171	.0003943	2 760	.004996	34 97	12.56	12.66	9.356	1076 8	14.736
40	0.24778	0 1217	0.0004090	2 863	0 005194	36 36	12 59	12 69	9 596	1077.2	15 191
41	.25755	.1265	.0004243	2 970	.005401	37 80	12.61	12.72	9.836	1077 7	15 657
42	.26773	.1315	.0004401	3 081	005616	39.31	12 64	12 75	10 08	1078 1	16 13
43	.27832	.1367	.0004566	3.196	.005840	40 88	12 66	12 78	10 32	1078 6	16 62
44	.28911	.1420	.0004735	3 315	006069	42 48	12.69	12 81	10 56	1079.0	17 11
45	0 30031	0 1475	0 0004909	3 436	0.006306	44 14	12 71	12 84	10 80	1079.5	17 61
46	.31191	.1532	0005088	3 562	006553	45.87	12.74	12 87	11 04	1079 9	18 12
47	.32393	1591	0005274	3.692	006808	47 66	12 76	12.90	11.28	1080 4	18 64
48	.33635	.1652	.0005465	3.826	007072	49.50	12 79	12.93	11.52	1080 8	19 16
49	.34917	.1715	.0005663	3.964	.007345	51 42	12 81	12.96	11 76	1081.3	19.70
50	0 36241	0 1780	0 0005866	4 106	0 007626	53 38	12 84	12 99	12.00	1081.7	20 25
51	37625	.1848	0006078	4 255	.007921	55 45	12 86	13 02	12 23	1082.2	20 80
52	.39051	.1918	.0006296	4 407	.008226	57 58	12 89	13 06	12 47	1082 6	21.38
53	.40496	1989	.0006516	4 561	008534	59 74	12 91	13 09	12 71	1083 1	21.95
54	.42003	2063	.0006746	4.722	.008856	61.99	12.94	13.12	12.95	1083.5	22.55
55	0.43570	0 2140	0.0006984	4 889	0 009192	64 34	12 96	13.15	13 19	1084.0	23 15
56	.45179	.2219	0007228	5 060	.009536	66 75	12 99	13 19	13.43	1084 4	23 77
57	.46828	.2300	.0007477	5.234	009890	69.23	13 01	13 22	13 67	1084.9	24.40
58	.48538	.2384	0007735	5 415	01026	71 82	13 04	13 25	13 91	1085.3	25.05
59	.50310	.2471	.0008003	5 602	.01064	74.48	13 06	13 29	14 15	1085.8	25.70
60	0.52142	0 2561	0 0008278	5 795	0 01103	77 21	13.09	13.32	14.39	1086.2	26.37
61	.54035	2654	0008562	5 993	01144	80 08	13 11	13 35	14 63	1086 7	27.06
62	.55970	.2749	0008852	6.196	01186	83 02	13 14	13.39	14 87	1087.1	27 76
63	.57985	.2848	0009153	6 407	.01229	86 03	13 16	13 42	15 11	1087.6	28 48
64	.60042	.2949	.0009460	6 622	.01274	89 18	13 19	13.46	15.35	1088.0	29.21
65	0 62179	0 3054	0 0009778	6 845	0.01320	92 40	13 21	13.49	15 59	1088 5	29.96
66	.64378	3162	.0010105	7 074	.01368	95.76	13 24	13 53	15 83	1088.9	30 73
67	.66638	.3273	.0010440	7 308	.01417	99.19	13 26	13 57	16.07	1089 4	31.51
68	.68980	.3388	.0010816	7.571	.01468	102 8	13 29	13 60	16 31	1089.8	32 31
69	.71382	.3506	.0011140	7.798	.01520	106 4	13.31	13.64	16.55	1090.3	33.12

[a]Compiled by W. M Sawdon, vapor pressures converted from *International Critical Tables*.

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp, F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In. Hg.		Enthalpy per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft Pounds	per Cu Ft Grains	per lb of Dry Air Pounds	per lb of Dry Air Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
70	0.73866	0 3628	0.0011507	8.055	0.01574	110 2	13 34	13 68	16 79	1090.7	33 96
71	.76431	.3754	.0011884	8 319	.01631	114.2	13 37	13 71	17 03	1091.2	34.83
72	.79058	.3883	.0012269	8.588	.01688	118.2	13.40	13.75	17 27	1091.6	35.70
73	.81766	.4016	.0012667	8 867	.01748	122 4	13.42	13 79	17 51	1092.1	36 60
74	.84555	.4153	.0013075	9 153	.01809	126.6	13.44	13 83	17.75	1092.5	37.51
75	0.87448	0.4295	0.0013497	9 448	0.01873	131 1	13.47	13 87	17 99	1093.0	38 46
76	.90398	.4440	.0013927	9.749	01938	135.7	13 49	13 91	18 23	1093.4	39.42
77	.93452	.4590	.0014371	10 06	02005	140 4	13 52	13 95	18.47	1093.9	40.40
78	.96588	.4744	.0014825	10 38	.02075	145 3	13.54	13 99	18 71	1094.3	41.42
79	.99825	.4903	.0015295	10.71	.02147	150 3	13.57	14 03	18.95	1094.8	42.46
80	1 0316	0.5067	0.0015777	11.04	0 02221	155.5	13.59	14 08	19 19	1095.2	43 51
81	1.0661	.5236	.0016273	11.39	.02298	160 9	13 62	14 12	19.43	1095.7	44 61
82	1 1013	.5409	.0016781	11 75	.02377	166 4	13 64	14.16	19 67	1096.1	45.72
83	1.1377	.5588	.0017304	12 11	02459	172 1	13 67	14 21	19 91	1096.6	46 88
84	1.1752	.5772	.0017841	12 49	.02543	178 0	13.69	14 26	20.15	1097.0	48.05
85	1 2135	0 5960	0 0018389	12 87	0 02629	184 0	13 72	14.30	20 39	1097.5	49.24
86	1.2527	.6153	.0018950	13 27	.02718	190.3	13 74	14.34	20 63	1097 9	50 47
87	1.2933	.6352	0019531	13 67	02810	196.7	13 77	14 39	20 87	1098 4	51 74
88	1 3346	.6555	.0020116	14.08	02904	203 3	13 79	14 44	21 11	1098 8	53.02
89	1 3774	.6765	.0020725	14.51	.03002	210 1	13 82	14.48	21 35	1099 3	54.35
90	1.4211	0.6980	0 0021344	14 94	0 03102	217 1	13 84	14 53	21.59	1099.7	55.70
91	1.4661	.7201	0021982	15 39	.03205	224 4	13 87	14 58	21 83	1100 2	57.09
92	1.5125	.7429	0022634	15 84	.03312	231 8	13.89	14 63	22 07	1100 6	58.52
93	1.5600	.7662	.0023304	16 31	.03421	239 5	13.92	14.69	22.32	1101.1	59.99
94	1.6088	.7902	.0023992	16 79	.03535	247 5	13 94	14.73	22.56	1101.5	61.50
95	1 6591	0 8149	0 0024697	17 28	0 03652	255 6	13 97	14 79	22 80	1102 0	63.05
96	1 7108	8403	0025425	17 80	.03772	264 0	13 99	14 84	23 04	1102 4	64.62
97	1 7638	8663	.0026164	18 31	.03896	272 7	14 02	14 90	23 28	1102.9	66.25
98	1 8181	.8930	.0026925	18 85	04024	281 7	14 04	14 95	23 52	1103.3	67.92
99	1 8741	.9205	.0027700	19 39	.04156	290 9	14 07	15 01	23.76	1103.8	69.63
100	1 9316	0 9487	0.0028506	19.95	0.04293	300 5	14 10	15.07	24.00	1104.2	71.40
101	1 9904	9776	.0029316	20 52	.04433	310.3	14.12	15.12	24.24	1104.7	73.21
102	2 0507	1 0072	.0030156	21.11	.04577	320.4	14 15	15 18	24.48	1105 1	75.06
103	2 1128	1 0377	.0031017	21.71	.04726	330 8	14.17	15.25	24 72	1105 6	76.97
104	2.1763	1 0689	.0031887	22.32	.04879	341.5	14.20	15.31	24.96	1106.0	78.92

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp, F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In Hg		Enthalpy per Lb		
	In of Hg	Lb per Sq In	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds	Grains	Pounds	Grains					
105	2.2414	1 1009	0 0032786	22 95	0 05037	352 6	14 22	15.37	25 20	1106 5	80 93
106	2 3084	1.1338	0033715	23 60	.05200	364 0	14 25	15 44	25 44	1106 9	83 00
107	2 3770	1.1675	.0034650	24 26	.05368	375 8	14 27	15 50	25 68	1107 4	85 13
108	2.4473	1.2020	.0035612	24.93	.05541	387 9	14 30	15 57	25 92	1107 8	87.30
109	2.5196	1.2375	.0036603	25.62	.05719	400 3	14 32	15 64	26.16	1108.3	89.54
110	2.5939	1.274	0.0037622	26.34	0 05904	413 3	14 35	15 71	26.40	1108 7	91.86
111	2 6692	1.311	.0038669	27 07	.06092	426 4	14 37	15 78	26.64	1109 2	94 21
112	2 7486	1 350	.0039729	27 81	.06292	440 4	14 39	15 85	26 88	1109 6	96.70
113	2.8280	1 389	.0040816	28 57	06493	454 5	14 42	15 93	27 12	1110 1	99 20
114	2.9094	1.429	.0041911	29.34	.06700	469 0	14 45	16 00	27.36	1110 5	101 76
115	2 9929	1.470	0 0043047	30 13	0 06913	483 9	14 47	16 08	27 60	1111 0	104.40
116	3.0784	1.512	.0044208	30 95	.07134	499 4	14 50	16 16	27.84	1111.4	107.13
117	3.1660	1 555	.0045372	31 76	.07361	515 3	14 52	16 24	28 08	1111.9	109.92
118	3 2576	1.600	.0046620	32 63	.07600	532.0	14 55	16 32	28 32	1112 3	112 85
119	3.3492	1.645	.0047846	33 49	07840	548 8	14.57	16.41	28.56	1112 8	115.80
120	3 4449	1 692	0.0049115	34.38	0 08093	566 5	14 60	16.50	28 80	1113 2	118 89
121	3.5406	1.739	.005040	35.28	08348	584.4	14 62	16.58	29.04	1113.7	122.01
122	3.6404	1.788	.005173	36.21	.08616	603.1	14 65	16 68	29 28	1114.1	125.27
123	3 7422	1.838	.005311	37 18	.08892	622.4	14 67	16.77	29.52	1114 6	128.63
124	3.8460	1 889	.005450	38 15	.09175	642.3	14 70	16 87	29.76	1115.0	132.06
125	3.9519	1.941	0.005590	39.13	0.09466	662 6	14 72	16 96	30.00	1115.5	135.59
126	4.0618	1 995	.005734	40 14	09770	683 9	14.75	17 06	30 24	1115.9	139.26
127	4.1718	2.049	.005882	41.17	.1008	705.6	14 77	17 17	30.48	1116.4	143.01
128	4.2858	2 105	.006031	42.22	.1040	728 0	14 80	17.27	30.72	1116 8	146.87
129	4.4039	2.163	.006188	43.32	.1074	751 8	14 83	17.38	30 96	1117 3	150.96
130	4.5220	2.221	0.006344	44.41	0.1107	774 9	14 85	17 49	31.20	1117.7	154.93
131	4.6441	2.281	.006504	45 53	.1143	800 1	14 88	17 61	31.45	1118 2	159.26
132	4.7703	2.343	.006671	46 70	.1180	826 0	14 90	17 73	31.69	1118 6	163 68
133	4.8986	2 406	.006839	47.87	.1218	852 6	14.93	17 85	31.93	1119 1	168.24
134	5.0289	2.470	.007010	49 07	.1257	879 9	14 95	17.97	32.17	1119.5	172.89
135	5.1633	2.536	0.007185	50.30	0.1297	907 9	14 98	18 10	32.41	1120.0	177.67
136	5.2997	2.603	.007364	51 55	.1339	937 3	15 00	18 23	32 65	1120 4	182 67
137	5.4402	2.672	.007547	52 83	.1382	967 4	15 03	18 36	32 89	1120 9	187 80
138	5.5827	2.742	.007732	54 12	.1427	998 9	15 05	18 50	33 13	1121.3	193.14
139	5.7293	2 814	.007923	55 46	.1473	1031 1	15.08	18 65	33 37	1121 8	198.61

[a]Compiled by W. M Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp, F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In Hg		Enthalpy per Lb		
			per Cu Ft		per lb of Dry Air						
	In of Hg.	Lb per Sq In	Pounds	Grains	Pounds	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
140	5.8779	2.887	0.008116	56.81	0.1521	1064.7	15.10	18.79	33.61	1122.2	204.30
141	6.0306	2.962	.008313	58.19	.1570	1099.0	15.13	18.94	33.85	1122.7	210.11
142	6.1874	3.039	.008516	59.61	.1622	1135.4	15.15	19.10	34.09	1123.1	216.26
143	6.3482	3.118	.008724	61.07	.1675	1172.5	15.18	19.26	34.33	1123.6	222.53
144	6.5111	3.198	.008933	62.53	.1730	1211.0	15.20	19.43	34.57	1124.0	229.02
145	6.6781	3.280	0.009148	64.04	0.1787	1250.9	15.23	19.60	34.81	1124.5	235.76
146	6.8471	3.363	.009366	65.56	.1846	1292.2	15.25	19.78	35.05	1124.9	242.71
147	7.0222	3.449	.009590	67.13	.1908	1335.6	15.28	19.96	35.29	1125.4	250.02
148	7.1993	3.536	.009817	68.72	.1971	1379.7	15.30	20.15	35.53	1125.8	257.43
149	7.3805	3.625	.010040	70.28	.2037	1425.9	15.33	20.35	35.77	1126.3	265.20
150	7.5658	3.716	0.010284	71.99	0.2105	1473.5	15.35	20.55	36.02	1126.7	273.19
151	7.7551	3.809	.010526	73.68	.2176	1523.2	15.38	20.76	36.26	1127.2	281.54
152	7.9485	3.904	.010772	75.40	.2250	1575.0	15.40	20.97	36.50	1127.6	290.21
153	8.1460	4.001	.011022	77.15	.2327	1628.9	15.43	21.20	36.74	1128.1	299.25
154	8.3476	4.100	.011279	78.95	.2407	1684.9	15.45	21.43	36.98	1128.5	308.61
155	8.5532	4.201	0.011539	80.77	0.2490	1743.0	15.48	21.67	37.22	1129.0	318.34
156	8.7650	4.305	.011807	82.65	.2577	1803.9	15.50	21.93	37.46	1129.4	328.51
157	8.9788	4.410	.012077	84.54	.2667	1866.9	15.53	22.19	37.70	1129.9	339.04
158	9.1986	4.518	.012354	86.48	.2761	1932.7	15.56	22.46	37.94	1130.3	350.02
159	9.4206	4.627	.012634	88.44	.2858	2000.6	15.58	22.74	38.18	1130.8	361.36
160	9.6486	4.739	0.012919	90.43	0.2961	2072.7	15.61	23.03	38.43	1131.2	373.38
161	9.8807	4.853	.013211	92.48	.3067	2146.9	15.63	23.33	38.67	1131.7	385.76
162	10.119	4.970	.013509	94.56	.3179	2225.3	15.66	23.65	38.91	1132.1	398.80
163	10.361	5.089	.013812	96.68	.3295	2306.5	15.68	23.98	39.15	1132.5	412.34
164	10.608	5.210	.014120	98.84	.3416	2391.2	15.71	24.33	39.39	1133.0	426.42
165	10.860	5.334	0.014434	101.0	0.3544	2480.8	15.73	24.69	39.63	1133.5	441.34
166	11.117	5.460	.014753	103.3	.3677	2573.9	15.76	25.07	39.87	1133.9	456.81
167	11.379	5.589	.015080	105.6	.3817	2671.9	15.78	25.46	40.11	1134.4	473.11
168	11.646	5.720	.015410	107.9	.3964	2774.8	15.81	25.88	40.35	1134.8	490.18
169	11.919	5.854	.015750	110.3	.4118	2882.6	15.83	26.31	40.59	1135.3	508.11
170	12.196	5.990	0.016092	112.6	0.4280	2996.0	15.86	26.77	40.83	1135.7	526.91
171	12.480	6.130	.016444	115.1	.4451	3115.7	15.88	27.24	41.07	1136.2	546.79
172	12.770	6.272	.016801	117.6	.4631	3241.7	15.91	27.74	41.32	1136.6	567.68
173	13.065	6.417	.017164	120.1	.4821	3374.7	15.93	28.28	41.56	1137.1	589.76
174	13.366	6.565	.017534	122.7	.5022	3515.4	15.96	28.84	41.80	1137.5	613.05

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONCLUDED)

Temp, F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In Hg		Enthalpy per Lb		
			per Cu Ft		per lb of Dry Air						
	In of Hg	Lb per Sq In	Pounds	Grains	Pounds	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
175	13.671	6.716	0.017914	125.4	0.5235	3664.5	15.98	29.43	42.04	1138.0	637.78
176	13.985	6.869	.018294	128.1	5459	3821.3	16.01	30.05	42.28	1138.4	663.73
177	14.303	7.025	018684	130.8	5697	3987.9	16.03	30.71	42.52	1138.9	691.35
178	14.627	7.184	019080	133.6	.5949	4164.3	16.06	31.41	42.76	1139.3	720.53
179	14.954	7.345	019477	136.3	6215	4350.5	16.08	32.15	43.00	1139.8	751.39
180	15.290	7.510	0.019888	139.2	0.6501	4550.7	16.11	32.94	43.24	1140.2	784.48
181	15.632	7.678	020304	142.1	.6805	4763.5	16.13	33.78	43.49	1140.7	819.74
182	15.981	7.849	.020729	145.1	.7131	4991.7	16.16	34.68	43.73	1141.1	857.45
183	16.337	8.024	021159	148.1	.7481	5236.7	16.18	35.65	43.97	1141.6	898.00
184	16.697	8.201	021598	151.2	.7854	5497.8	16.21	36.67	44.21	1142.0	941.14
185	17.066	8.382	0.022045	154.3	0.8258	5780.6	16.23	37.78	44.45	1142.5	987.93
186	17.440	8.566	022497	157.5	8693	6085.1	16.26	38.98	44.69	1142.9	1038.21
187	17.821	8.753	022956	160.7	.9162	6413.4	16.28	40.27	44.93	1143.4	1092.51
188	18.210	8.944	023424	164.0	.9673	6771.1	16.31	41.67	45.18	1143.8	1151.58
189	18.605	9.138	023900	167.3	1.0227	7158.9	16.34	43.04	45.42	1144.3	1216.04
190	19.008	9.336	0.024384	170.7	1.083	7581.0	16.36	44.85	45.66	1144.7	1285.37
191	19.419	9.538	024881	174.2	1.150	8050.0	16.39	46.68	45.90	1145.2	1362.88
192	19.839	9.744	025380	177.7	1.224	8568.0	16.41	48.70	46.14	1145.6	1448.35
193	20.266	9.954	025893	181.3	1.306	9142.0	16.44	50.93	46.38	1146.1	1543.19
194	20.702	10.168	.026413	184.9	1.397	9779.0	16.46	53.42	46.62	1146.5	1648.28
195	21.144	10.385	0.026939	188.6	1.499	10493.0	16.49	56.20	46.86	1147.0	1766.21
196	21.592	10.605	027472	192.3	1.613	11291.0	16.51	59.31	47.10	1147.4	1897.86
197	22.048	10.829	028019	196.1	1.742	12194.0	16.54	62.85	47.34	1147.9	2046.98
198	22.512	11.057	.028571	200.0	1.890	13230.0	16.56	66.88	47.59	1148.3	2217.88
199	22. 84	11.289	.029129	203.9	2.061	14427.0	16.59	71.54	47.83	1148.8	2415.51
200	3.465	11.525	0.029700	207.9	2.261	15827.0	16.61	76.99	48.07	1149.2	2646.41

[a]Compiled by W. M. Sawdon vapor pressures converted from *International Critical Tables*

against heat transfer to or from its surroundings, and which contains an exposed water surface. Alternatively, let the air pass through an insulated *air washer* whose spray water is recirculated continuously without being heated or cooled externally. In either case, when the apparatus has reached equilibrium temperatures throughout, the *water* will have attained a temperature (the temperature of adiabatic saturation) closely approximating the initial wet-bulb temperature of the air, and the *air* will have become saturated at the temperature of the water (or will have approached saturation at that temperature as a limit, the degree of saturation depending on the time and efficiency of contact of air and water).

Example 4. If air with a dry-bulb of 85 F and a wet-bulb of 70 F be saturated adiabatically by spraying with recirculated water, what will be the final temperature and the vapor content of the air?

Solution. The final temperature will be equal to the initial wet-bulb temperature or 70 F, and since the air is saturated at this temperature, from Table 6, W = 0 01574 lb per pound of dry air.

The energy for evaporating moisture into the air comes only from the air and its initially superheated vapor, which led Carrier to formulate the following energy equation for adiabatic saturation:

$$h'_{fg}\,(W_{t'} - W) = c_{p_a}\,(t - t') + c_{p_s}W\,(t - t') \tag{9}$$

and using $c_{p_a} = 0.24$ and $c_{p_s} = 0.45$

$$h'_{fg}\,(W_{t'} - W) = (0.24 + 0.45W)\,(t - t') \tag{9}$$

where

h'_{fg} = latent heat of vaporization at t', Btu per pound.

$(W_{t'} - W)$ = increase in vapor associated with 1 lb of dry air when it is saturated adiabatically from an initial dry-bulb temperature, t, and an initial vapor content, W, pounds.

Knowing any two of the three primary variables, t, t', or W, the third may be found from this equation for any process of adiabatic saturation.

TOTAL HEAT AND ENTHALPY

The *total heat* of a mixture of dry air and water vapor was originally defined by W. H. Carrier as:

$$\Sigma = c_{p_a}\,(t - 0) + W\,[h'_{fg} + c_{p_s}\,(t - t')] \tag{10}$$

where

Σ = total heat of the mixture, Btu per pound of dry air.
c_{p_a} = mean specific heat at constant pressure of dry air.
c_{p_s} = mean specific heat at constant pressure of water vapor.
t = dry-bulb temperature, degrees Fahrenheit.
t' = wet-bulb temperature, degrees Fahrenheit.
W = weight of water vapor mixed with each pound of dry air, pounds.
h'_{fg} = latent heat of vaporization at t', Btu per pound.

Since this definition holds for any mixture of dry air and water vapor,

the *total heat* of a mixture with a relative humidity of 100 per cent and at a temperature equal to the wet-bulb temperature (t') is:

$$\Sigma' = c_{p_a}\,(t' - 0) + W_{t'}\,h'_{fg} \tag{11}$$

By equating Equation 10 to Equation 11, the equation for the adiabatic saturation process, Equation 9a, follows. This demonstrates that the adiabatic saturation process at approximately constant wet-bulb temperature is also approximately a process of constant *total heat*. In short, the *total heat* of a mixture of dry air and water vapor is the same for any two states of the mixture at the same wet-bulb temperature. This fact furnishes a convenient means of finding the *total heat* of an air-vapor mixture in any state.

Enthalpy

This *total heat* of an air-vapor mixture is not equal to the enthalpy of the mixture, since the enthalpy of the liquid is not included in Equation 10. With the meaning of enthalpy in agreement with present practice in other branches of thermodynamics, the true enthalpy of a mixture of dry air and water vapor (with 0 F as the datum for dry air, and the saturated liquid at 32 F as the datum for the water vapor) is:

$$h = c_{p_a}\,(t - 0) + W\,h_s = 0.24\,(t - 0) + W\,h_s \tag{12}$$

where

h = the enthalpy of the mixture, Btu per pound of dry air.
t = the dry-bulb temperature, degrees Fahrenheit.
W = the weight of vapor per pound of dry air, pounds.
h_s = the enthalpy of the vapor in the mixture, Btu per pound.

The enthalpy of the water vapor in the mixture may be found in steam charts or tables when the dry-bulb temperature and the partial pressure of the vapor are known. Or, since the enthalpy of steam at low partial pressures, whether superheated or saturated, depends only upon temperature, the following empirical equation may be used:

$$h_s = 1059.2 + 0.45\,t \tag{13}$$

Substituting this value of h_s in Equation 12, the enthalpy of the mixture is:

$$h = 0.24\,(t - 0) + W\,(1059.2 + 0.45\,t) \tag{14}$$

Example 5. Find the enthalpy of an air-vapor mixture having a dry-bulb temperature of 85 F and a wet-bulb temperature of 70 F and a barometric pressure of 29.0 in Hg.

Solution. From Equation 2b and Table 6,

$$e = 0.7387 - \frac{(29.0 - 0.7387)\,(85 - 70)}{2800 - (1.3 \times 70)} = 0.05822$$

From Equation 5a,

$$W = 0.622\,\frac{0.5822}{29.0 - 0.5822} = 0.01274.$$

From Equation 14,

$$h = (0.24 \times 85) + [0.01274\,(1059.2 + 0.45 \times 85)] = 34.38 \text{ Btu per pound dry air.}$$

Since the enthalpy is nearly constant along a wet-bulb temperature line in any air-water vapor mixture, it may be found, approximately, when the wet-bulb temperature is known by using the temperature in Table 6 as wet-bulb temperatures and reading the corresponding enthalpy from the last column, provided the barometric pressure is 29.92 in Hg.

ENERGY EQUATION

An energy equation can be written that applies, in general, to various air conditioning processes, and this equation can be used to determine the quantity of heat transferred during such processes. In the most general form, this equation may be explained with the aid of Fig. 1 as follows:

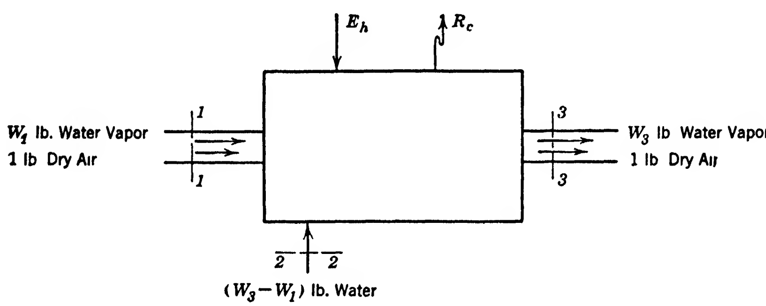


FIG. 1. DIAGRAM ILLUSTRATING ENERGY EQUATION 15

The rectangle may represent any apparatus, *e.g.*, a drier, humidifier, dehumidifier, cooling tower, or the like, by proper choice of the direction of the arrows.

In general, a mixture of air and water vapor, such as atmospheric air, enters the apparatus at 1 and leaves at 3. Water is supplied at some temperature, t_2. For the flow of 1 lb of dry air (with accompanying vapor) through the apparatus, provided there is no appreciable change in the elevation or velocity of the fluids and no mechanical energy delivered to or by the apparatus,

$$h_1 + E_h + (W_3 - W_1)\, h_2 = h_3 + R_c$$

or

$$E_h - R_c = h_3 - h_1 - (W_3 - W_1)\, h_2 \qquad (15)$$

where

E_h = the quantity of heat supplied per pound of dry air, Btu

R_c = the quantity of heat lost externally by heat transfer from the apparatus, Btu per pound of dry air.

W_1 = the weight of water vapor entering, per pound of dry air.

W_3 = the weight of water vapor leaving, per pound of dry air.

h_2 = the enthalpy of the water supplied at t_2, Btu per pound.

$h_3 - h_1$ = the increase in the enthalpy of the air-water vapor mixture in passing through the apparatus, Btu per pound of dry air

$$= 0.24\,(t_3 - t_1) + W_3\,(1059.2 + 0.45\,t_3) - W_1\,(1059.2 + 0.45t_1)$$

The net quantity of heat added to or removed from air-water vapor mixtures in air conditioning work is frequently *approximated* by taking the differences in total heat at exit and entrance.

For example, in Fig. 1, an *approximate* result is:

$$E_h - R_c = \Sigma_3 - \Sigma_1 \tag{16}$$

From the definitions of *total heat* and *enthalpy*, it may be demonstrated that Equation 16 is exactly equivalent to Equation 15, when, and only when, $t'_3 = t'_1 = t_2$; *i.e.*, when the initial and final wet-bulb temperatures and the temperature of the water supplied are equal. The one process that meets these conditions is adiabatic saturation, and either equation will give a result of zero; for other conditions, Equation 16 is approximate but satisfactory for many calculations.

The following problems illustrate the application of these principles:

Example 6. Heating (data from Example 2). Assuming the water to be supplied at 50 F, the net quantity of heat supplied is, from Equation 15,

From Equation 15,

$E_h - R_c = h_3 - h_1 - (W_3 - W_1)(50 - 32)$

$h_3 = (0.24 \times 70) + [0.00618\ (1059.2 + 0.45 \times 70)] = 23.54$ Btu per pound leaving dry air

$h_1 = (0.24 \times 0) + [0.000548\ (1059.2 + 0.45 \times 0)] = 0.58$ Btu per pound entering dry air

$E_h - R_c = 23.54 - 0.58 - [0.005632\ (50 - 32)] = 22.86$ Btu per pound dry air, net heat supplied

Example 7. Cooling (data from Example 3). If the condensate is removed at 54 F the quantity of heat removed is found from Equation 15, by proper regard to the arrow direction in Fig. 1,

From Equation 15,

$E_h + R_c = h_1 - h_3 - (W_1 - W_3)(54 - 32)$

$h_1 = (0.24 \times 84) + [0.01248\ (1059.2 + 0.45 \times 84)] = 33.85$ Btu per pound entering dry air

$h_3 = (0.24 \times 54) + [0.00887\ (1059.2 + 0.45 \times 54)] = 22.57$ Btu per pound leaving dry air

$E_h + R_c = 33.85 - 22.57 - [(0.00361\ (54 - 32)] = 11.20$ Btu per pound dry air, net heat removed

Using Table 6, the initial enthalpy of the air-vapor mixture, since the wet-bulb temperature is 70 F, is 33.96 Btu per pound of dry air.

The final enthalpy is, from Table 6, since the exit air is saturated, 22.55 Btu per pound. Hence, using Equation 16, the quantity of heat removed is, approximately, (33.96 − 22.55) or 11.41 Btu per pound of dry air. The degree of approximation to the correct result is evident in this example

PSYCHROMETRIC CHART

Many types of charts which give graphical solutions of the psychrometric equations and other useful data have been devised. One of these, the revised Bulkeley Psychrometric Chart[6], will be found attached to the inside back cover. It shows graphically the relationship expressed in Equations 9a and 9b. It also gives the grains of moisture per pound of dry air for saturation, the grains of moisture per cubic foot of saturated air, the *total heat* in Btu per pound of dry air saturated with moisture, and the weight of the dry air in pounds per cubic foot. Fig. 2 shows the procedure to follow in using the Bulkeley Chart. The directrix curves above the saturation line are as follows:

[6]The Bulkeley Psychrometric Chart was presented to the Society in 1926 (See A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 163). Single copy of the chart can be furnished at a cost of $.25

A is the *total heat* in Btu contained in the mixture above 0 F, and is to be referred to the column of figures at the left side of the chart. Enthalpy of the liquid is not included.

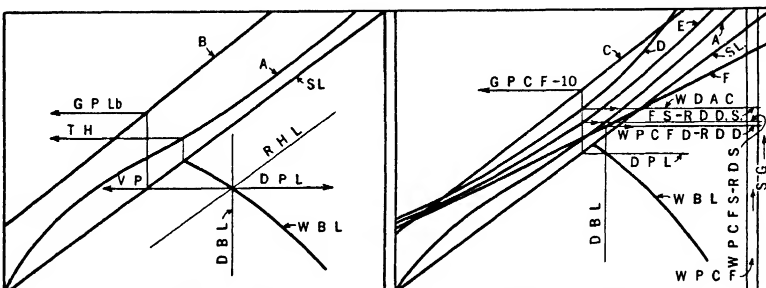
B is the grains of moisture or water vapor accompanying each pound of dry air and is to be referred to the figures at the left side of the chart.

C is the grains of moisture or water vapor per cubic foot of saturated mixture, and is to be referred to the figures at the left side of the chart which are to be divided by 10.

D is the weight in decimal fractions of a pound of dry air in one cubic foot of the saturated mixture, and is referred to the first column of figures to the right of the saturation line between the vertical dry-bulb temperature lines 170 and 180 F. The relative density of the mixture is read in a similar manner from the same curve by the column of figures between the vertical dry-bulb temperature lines 180 and 190 F.

E is similar to *D* but is for one cubic foot of the saturated mixture.

F is similar to *D* but is for dry air, devoid of all moisture or water vapor. For convenience, the approximate absolute temperature of 500 F is given at 40 F on the saturation line for the purpose of calculating volume, weight per cubic foot, and relative density at partial saturation.



A, B, C, D, E, F=Directrix lines
D B L=Dry bulb line
D P L=Dew point line

R H L=Relative humidity line
W B L=Wet bulb line
S L=Saturation line

G P Lb=Grains moisture sat mixture per lb dry air
T H=Total heat sat mixture per lb dry air
V P=Vapor pressure in mm mercury
G P C F S=Grains moisture per cu ft sat air
W D A C F S=Weight dry air per cu ft in lbs saturated

R D D S=Relative density dry air per cu ft saturated
W P C F S=Weight per cu ft in lbs saturated
R D S=Relative density per cu ft saturated
W P C F D=Weight per cu ft in lbs dry
R D D=Relative density per cu ft dry

$$\text{G P C F} \times \frac{\text{Abs temp at D P}}{\text{Abs temp at D B}} = \text{G P C F at partial saturation}$$

$$\text{W P C F} \times \frac{\text{Abs temp at D P}}{\text{Abs temp at D B}} = \text{W P C F at partial saturation}$$

$$\text{R D} \times \frac{\text{Abs temp at D P}}{\text{Abs temp at D B}} = \text{R D at partial saturation}$$

FIG. 2 DIAGRAMS SHOWING PROCEDURE TO FOLLOW IN USING BULKELEY CHART

METHOD OF USING THE CHART

Example 8. Relative Humidity: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, the relative humidity is read directly on the straight diagonal lines as 46 per cent.

Example 9. Dew-Point: At the intersection of the 78 F wet-bulb line, the dew-point temperature is read directly on the horizontal temperature lines as 70 9 F.

Example 10. Vapor Pressure: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, pass in a horizontal direction to the left of the chart and on the logarithmic scale read the vapor pressure as 19.4 millimeters of mercury. (Divide by 25.4 for inches).

Example 11. Total Heat Above 0 F in Mixture per Pound of Dry Air Saturated with Moisture: From where the wet-bulb line joins the saturation line, pass in a vertical

direction on the 78 F dry-bulb line to its intersection with curve A and on the logarithmic scale at the left of the chart read 40.6 Btu per pound of mixture. The use of this curve to obtain the total heat in the mixture at any wet-bulb temperature is a great convenience, as the number of Btu required to heat the mixture and humidify it, as well as the refrigeration required to cool and dehumidify the mixture, can be obtained by taking the difference in total heat before and after treatment of the mixture.

Example 12. Grains of Moisture per Pound of Dry Air Saturated with Moisture: From 70.9 F dew-point temperature on the saturation line, pass vertically to the intersection with curve B and on the logarithmic scale at the left read 114 grains of moisture per pound.

Example 13. Grains of Moisture per Cubic Foot of Mixture, Partially Saturated: From 70.9 F dew-point temperature on the saturation line proceed in a vertical direction to curve C, and on the logarithmic scale to the left read 83.3 which, divided by 10, gives 8.33 grains. A temperature of 70.9 F is equal to an absolute temperature of 530.9, and 95 F equals 555, absolute temperature. Therefore, $\frac{530.9}{555} \times 8.33 = 7.97$ grains per cubic foot of partially saturated mixture

Example 14. Grains of Moisture per Cubic Foot of Saturated Air: Starting at the saturation line at the desired temperature, pass in a vertical direction to curve C and on the logarithmic scale at the left, read a number which, divided by 10, will give the answer.

Example 15. Weight per Cubic Foot of Dry Air and Relative Density: From the point where, for example, the 70 F vertical dry-bulb line intersects curve E, pass to right side and read 0 075 lb, if cubic feet per pound are desired, divide 1 by this amount. The relative density is read immediately to the right as 1.00.

Example 16. Weight of Dry Air per Cubic Foot of Saturated Mixture and Relative Density: From the point where, for example, the 70 F vertical line intersects the curve D, pass to the right and read weight per cubic foot as 0.07316 with a relative density of 0.9755 for saturated air at 70 F.

Example 17. Weight of Dry Air per Cubic Foot and Relative Density of Partially Saturated Air: Air at 50 F and a wet-bulb temperature of 46 F is to be heated to 130 F. The wet- and dry-bulb lines intersect at a dew-point temperature of 42 F. Pass to the left where this dew-point line intersects the saturation line and then pass in a vertical direction to where the 42 F dry-bulb line intersects with curve D. Then pass directly to the right and read the weight per cubic foot of saturated air at 42 F as 0 07844 and the relative density as 1 046. The absolute temperature at 42 F is 502, and at 130 F is 590. Therefore, $\frac{502}{590} = 0\ 851$. The weight of 1 cu ft of air at 50 F dry-bulb and 46 F wet-bulb when heated to 130 F is $0\ 07844 \times 0.851 = 0.06675$, and the relative density is $1.046 \times 0\ 851 = 0.89$.

PROPERTIES OF WATER

Composition of Water. Water is a chemical compound (H_2O) formed by the union of two volumes of hydrogen and one volume of oxygen, or two parts by weight of hydrogen and 16 parts by weight of oxygen.

Density of Water. Water has its greatest density at 39.2 F, and it expands when heated or cooled from this temperature. At 62 F a U. S. gallon of 231 cu in. of water weighs approximately 8⅓ lb, and a cubic foot of water is equal to 7.48 gal. The specific volume of water depends on the temperature and it is always the reciprocal of its density. (See Table 7).

Water Pressures. Pressures are often stated in feet or inches of water column. At 62 F, with h equal to the head in feet, the pressure of a column of water is $62.383h$ lb per square foot, or $0.433h$ lb per square inch. A column of water 2.309 ft (27.71 in.) high exerts a pressure of one pound per square inch at 62 F.

Boiling Point of Water. The boiling point of water varies with the pressure; it is lower at higher altitudes. A change in pressure will always be accompanied by a change in the boiling point, and there will be a corresponding change in the latent heat of evaporation. These values are given in Table 8.

Specific Heat. The specific heat of water, or the amount of heat (Btu) required to raise the temperature of one pound of water one degree Fahrenheit, varies with the temperature, but it is commonly assumed to be

TABLE 7. THERMAL PROPERTIES OF WATER

Temperature Deg F	Sat Press Lb per Sq In	Volume Cu Ft per Lb	Weight Lb per Cu Ft	Specific Heat
32	0.0887	0.01602	62.42	1.0093
40	0.1217	0 01602	62.42	1.0048
50	0.1780	0 01602	62.42	1 0015
60	0.2561	0.01603	62.38	0 9995
70	0.3628	0 01605	62.31	0.9982
80	0.5067	0.01607	62.23	0.9975
90	0.6980	0 01610	62 11	0 9971
100	0.9487	0 01613	62 00	0.9970
110	1.274	0.01616	61.88	0 9971
120	1.692	0.01620	61.73	0.9974
130	2.221	0.01625	61.54	0 9978
140	2.887	0 01629	61 39	0 9984
150	3.716	0.01634	61 20	0 9990
160	4.739	0.01639	61.01	0 9998
170	5.990	0 01645	60 79	1 0007
180	7.510	0 01650	60 61	1 0017
190	9.336	0 01656	60 39	1 0028
200	11.525	0 01663	60 13	1 0039
210	14.123	0.01669	59.92	1 0052
212	14.696	0.01670	59.88	1.0055
220	17.188	0.01676	59.66	1 0068
240	24 97	0 01690	59.17	1 0104
260	35.43	0 01706	58 62	1 0148
280	49.20	0.01723	58.04	1 0200
300	67.01	0 01742	57 41	1 0260
350	134 62	0.01797	55 65	1.0440
400	247.25	0.01865	53.62	1 0670
450	422 61	0.01950	51.30	1 0950
500	681.09	0.02050	48 80	1.1300
550	1045.4	0.02190	45.70	1 2000
600	1544.6	0 02410	41 50	1 3620
700	3096.4	0.03940	25.40	....

unity at all temperatures. Steam tables are based on exact values, however. The specific heat of ice at 32 F is 0.492 Btu per pound. The amount of heat required to raise one pound of water at 32 F through a known temperature interval depends on the average specific heat for the temperature range.

Sensible and Latent Heat. The heat necessary to raise the temperature of one pound of water from 32 F to the boiling point is known as the *heat of the liquid* or *sensible heat.* When more heat is added, the water begins to evaporate and expand at constant temperature until the water is entirely changed into steam. The heat thus added is known as the *latent heat of evaporation.*

Table 8. Properties of Saturated Steam: Pressure Table[a]

Abs Press. In. Hg. p	Temp F t	Specific Volume Sat Liquid V_f	Specific Volume Sat Vapor V_g	Enthalpy Sat Liquid h_f	Enthalpy Evap h_{fg}	Enthalpy Sat Vapor h_g	Entropy Sat Liquid S_f	Entropy Evap S_{fg}	Entropy Sat. Vapor S_g	Abs. Press. In Hg. p
0.25	40 23	0 01602	2423 7	8 28	1071 1	1079 4	0 0166	2 1423	2 1589	**0.25**
0.50	58 80	0 01604	1256 4	26 86	1060 6	1087 5	0 0532	2 0453	2.0985	**0.50**
0.75	70 43	0 01606	856 1	38 47	1054 0	1092 5	0 0754	1 9881	2 0635	**0.75**
1.00	79 03	0 01608	652 3	47.05	1049 2	1096 3	0 0914	1 9473	2 0387	**1.00**
1.5	91 72	0 01611	444 9	59 71	1042 0	1101 7	0 1147	1.8894	2 0041	**1.5**
2	101 14	0 01614	339 2	69 10	1036 6	1105 7	0 1316	1 8481	1 9797	**2.0**
4	125 43	0 01622	176 7	93 34	1022 7	1116 0	0 1738	1 7476	1 9214	**4.0**
6	140 78	0 01630	120 72	108 67	1013 6	1122 3	0 1996	1 6881	1 8877	**6**
8	152 24	0 01635	92 16	120 13	1006 9	1127 0	0 2186	1 6454	1 8640	**8**
10	161 49	0 01640	74 76	129 38	1001.4	1130 8	0 2335	1 6121	1 8456	**10**
12	169 28	0 01644	63 03	137 18	996 7	1133 9	0 2460	1 5847	1 8307	**12**
14	176 05	0 01648	54 55	143 96	992 6	1136 6	0 2568	1 5613	1 8181	**14**
16	182 05	0 01652	48 14	149 98	988 9	1138 9	0 2662	1 5410	1 8072	**16**
18	187 45	0 01655	43 11	155 39	985 7	1141 1	0 2746	1 5231	1 7977	**18**
20	192 37	0 01658	39 07	160 33	982 7	1143 0	0 2822	1 5069	1 7891	**20**
22	196 90	0 01661	35 73	164 87	979 8	1144 7	0 2891	1 4923	1 7814	**22**
24	201 09	0 01664	32 94	169 09	977 2	1146 3	0 2955	1 4789	1 7744	**24**
26	205 00	0 01667	30 56	173 02	974 8	1147 8	0 3014	1 4665	1 7679	**26**
28	208 67	0 01669	28 52	176 72	972.5	1149 2	0 3069	1 4550	1 7619	**28**
30	212 13	0 01672	26 74	180 19	970 3	1150 5	0 3122	1 4442	1 7564	**30**
Lb/Sq In.										**Lb/Sq In**
14 696	212 00	0 01672	26 80	180 07	970 3	1150 4	0 3120	1 4446	1 7566	**14.696**
16	216 32	0 01674	24 75	184 42	967 6	1152 0	0 3184	1 4313	1 7497	**16**
18	222 41	0 01679	22 17	190 56	963 6	1154 2	0 3275	1 4128	1 7403	**18**
20	227 96	0 01683	20 089	196 16	960 1	1156 3	0 3356	1 3962	1 7319	**20**
22	233 07	0 01687	18 375	201 33	956 8	1158 1	0 3431	1 3811	1 7242	**22**
24	237 82	0 01691	16 938	206 14	953 7	1159 8	0 3500	1 3672	1 7172	**24**
26	242 25	0 01694	15 715	210 62	950 7	1161 3	0 3564	1 3544	1 7108	**26**
28	246 41	0 01698	14 663	214 83	947 9	1162.7	0 3623	1 3425	1 7048	**28**
30	250 33	0 01701	13 746	218 82	945 3	1164 1	0 3680	1 3313	1 6993	**30**
32	254 05	0 01704	12 940	222 59	942 8	1165 4	0 3733	1 3209	1 6941	**32**
34	257 58	0 01707	12 226	226 18	940 3	1166 5	0 3783	1 3110	1 6893	**34**
36	260 95	0 01709	11 588	229 60	938 0	1167 6	0 3831	1 3017	1 6848	**36**
38	264 16	0 01712	11 015	232 89	935 8	1168 7	0 3876	1 2929	1 6805	**38**
40	267 25	0 01715	10 498	236 03	933 7	1169 7	0 3919	1 2844	1 6763	**40**
42	270 21	0 01717	10 029	239 04	931 6	1170 7	0 3960	1 2764	1 6724	**42**
44	273 05	0 01720	9 601	241 95	929 6	1171 6	0 4000	1 2687	1 6687	**44**
46	275 80	0 01722	9 209	244 75	927 7	1172 4	0 4038	1 2613	1 6652	**46**
48	278 45	0 01725	8 848	247 47	925 8	1173 3	0 4075	1 2542	1 6617	**48**
50	281 01	0 01727	8 515	250 09	924 0	1174 1	0 4110	1 2474	1 6585	**50**
52	283 49	0 01729	8 208	252 63	922 2	1174 8	0 4144	1 2409	1 6553	**52**
54	285 90	0 01731	7 922	255 09	920 5	1175 6	0 4177	1 2346	1 6523	**54**
56	288 23	0 01733	7 656	257 50	918 8	1176 3	0 4209	1 2285	1 6494	**56**
58	290 50	0 01736	7 407	259 82	917 1	1176 9	0 4240	1 2226	1 6466	**58**
60	292 71	0 01738	7 175	262 09	915 5	1177 6	0 4270	1 2168	1 6438	**60**
62	294 85	0 01740	6 957	264 30	913 9	1178 2	0 4300	1 2112	1 6412	**62**
64	296 94	0 01742	6 752	266 45	912 3	1178 8	0 4328	1 2059	1 6387	**64**
66	298 99	0 01744	6 560	268 55	910 8	1179 4	0 4356	1 2006	1 6362	**66**
68	300 98	0 01746	6 378	270 60	909 4	1180 0	0 4383	1 1955	1 6338	**68**
70	302 92	0 01748	6 206	272 61	907 9	1180 6	0 4409	1 1906	1 6315	**70**
72	304 83	0 01750	6 044	274 57	906 5	1181 1	0 4435	1 1857	1 6292	**72**
74	306 68	0 01752	5 890	276 49	905 1	1181 6	0 4460	1 1810	1 6270	**74**
76	308 50	0 01754	5 743	278 37	903 7	1182 1	0 4484	1 1764	1 6248	**76**
78	310 29	0 01755	5 604	280.21	902 4	1182 6	0 4508	1 1720	1 6228	**78**
80	312 03	0 01757	5 472	282 02	901 1	1183 1	0 4531	1 1676	1 6207	**80**
82	313 74	0 01759	5 346	283 79	899 7	1183 5	0 4554	1 1633	1 6187	**82**
84	315 42	0 01761	5 226	285 53	898 5	1184 0	0 4576	1 1592	1 6168	**84**
86	317 07	0 01762	5 111	287 24	897 2	1184 4	0 4598	1 1551	1 6149	**86**
88	318 68	0 01764	5 001	288 91	895 9	1184 8	0 4620	1.1510	1 6130	**88**
90	320 27	0 01766	4 896	290 56	894 7	1185 3	0 4641	1 1471	1 6112	**90**
92	321 83	0 01768	4 796	292 18	893 5	1185 7	0 4661	1 1433	1 6094	**92**
94	323 36	0 01769	4 699	293 78	892 3	1186 1	0 4682	1 1394	1 6076	**94**
96	324 87	0 01771	4 606	295 34	891 1	1186 4	0 4702	1 1358	1 6060	**96**
98	326 35	0 01772	4.517	296 89	889 9	1186 8	0 4721	1 1322	1 6043	**98**

[a]Reprinted by permission from *Thermodynamic Properties of Steam*, by J H Keenan and F G. Keyes, published by John Wiley and Sons, Inc.

TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE[a] (Continued)

Abs. Press. Lb / Sq In. p	Temp F t	Specific Volume		Enthalpy			Entropy			Abs. Press. Lb / Sq In p
		Sat Liquid v_f	Sat. Vapor v_g	Sat Liquid h_f	Evap h_{fg}	Sat Vapor h_g	Sat Liquid s_f	Evap s_{fg}	Sat Vapor s_g	
100	327.81	0 01774	4 432	298 40	888 8	1187.2	0 4740	1 1286	1 6026	**100**
102	329 25	0 01775	4 350	299 90	887 6	1187 5	0 4759	1 1251	1 6010	**102**
104	330 66	0 01777	4 271	301.37	886 5	1187 9	0 4778	1 1216	1 5994	**104**
106	332 05	0 01778	4 194	302 82	885 4	1188 2	0 4796	1 1182	1 5978	**106**
108	333 42	0 01780	4 120	304.26	884 3	1188 6	0 4814	1.1149	1 5963	**108**
110	334 77	0 01782	4 049	305 66	883 2	1188 9	0 4832	1 1117	1.5948	**110**
112	336 11	0.01783	3 981	307.06	882 1	1189 2	0 4849	1 1085	1 5934	**112**
114	337 42	0 01784	3.914	308 43	881 1	1189 5	0 4866	1.1053	1 5919	**114**
116	338 72	0 01786	3 850	309 79	880 0	1189 8	0 4883	1 1022	1 5905	**116**
118	339 99	0.01787	3 788	311 12	879 0	1190 1	0.4900	1 0992	1.5891	**118**
120	341 25	0 01789	3 728	312 44	877 9	1190 4	0 4916	1 0962	1 5878	**120**
122	342 50	0 01791	3 670	313 75	876 9	1190 7	0 4932	1 0933	1 5865	**122**
124	343 72	0 01792	3 614	315 04	875 9	1190 9	0 4948	1 0903	1 5851	**124**
126	344 94	0 01793	3 560	316 31	874 9	1191 2	0 4964	1 0874	1 5838	**126**
128	346 13	0 01794	3 507	317 57	873 9	1191 5	0 4980	1 0845	1 5825	**128**
130	347.32	0 01796	3 455	318 81	872 9	1191 7	0 4995	1 0817	1 5812	**130**
132	348 48	0 01797	3 405	320 04	872 0	1192.0	0 5010	1 0790	1 5800	**132**
134	349 64	0 01799	3 357	321 25	871 0	1192 2	0 5025	1 0762	1 5787	**134**
136	350.78	0 01800	3 310	322 45	870 1	1192 5	0 5040	1 0735	1 5775	**136**
138	351 91	0 01801	3 264	323 64	869 1	1192.7	0 5054	1 0709	1 5763	**138**
140	353 02	0 01802	3 220	324 82	868 2	1193 0	0 5069	1 0682	1 5751	**140**
142	354 12	0 01804	3 177	325 98	867 2	1193 2	0 5083	1 0657	1 5740	**142**
144	355 21	0 01805	3 134	327 13	866 3	1193 4	0 5097	1 0631	1 5728	**144**
146	356 29	0 01806	3 094	328 27	865 3	1193 6	0 5111	1 0605	1 5716	**146**
148	357 36	0 01808	3.054	329 39	864 5	1193 9	0 5124	1 0580	1 5705	**148**
150	358 42	0 01809	3 015	330.51	863 6	1194 1	0 5138	0 0556	1 5694	**150**
152	359.46	0 01810	2 977	331 61	862 7	1194 3	0 5151	1 0532	1 5683	**152**
154	360 49	0 01812	2 940	332 70	861 8	1194 5	0 5165	1 0507	1 5672	**154**
156	361 52	0.01813	2 904	333 79	860 9	1194 7	0 5178	1 0483	1 5661	**156**
158	362 52	0 01814	2 869	334.86	860 0	1194 9	0 5191	1 0459	1 5650	**158**
160	363 53	0 01815	2 834	335 93	859 2	1195 1	0 5204	1 0436	1.5640	**160**
162	364 53	0 01817	2 801	336.98	858 3	1195 3	0 5216	1 0414	1 5630	**162**
164	365.51	0 01818	2 768	338 02	857 5	1195 5	0 5229	1 0391	1 5620	**164**
166	366 48	0 01819	2 736	339 05	856 6	1195 7	0 5241	1 0369	1 5610	**166**
168	367 45	0 01820	2 705	340 07	855 7	1195 8	0 5254	1 0346	1 5600	**168**
170	368 41	0.01822	2 675	341 09	854 9	1196.0	0 5266	1 0324	1 5590	**170**
172	369 35	0 01823	2 645	342 10	854 1	1196 2	0 5278	1 0302	1 5580	**172**
174	370 29	0 01824	2.616	343.10	853 3	1196 4	0 5290	1 0280	1 5570	**174**
176	371 22	0 01825	2 587	344 09	852 4	1196 5	0 5302	1 0259	1 5561	**176**
178	372 14	0.01826	2 559	345 06	851 6	1196 7	0 5313	1 0238	1 5551	**178**
180	373 06	0 01827	2 532	346 03	850 8	1196 9	0 5325	1 0217	1 5542	**180**
182	373 96	0 01829	2.505	317 00	850 0	1197 0	0 5336	1 0196	1 5532	**182**
184	374 86	0 01830	2 479	347 96	849 2	1197 2	0 5348	1 0175	1 5523	**184**
186	375 75	0 01831	2.454	348 92	818 4	1197 3	0 5359	1 0155	1 5514	**186**
188	376 64	0 01832	2 429	349 86	847 6	1197 5	0 5370	1 0136	1 5506	**188**
190	377 51	0 01833	2 404	350 79	846 8	1197 6	0 5381	1 0116	1 5497	**190**
192	378 38	0 01834	2 380	351 72	846 1	1197 8	0.5392	1 0096	1 5488	**192**
194	379 24	0.01835	2.356	352 64	845 3	1197 9	0 5403	1 0076	1 5479	**194**
196	380 10	0.01836	2 333	353 55	844 5	1198 1	0.5414	1.0056	1 5470	**196**
198	380 95	0 01838	2.310	354 46	843 7	1198 2	0.5425	1.0037	1 5462	**198**
200	381 79	0 01839	2 288	355 36	843 0	1198 4	0 5435	1 0018	1.5453	**200**
205	383 86	0 01842	2 234	357 58	841 1	1198 7	0 5461	0 9971	1 5432	**205**
210	385 90	0 01844	2 183	359.77	839 2	1199 0	0 5487	0 9925	1 5412	**210**
215	387.89	0.01847	2 134	361 91	837 4	1199 3	0 5512	0 9880	1 5392	**215**
220	389 86	0 01850	2.087	364 02	835.6	1199 6	0 5537	0 9835	1 5372	**220**
225	391 79	0 01852	2 0422	366 09	833 8	1199 9	0 5561	0 9792	1 5353	**225**
230	393.68	0 01854	1 9992	368 13	832.0	1200 1	0 5585	0.9750	1 5334	**230**
235	395 54	0 01857	1.9579	370 14	830 3	1200 4	0 5608	0 9708	1 5316	**235**
240	397 37	0 01860	1.9183	372 12	828 5	1200.6	0 5631	0 9667	1 5298	**240**
245	399 18	0.01863	1 8803	374 08	826 8	1200 9	0 5653	0 9627	1 5280	**245**

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TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE[a] (Concluded)

ABS PRESS LB / SQ IN p	TEMP F t	SPECIFIC VOLUME		ENTHALPY			ENTROPY			ABS. PRESS. LB / SQ IN. p
		Sat Liquid V_f	Sat Vapor V_g	Sat. Liquid h_f	Evap. h_{fg}	Sat Vapor h_g	Sat Liquid S_f	Evap. S_{fg}	Sat Vapor S_g	
250	400 95	0 01865	1 8438	376 00	825.1	1201 1	0 5675	0 9588	1.5263	250
260	404 42	0 01870	1 7748	379 76	821 8	1201.5	0 5719	0.9510	1 5229	260
270	407 78	0 01875	1 7107	383 42	818 5	1201 9	0 5760	0 9436	1.5196	270
280	411 05	0 01880	1 6511	386 98	815 3	1202 3	0 5801	0 9363	1 5164	280
290	414 23	0 01885	1 5954	390 46	812 1	1202 6	0 5841	0 9292	1.5133	290
300	417 33	0 01890	1 5433	393 84	809 0	1202 8	0 5879	0 9225	1 5104	300
320	423 29	0 01899	1 4485	400 39	803 0	1203 4	0 5952	0 9094	1 5046	320
340	428 97	0 01908	1 3645	406 66	797 1	1203 7	0 6022	0 8970	1.4992	340
360	434 40	0 01917	1 2895	412 67	791 4	1204 1	0 6090	0 8851	1 4941	360
380	439 60	0 01925	1 2222	418 45	785 8	1204 3	0 6153	0 8738	1.4891	380
400	444 59	0 0193	1 1613	424 0	780 5	1204 5	0 6214	0 8630	1 4844	400
420	449 39	0 0194	1 1061	429 4	775 2	1204 6	0 6272	0 8527	1 4799	420
440	454 02	0 0195	1 0556	434 6	770 0	1204 6	0 6329	0 8426	1 4755	440
460	458 50	0 0196	1 0094	439 7	764 9	1204 6	0 6383	0 8330	1 4713	460
480	462 82	0 0197	0 9670	444 6	759 9	1204 5	0 6436	0 8237	1.4673	480
500	467 01	0 0197	0 9278	449 4	755 0	1204 4	0 6487	0 8147	1 4634	500
520	471 07	0 0198	0 8915	454 1	750 1	1204 2	0 6536	0 8060	1 4596	520
540	475 01	0 0199	0 8578	458 6	745 4	1204 0	0 6584	0 7976	1 4560	540
560	478 85	0 0200	0 8265	463 0	740 8	1203 8	0 6631	0 7893	1.4524	560
580	482 58	0 0201	0 7973	467 4	736 1	1203 5	0 6676	0 7813	1 4489	580
600	486 21	0 0201	0 7698	471 6	731 6	1203.2	0 6720	0 7734	1 4454	600
620	489 75	0 0202	0 7440	475 7	727 2	1202 9	0 6763	0 7658	1 4421	620
640	493 21	0 0203	0 7198	479 8	722 7	1202.5	0 6805	0 7584	1 4389	640
660	496 58	0 0204	0 6971	483 8	718 3	1202 1	0 6846	0 7512	1 4358	660
680	499.88	0 0204	0 6757	487 7	714 0	1201 7	0 6886	0 7441	1 4327	680
700	503 10	0 0205	0 6554	491 5	709 7	1201 2	0 6925	0 7371	1 4296	700
720	506 25	0 0206	0 6362	495 3	705 4	1200 7	0 6963	0 7303	1 4266	720
740	509 34	0 0207	0.6180	499 0	701 2	1200 2	0 7001	0 7237	1 4237	740
760	512 36	0 0207	0 6007	502 6	697 1	1199 7	0 7037	0 7172	1 4209	760
780	515 33	0 0208	0 5843	506 2	692 9	1199 1	0 7073	0 7108	1 4181	780
800	518 23	0 0209	0.5687	509 7	688 9	1198 6	0 7108	0.7045	1 4153	800
820	521 08	0 0209	0 5538	513 2	684 8	1198 0	0 7143	0 6983	1 4126	820
840	523 88	0 0210	0 5396	516 6	680 8	1197 4	0 7177	0 6922	1 4099	840
860	526 63	0 0211	0 5260	520 0	676 8	1196 8	0.7210	0 6862	1 4072	860
880	529 33	0 0212	0 5130	523 3	672.8	1196 1	0 7243	0 6803	1 4046	880
900	531 98	0 0212	0 5006	526 6	668 8	1195 4	0 7275	0 6744	1 4020	900
920	534.59	0 0213	0 4886	529 8	664 9	1194 7	0 7307	0 6687	1 3995	920
940	537 16	0 0214	0 4772	533 0	661 0	1194 0	0 7339	0 6631	1 3970	940
960	539 68	0 0214	0 4663	536 2	657 1	1193 3	0 7370	0 6576	1 3945	960
980	542 17	0 0215	0 4557	539 3	653 3	1192 6	0 7400	0 6521	1 3921	980
1000	544 61	0 0216	0 4456	542 4	649 4	1191 8	0 7430	0 6467	1 3897	1000
1050	550 57	0 0218	0 4218	550 0	639 9	1189 9	0 7504	0 6334	1 3838	1050
1100	556 31	0 0220	0 4001	557 4	630 4	1187 8	0 7575	0 6205	1 3780	1100
1150	561 86	0 0221	0 3802	564 6	621 0	1185 6	0 7644	0 6079	1 3723	1150
1200	567 22	0 0223	0 3619	571 7	611 7	1183 4	0 7711	0 5956	1 3667	1200
1250	572 42	0 0225	0 3450	578 6	602 4	1181 0	0 7776	0 5836	1 3612	1250
1300	577 46	0 0227	0 3293	585 4	593 2	1178 6	0 7840	0 5719	1 3559	1300
1350	582 35	0 0229	0 3148	592 1	584 0	1176 1	0 7902	0 5604	1 3506	1350
1400	587 10	0 0231	0 3012	598 7	574 7	1173 4	0 7963	0 5491	1 3454	1400
1450	591 73	0 0233	0 2884	605 2	565 5	1170 7	0 8023	0 5379	1 3402	1450
1500	596 23	0 0235	0 2765	611 6	556 3	1167 9	0 8082	0 5269	1 3351	1500
1600	604 90	0 0239	0 2548	624 1	538 0	1162 1	0 8196	0 5053	1 3249	1600
1700	613 15	0 0243	0 2354	636 3	519 6	1155 9	0 8306	0 4843	1 3149	1700
1800	621 03	0 0247	0 2179	648 3	501 1	1149 4	0 8412	0 4637	1 3049	1800
1900	628 58	0 0252	0 2021	660 1	482 4	1142 4	0 8516	0 4433	1 2949	1900
2000	635 82	0 0257	0 1878	671 7	463 4	1135 1	0 8619	0 4230	1 2849	2000
2200	649 46	0 0268	0 1625	694 8	424 4	1119 2	0 8820	0 3826	1 2646	2200
2400	662 12	0 0280	0 1407	718 4	382 7	1101 1	0 9023	0 3411	1 2434	2400
2600	673 94	0 0295	0 1213	743 0	337 2	1080 2	0 9232	0 2973	1 2205	2600
2800	684 99	0 0315	0 1035	770 1	284 7	1054 8	0 9459	0 2487	1 1946	2800
3000	695 36	0 0346	0 0858	802 5	217 8	1020 3	0 9731	0 1885	1 1615	3000
3200	705 11	0 0444	0 0580	872.4	62 0	934 4	1 0320	0 0532	1 0852	3200
3206.2	705 40	0 0503	0 0503	902.7	0	902 7	1 0580	0	1 0580	3206.2

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PROPERTIES OF STEAM

Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. This change in state takes place at a definite and constant temperature which is determined solely by the pressure of the steam. The volume of a pound of steam is the *specific volume* which decreases as the pressure increases. The reciprocal of this, or the weight of steam per cubic foot, is the *density*. (See Table 8).

Steam which is in contact with the water from which it was generated is known as *saturated steam*. If it contains no actual water in the form of mist or priming, it is called *dry saturated steam*. If this be heated and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated*, that is, its temperature will be higher than that of saturated steam at the same pressure.

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PROBLEMS IN PRACTICE

1 ● Given air at 70 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.00 in. Hg., find the weight of vapor per pound of dry air.

Pressure of saturated vapor = e_t = 0.7387 in. Hg. (Table 6).
From Equation 5a,

$$W = 0.622 \left(\frac{0.5 \times 0.7387}{29.00 - (0.5) \times (0.7387)}\right)$$

W = 0.008024 lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

Approximate Method:

Weight of saturated vapor per pound of dry air = W_t = 0.01574 lb (Table 6). 0 01574 × 0.5 = 0.00787 lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

2 ● Given air with a dry-bulb temperature of 80 F, relative humidity of 55 per cent, and a barometric pressure of 28.85 in. Hg., calculate the weight of a cubic foot of mixture.

Pressure of saturated vapor at 80 F = e_t = 1.0316 in. Hg. (Table 6).

Pressure of the vapor in the mixture = 1.0316 × 0.55 = 0.5676 in. Hg.

Pressure of the dry air in the mixture = 28.85 − 0.5676 = 28.282 in. Hg.

$pV = wR\ (t + 460)$ (R = 0.753 when partial pressure of air is expressed in in. Hg.). 28.282 × 1 = d_a × 0.753 × (80 + 460)

$$d_a = \frac{28.282}{0.753 \times 540} = 0.06955 \text{ lb} = \text{weight of dry air in 1 cu ft of the mixture.}$$

Likewise from Equation 4a,

$$d_v = \frac{0.5676}{1\ 21 \times 540} = 0.000868 \text{ lb} = \text{weight of vapor per cubic foot at 55 per cent}$$
relative humidity.

Weight of 1 cu ft of the mixture = 0.06955 + 0.000868 = 0.070418 lb.

3 ● Given air with a dry-bulb temperature of 75 F, a relative humidity of 60 per cent, and a barometric pressure of 28.80 in. Hg., calculate the volume of 1 lb of the mixture.

Pressure of saturated vapor at 75 F = e_t = 0 8745 in. Hg.

Pressure of vapor in the mixture = 0.8745 × 0.6 = 0.525 in. Hg.

Pressure of dry air in the mixture = 28.80 − 0.525 = 28.275 in. Hg.

$$d_a = \frac{28\ 275}{0.753 \times 535} = 0.07018 \text{ lb} = \text{weight of dry air in 1 cu ft of the mixture.}$$

From Equation 4a,

$$d_v = \frac{0.525}{1\ 21 \times 535} = 0\ 000811 \text{ lb} = \text{weight of vapor per cubic foot at 55 per cent}$$
relative humidity.

Weight of 1 cu ft of the mixture = 0.07018 + 0.000811 = 0 070991 lb

$$\text{Volume of 1 lb of the mixture} = \frac{1}{0.070991} = 14.08 \text{ cu ft.}$$

4 ● It is desired to maintain a temperature of 80 F and a relative humidity of 50 per cent in a factory where the equipment gives off 6000 Btu per hour. If the entering air is at 70 F with an average barometric pressure of 29.92 in. Hg.; determine the relative humidity, and the pounds of air required per hour if there is no heat interchange between the walls, windows, or floors of the building.

Pressure of saturated vapor at 80 F = 1 0316 in. Hg. (Table 6).

Pressure of vapor in the mixture = 1.0316 × 0.5 = 0.5158 in. Hg.

$$W = 0\ 622 \left(\frac{0.5158}{29\ 92 - 0.5158}\right) = 0.01091 \text{ lb.}$$

Pressure of saturated vapor at 70 F = 0.7387 in. Hg.

With the same specific humidity

$$0.01091 = 0.622 \left(\frac{0.7387 \times \Phi}{29\ 92 - (0.7387 \times \Phi)}\right)$$

Φ = 69.8 per cent relative humidity at 70 F.

$h = 0.24 \times 80 + 0.01091\,[1059.2 + (0.45 \times 80)] = 31.15$ Btu per pound, the enthalpy of the mixture at 80 F and 50 per cent relative humidity.

$h = 0.24 \times 70 + 0\,01091\,[1059.2 + (0.45 \times 70)] = 28.70$ Btu per pound, the enthalpy of the mixture at 70 F and the same specific humidity.

$31.15 - 28.70 = 2.45$ Btu to be removed per pound of air.

6000 Btu = heat given off by equipment per hour.

$\frac{6000}{2.45} = 2449$ lb of air required per hour.

5 ● A building requires 50,000 cu ft of air per hour measured at standard barometric pressure of 29.92 in. Hg. to be raised from −10 F dry-bulb and 75 per cent relative humidity to 72 F dry-bulb and 30 per cent relative humidity. Determine the amount of heat and the weight of water which it is necessary to supply per hour if the temperature of the supply water is 50 F and the barometric pressure is 28.75 in. Hg.

Assume air volume to be dry air at 70 F.

Weight of air $= 0.075 \times 50{,}000 = 3750$ lb per hour.

From Table 6,

Pressure of vapor in the mixture, outside air $= 0.75 \times 0.0221 = 0.0166$ in. Hg.

Specific humidity, outside air $= 0.622 \left(\frac{0.0166}{28.75 - 0.0166}\right) = 0.0003589$ lb.

From Table 6,

Pressure of vapor in the mixture, inside air $= 0.30 \times 0.7906 = 0.2372$ in. Hg.

Specific humidity, inside air $= 0.622 \left(\frac{0.2372}{28.75 - 0.23718}\right) = 0.005174$ lb.

Water to be added $= 3750\,(0.005174 - 0.0003589) = 18.06$ lb per hour.

Enthalpy, inside air $= 0.24 \times 72 + 0.005174\,[1059.2 + (0.45 \times 72)] = 22.925$ Btu per pound.

Enthalpy, outside air $= 0.24 \times (-10) + 0.0003589\,[1059.2 + (0\,45 \times -10)] = -2.021$ Btu per pound.

Btu added incident to the water per pound of dry air.

$(0.005174 - 0.0003589)\,(50 - 32) = 0.0867$ Btu per pound.

Heat requirement per hour $= [22.925 - (-2.021 + 0.0867)] \times 3750 = 93{,}221$ Btu.

6 ● Determine the amount of heat and water that must be extracted to cool 3750 lb of air (weighed dry) from 95 F and 60 per cent relative humidity to 50 F and 100 per cent relative humidity with a barometric pressure of 28.75 in. Hg.

Pressure of vapor in the mixture, outside air $= 0.6 \times 1.659 = 0.995$ in. Hg.

Specific humidity, outside air $= 0.622 \left(\frac{0.995}{28.75 - 0.995}\right) = 0.0223$ lb.

Specific humidity, inside air = 0.007626 lb.

Weight of water to be extracted per hour $= (0.0223 - 0.007626) \times 3750 = 55.03$ lb.

Enthalpy, outside air $= 0.24 \times 95 + 0.0223\,[1059.2 + (0.45 \times 95)] = 47.37$ Btu per pound.

Enthalpy, inside air $= 0.24 \times 50 + 0.00764\,[1059.2 + (0.45 \times 50)] = 20.26$ Btu per pound.

Heat to be extracted $= (47.37 - 20.26) \times 3750 = 101{,}662$ Btu.

Chapter 2

REFRIGERANTS AND AIR DRYING AGENTS

Properties of Refrigerant Substances, Selection Factors, Solid Adsorbents, Liquid Absorbents, Nature of Processes, Temperature Pressure Concentration Relations

BOTH cooling and dehumidification of air are usually desirable at certain times. Cooling may be regarded as the extraction of sensible heat while dehumidification necessitates the removal of latent heat. By a suitable selection and combination of methods and equipment these two processes may be accomplished simultaneously or they may be secured independent of each other and at different times as desired. On occasion, the desired result can be secured by extracting sensible heat only while in other cases drying alone will produce the end conditions sought. Suitable substances must be available for use in every particular case.

Generally, when both cooling and dehumidification are sought current practice makes use of artificially produced refrigeration in some form. The substances used as the heat-carrying agents in these applications are called refrigerants. Air drying agents are substances which permit dehumidification of air as a process separate and independent of the extraction of sensible heat from it. Refrigerants and air drying agents are treated separately in this chapter where the aim is to present a statement of the properties of these substances which are of especial importance in air conditioning work. Little mention is made here of methods, systems, or equipment whereby these substances may be applied, which information may be found in Chapter 23.

REFRIGERANTS

Since air cooling and dehumidification are frequently accomplished by evaporating a liquid under circumstances which will permit the heat necessary to be extracted from the air, the refrigerants are substances which are capable of being changed into liquids or vapors within workable temperature and pressure ranges. There are many substances which might be so used but in practice the choice is limited by a wide variety of considerations including availability, cost, safety, chemical stability and adaptability to the type of refrigerating system to be used.

In this chapter detailed consideration is limited to six substances, viz: ammonia, carbon dioxide, dichlorodifluoromethane (F_{12}), methyl

TABLE 1. PROPERTIES OF AMMONIA

Sat. Temp F	Abs Press Lb per Sq In	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct	Entropy	Ht Ct	Entropy
0	30.42	0.02419	9.116	42.9	611.8	0.0975	1.3352	666.8	1.4439	720.3	1.5317
2	31.92	0.02424	8.714	45.1	612.4	0.1022	1.3312	667.6	1.4400	721.2	1.5277
4	33.47	0.02430	8.333	47.2	613.0	0.1069	1.3273	668.4	1.4360	722.2	1.5236
5	34.27	0.02432	8.150	48.3	613.3	0.1092	1.3253	668.8	1.4340	722.6	1.5216
6	35.09	0.02435	7.971	49.4	613.6	0.1115	1.3234	669.3	1.4321	723.1	1.5196
8	36.77	0.02440	7.629	51.6	614.3	0.1162	1.3195	670.1	1.4281	724.1	1.5155
10	38.51	0.02446	7.304	53.8	614.9	0.1208	1.3157	670.9	1.4242	725.0	1.5115
12	40.31	0.02451	6.996	56.0	615.5	0.1254	1.3118	671.7	1.4205	725.9	1.5077
14	42.18	0.02457	6.703	58.2	616.1	0.1300	1.3081	672.5	1.4168	726.8	1.5039
16	44.12	0.02462	6.425	60.3	616.6	0.1346	1.3043	673.4	1.4130	727.8	1.5001
18	46.13	0.02468	6.161	62.5	617.2	0.1392	1.3006	674.2	1.4093	728.7	1.4963
20	48.21	0.02474	5.910	64.7	617.8	0.1437	1.2969	675.0	1.4056	729.6	1.4925
22	50.36	0.02479	5.671	66.9	618.3	0.1483	1.2933	675.8	1.4021	730.5	1.4889
24	52.59	0.02485	5.443	69.1	618.9	0.1528	1.2897	676.6	1.3985	731.4	1.4853
26	54.90	0.02491	5.227	71.3	619.4	0.1573	1.2861	677.3	1.3950	732.4	1.4816
28	57.28	0.02497	5.021	73.5	619.9	0.1618	1.2825	678.1	1.3914	733.3	1.4780
30	59.74	0.02503	4.825	75.7	620.5	0.1663	1.2790	678.9	1.3879	734.2	1.4744
32	62.29	0.02508	4.637	77.9	621.0	0.1708	1.2755	679.7	1.3846	735.1	1.4710
34	64.91	0.02514	4.459	80.1	621.5	0.1753	1.2721	680.4	1.3812	736.0	1.4676
36	67.63	0.02521	4.289	82.3	622.0	0.1797	1.2686	681.2	1.3779	736.8	1.4643
38	70.43	0.02527	4.126	84.6	622.5	0.1841	1.2652	681.9	1.3745	737.7	1.4609
39	71.87	0.02530	4.048	85.7	622.7	0.1863	1.2635	682.3	1.3729	738.2	1.4592
40	73.32	0.02533	3.971	86.8	623.0	0.1885	1.2618	682.7	1.3712	738.6	1.4575
41	74.80	0.02536	3.897	87.9	623.2	0.1908	1.2602	683.1	1.3696	739.0	1.4559
42	76.31	0.02539	3.823	89.0	623.4	0.1930	1.2585	683.4	1.3680	739.5	1.4542
44	79.38	0.02545	3.682	91.2	623.9	0.1974	1.2552	684.2	1.3648	740.4	1.4510
46	82.55	0.02551	3.547	93.5	624.4	0.2018	1.2519	684.9	1.3616	741.3	1.4477
48	85.82	0.02558	3.418	95.7	624.8	0.2062	1.2486	685.6	1.3584	742.2	1.4445
50	89.19	0.02564	3.294	97.9	625.2	0.2105	1.2453	686.4	1.3552	743.1	1.4412
52	92.66	0.02571	3.176	100.2	625.7	0.2149	1.2421	687.1	1.3521	744.0	1.4382
54	96.23	0.02577	3.063	102.4	626.1	0.2192	1.2389	687.8	1.3491	744.8	1.4351
56	99.91	0.02584	2.954	104.7	626.5	0.2236	1.2357	688.5	1.3460	745.7	1.4321
58	103.7	0.02590	2.851	106.9	626.9	0.2279	1.2325	689.2	1.3430	746.5	1.4290
60	107.6	0.02597	2.751	109.2	627.3	0.2322	1.2294	689.9	1.3399	747.4	1.4260
62	111.6	0.02604	2.656	111.5	627.7	0.2365	1.2262	690.6	1.3370	748.2	1.4231
64	115.7	0.02611	2.565	113.7	628.0	0.2408	1.2231	691.3	1.3341	749.1	1.4202
66	120.0	0.02618	2.477	116.0	628.4	0.2451	1.2201	691.9	1.3312	749.9	1.4172
68	124.3	0.02625	2.393	118.3	628.8	0.2494	1.2170	692.6	1.3283	750.8	1.4143
70	128.8	0.02632	2.312	120.5	629.1	0.2537	1.2140	693.3	1.3254	751.6	1.4114
72	133.4	0.02639	2.235	122.8	629.4	0.2579	1.2110	694.0	1.3226	752.4	1.4086
74	138.1	0.02646	2.161	125.1	629.8	0.2622	1.2080	694.6	1.3199	753.3	1.4059
76	143.0	0.02653	2.089	127.4	630.1	0.2664	1.2050	695.3	1.3171	754.1	1.4031
78	147.9	0.02661	2.021	129.7	630.4	0.2706	1.2020	695.9	1.3144	755.0	1.4004
80	153.0	0.02668	1.955	132.0	630.7	0.2749	1.1991	696.6	1.3116	755.8	1.3976
82	158.3	0.02675	1.892	134.3	631.0	0.2791	1.1962	697.2	1.3089	756.6	1.3949
84	163.7	0.02684	1.831	136.6	631.3	0.2833	1.1933	697.8	1.3063	757.4	1.3923
86	169.2	0.02691	1.772	138.9	631.5	0.2875	1.1904	698.5	1.3040	758.3	1.3896
88	174.8	0.02699	1.716	141.2	631.8	0.2917	1.1875	699.1	1.3010	759.1	1.3870
90	180.6	0.02707	1.661	143.5	632.0	0.2958	1.1846	699.7	1.2983	759.9	1.3843
92	186.6	0.02715	1.609	145.8	632.2	0.3000	1.1818	700.3	1.2957	760.7	1.3818
94	192.7	0.02723	1.559	148.2	632.5	0.3041	1.1789	700.9	1.2932	761.5	1.3793
96	198.9	0.02731	1.510	150.5	632.6	0.3083	1.1761	701.5	1.2906	762.2	1.3768
98	205.3	0.02739	1,464	152.9	632.9	0.3125	1.1733	702.1	1.2881	763.0	1.3743
100	211.9	0.02747	1.419	155.2	633.0	0.3166	1.1705	702.7	1.2855	763.8	1.3718
102	218.6	0.02756	1.375	157.6	633.2	0.3207	1.1677	703.3	1.2830	764.6	1.3693
104	225.4	0.02764	1.334	159.9	633.4	0.3248	1.1649	703.8	1.2805	765.3	1.3668
106	232.5	0.02773	1.293	162.3	633.5	0.3289	1.1621	704.3	1.2780	766.1	1.3643
108	239.7	0.02782	1.254	164.6	633.6	0.3330	1.1593	705.0	1.2755	766.9	1.3619
110	247.0	0.02790	1.217	167.0	633.7	0.3372	1.1566	705.5	1.2731	767.6	1.3596
112	254.5	0.02799	1.180	169.4	633.8	0.3413	1.1538	706.1	1.2708	768.3	1.3573
114	262.2	0.02808	1.145	171.8	633.9	0.3453	1.1510	706.6	1.2684	769.1	1.3550
116	270.1	0.02817	1.112	174.2	634.0	0.3495	1.1483	707.2	1.2661	769.8	1.3527
118	278.2	0.02827	1.079	176.6	634.0	0.3535	1.1455	707.7	1.2636	770.5	1.3503
120	286.4	0.02836	1.047	179.0	634.0	0.3576	1.1427	708.2	1.2612	771.3	1.3479
122	294.8	0.02846	1.017	181.4	634.0	0.3618	1.1400	708.6	1.2587	772.0	1.3455
124	303.4	0.02855	0.987	183.9	634.0	0.3659	1.1372	709.1	1.2563	772.8	1.3431
126	312.2	0.02865	0.958	186.3	633.9	0.3700	1.1344	709.6	1.2538	773.5	1.3407
128	321.2	0.02875	0.931	188.8	633.9	0.3741	1.1316	710.0	1.2513	774.2	1.3383

TABLE 2. PROPERTIES OF DICHLORODIFLUOROMETHANE(F_{12})

Sat Temp F	Abs Press Lb per Sq In	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct.	Entropy	Ht Ct.	Entropy
0	23 87	0 0110	1 637	8 25	78 21	0 01869	0 17091	81 71	0 17829	85 26	0 18547
2	24 89	0 0110	1 574	8 67	78 44	0 01961	0 17075	81 94	0 17812	85 51	0 18529
4	25 96	0 0111	1 514	9 10	78 67	0 02052	0 17060	82 17	0 17795	85 76	0 18511
5	26 51	0 0111	1 485	9 32	78 79	0 02097	0 17052	82.29	0 17786	85 89	0 18502
6	27 05	0 0111	1 457	9 53	78 90	0 02143	0 17045	82 41	0.17778	86 01	0 18494
8	28 18	0 0111	1 403	9 96	79 13	0 02235	0.17030	82 66	0 17763	86 26	0 18477
10	29 35	0 0112	1 351	10 39	79 36	0 02328	0 17015	82 90	0 17747	86.51	0 18460
12	30 56	0 0112	1 301	10 82	79 59	0 02419	0 17001	83 14	0 17733	86.76	0 18444
14	31 80	0 0112	1 253	11 26	79 82	0 02510	0 16987	83 38	0 17720	87 01	0 18429
16	33 08	0 0112	1.207	11 70	80 05	0 02601	0 16974	83.61	0 17706	87 26	0 18413
18	34 40	0 0113	1 163	12 12	80 27	0 02692	0 16961	83 85	0 17693	87 51	0 18397
20	35 75	0 0113	1 121	12 55	80 49	0 02783	0 16949	84 09	0 17679	87 76	0 18382
22	37 15	0 0113	1 081	13 00	80 72	0 02873	0 16938	84 32	0 17666	88 00	0 18369
24	38 58	0 0113	1 043	13 44	80 95	0 02963	0 16926	84 55	0 17652	88 24	0 18355
26	40 07	0 0114	1 007	13 88	81 17	0 03053	0 16913	84 79	0 17639	88 49	0 18342
28	41 59	0 0114	0 973	14 32	81 39	0 03143	0 16900	85 02	0 17625	88 73	0 18328
30	43 16	0 0115	0 939	14 76	81 61	0 03233	0 16887	85 25	0 17612	88 97	0 18315
32	44 77	0 0115	0 908	15 21	81 83	0 03323	0 16876	85 48	0 17600	89 21	0 18303
34	46 42	0 0115	0 877	15 65	82 05	0 03413	0 16865	85 71	0 17589	89 45	0 18291
36	48 13	0 0116	0 848	16 10	82 27	0 03502	0 16854	85 95	0.17577	89 68	0 18280
38	49 88	0 0116	0 819	16 55	82 49	0 03591	0 16843	86 18	0 17566	89 92	0 18268
39	50 78	0 0116	0 806	16 77	82 60	0 03635	0 16838	86 29	0 17560	90 04	0 18262
40	51 68	0 0116	0 792	17 00	82 71	0 03680	0 16833	86 41	0 17554	90 16	0 18256
41	52 70	0 0116	0 779	17 23	82 82	0 03725	0 16828	86 52	0 17549	90 28	0 18251
42	53 51	0 0116	0 767	17 46	82 93	0 03770	0 16823	86 64	0 17544	90 40	0 18245
44	55 40	0 0117	0 742	17 91	83 15	0 03859	0 16813	86 86	0 17534	90 65	0 18235
46	57 35	0 0117	0 718	18 36	83 36	0 03948	0 16803	87 09	0 17525	90 89	0 18224
48	59 35	0 0117	0 695	18 82	83 57	0 04037	0 16794	87 31	0 17515	91 14	0 18214
50	61 39	0 0118	0 673	19 27	83 78	0 04126	0 16785	87 54	0 17505	91 38	0 18203
52	63 49	0 0118	0 652	19 72	83 99	0 04215	0 16776	87 76	0 17496	91 61	0 18193
54	65 63	0 0118	0 632	20 18	84 20	0 04304	0 16767	87 98	0 17486	91 83	0 18184
56	67 84	0 0119	0 612	20 64	84 41	0 04392	0 16758	88 20	0 17477	92 06	0 18174
58	70 10	0 0119	0 593	21 11	84 62	0 04480	0 16749	88 42	0 17467	92 28	0 18165
60	72 41	0 0119	0 575	21 57	84 82	0 04568	0 16741	88 64	0 17458	92 51	0 18155
62	74 77	0 0120	0 557	22 03	85 02	0 04657	0 16733	88 86	0 17450	92 74	0 18147
64	77 20	0 0120	0 540	22 49	85 22	0 04745	0 16725	89 07	0 17442	92 97	0 18139
66	79 67	0 0120	0 524	22 95	85 42	0 04833	0 16717	89 29	0 17433	93 20	0 18130
68	82 24	0 0121	0 508	23 42	85 62	0 04921	0 16709	89 50	0 17425	93 43	0 18122
70	84 82	0 0121	0 493	23 90	85 82	0 05009	0 16701	89 72	0 17417	93 66	0 18114
72	87 50	0 0121	0 479	24 37	86 02	0 05097	0 16693	89 93	0 17409	93 99	0 18106
74	90 20	0 0122	0 464	24 84	86 22	0 05185	0 16685	90 14	0 17402	94 12	0 18098
76	93 00	0 0122	0 451	25 32	86 42	0 05272	0 16677	90 36	0 17394	94 34	0 18091
78	95 85	0 0123	0 438	25 80	86 61	0 05359	0 16669	90 57	0 17387	94 57	0 18083
80	98 76	0 0123	0 425	26 28	86 80	0 05446	0 16662	90 78	0 17379	94 80	0 18075
82	101 70	0 0123	0 413	26 76	86 99	0 05534	0 16655	90 98	0 17372	95 01	0 18068
84	104 8	0 0124	0 401	27 24	87 18	0 05621	0 16648	91 18	0 17365	95 22	0 18061
86	107 9	0 0124	0 389	27 72	87 37	0 05708	0 16640	91 37	0 17358	95 44	0 18054
88	111 1	0 0124	0 378	28 21	87 56	0 05795	0 16632	91 57	0 17351	95 65	0 18047
90	114 3	0 0125	0 368	28 70	87 74	0 05882	0 16624	91 77	0 17344	95 86	0 18040
92	117 7	0 0125	0 357	29 19	87 92	0 05969	0 16616	91 97	0 17337	96 07	0 18033
94	121 0	0 0126	0 347	29 68	88 10	0 06056	0 16608	92 16	0 17330	96 28	0 18026
96	124 5	0 0126	0 338	30 18	88 28	0 06143	0 16600	92 36	0 17322	96 50	0 18018
98	128 0	0 0126	0 328	30 67	88 45	0 06230	0 16592	92 55	0 17315	96 71	0 18011
100	131 6	0 0127	0 319	31 16	88,62	0 06316	0 16584	92 75	0 17308	96 92	0 18004
102	135 3	0 0127	0 310	31 65	88 79	0 06403	0 16576	92 93	0 17301	97 12	0 17998
104	139 0	0 0128	0 302	32 15	88 95	0 06490	0 16568	93 11	0 17294	97 32	0 17993
106	142 8	0 0128	0 293	32 65	89 11	0 06577	0 16560	93 30	0 17288	97 53	0 17987
108	146 8	0 0129	0 285	33 15	89 27	0 06663	0 16551	93 48	0 17281	97 73	0 17982
110	150 7	0 0129	0 277	33 65	89 43	0 06749	0 16542	93 66	0 17274	97 93	0 17976
112	154 8	0 0130	0 269	34 15	89 58	0 06836	0 16533	93 82	0 17266	98 11	0 17969
114	158 9	0 0130	0 262	34 65	89 73	0 06922	0 16524	93 98	0 17258	98 29	0 17961
116	163 1	0 0131	0 254	35 15	89 87	0 07008	0 16515	94 15	0 17249	98 48	0 17954
118	167 4	0 0131	0 247	35 65	90 01	0 07094	0 16505	94 31	0 17241	98 66	0 17946
120	171 8	0 0132	0 240	36 16	90 15	0 07180	0 16495	94 47	0 17233	98 84	0 17939
122	176 2	0 0132	0 233	36 66	90 28	0 07266	0 16484	94 63	0 17224	99 01	0 17931
124	180 8	0 0133	0 227	37 16	90 40	0 07352	0 16473	94 78	0 17215	99 18	0 17922
126	185 4	0 0133	0 220	37 67	90 52	0 07437	0 16462	94 94	0 17206	99 35	0 17914
128	190 1	0 0134	0 214	38 18	90 64	0 07522	0 16450	95 09	0 17196	99 53	0 17906
130	194 9	0 0134	0 208	38 69	90 76	0 07607	0 16438	95 25	0 17186	99 70	0 17897
132	199 8	0 0135	0 202	39 19	90 86	0 07691	0 16425	95 41	0 17176	99 87	0 17889
134	204 8	0 0135	0 196	39 70	90 96	0 07775	0 16411	95 56	0 17166	100 04	0 17881
136	209 9	0 0136	0.191	40.21	91 06	0 07858	0 16396	95 72	0 17156	100 22	0 17873
138	215 0	0 0137	0 185	40 72	91 15	0 07941	0 16380	95 87	0 17145	100 39	0 17864
140	220.2	0 0138	0 180	41 24	91 24	0 08024	0 16363	96 03	0 17134	100 56	0 17856

TABLE 3. PROPERTIES OF METHYL CHLORIDE

Sat. Temp. F	Abs Press Lb per Sq In	Volume		Heat Content and Entropy Taken From −40 F: Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct	Entropy	Ht. Ct	Entropy
0	18.73	0.0162	5.052	14.4	192.4	0.0328	0.4197	215.6	0.467	237.2	0.507
2	19.60	0.0162	4.856	15.1	193.1	0.0344	0.4196	216.2	0.466	237.7	0.505
4	20.47	0.0163	4.661	15.8	193.8	0.0360	0.4195	216.7	0.465	238.2	0.504
5	20.91	0.0163	4.563	16.2	194.1	0.0368	0.4195	217.0	0.464	238.5	0.503
6	21.39	0.0163	4.476	16.6	194.4	0.0376	0.4194	217.3	0.464	238.8	0.502
8	22.34	0.0164	4.303	17.3	195.1	0.0391	0.4193	217.9	0.463	239.4	0.501
10	23.30	0.0164	4.129	18.1	195.8	0.0407	0.4192	218.5	0.463	240.0	0.500
12	24.38	0.0164	3.984	18.8	196.3	0.0423	0.4184	219.0	0.462	240.5	0.499
14	25.46	0.0164	3.839	19.6	196.7	0.0439	0.4176	219.5	0.462	241.0	0.498
16	26.55	0.0165	3.693	20.3	197.2	0.0454	0.4168	220.0	0.461	241.5	0.498
18	27.63	0.0165	3.548	21.1	197.6	0.0472	0.4160	220.5	0.461	242.0	0.497
20	28.71	0.0166	3.403	21.8	198.1	0.0486	0.4152	221.0	0.460	242.5	0.496
22	29.98	0.0166	3.288	22.5	198.5	0.0501	0.4148	221.5	0.459	243.0	0.495
24	31.25	0.0166	3.172	23.3	198.9	0.0516	0.4143	222.0	0.459	243.6	0.495
26	32.53	0.0167	3.057	24.0	199.3	0.0532	0.4139	222.4	0.458	244.1	0.494
28	33.80	0.0167	2.941	24.8	199.7	0.0547	0.4134	222.9	0.458	244.7	0.494
30	35.07	0.0168	2.826	25.5	200.1	0.0562	0.4130	223.4	0.457	245.2	0.493
32	36.55	0.0168	2.734	26.2	200.5	0.0577	0.4124	223.9	0.456	245.7	0.492
34	38.03	0.0169	2.642	27.0	200.9	0.0592	0.4118	224.3	0.455	246.2	0.492
36	39.51	0.0169	2.549	27.7	201.4	0.0607	0.4111	224.8	0.455	246.7	0.491
38	40.99	0.0169	2.457	28.5	201.8	0.0622	0.4105	225.2	0.454	247.2	0.491
39	41.73	0.0170	2.411	28.8	202.0	0.0629	0.4102	225.5	0.453	247.4	0.490
40	42.47	0.0170	2.365	29.2	202.2	0.0637	0.4099	225.7	0.453	247.7	0.490
41	43.33	0.0170	2.328	29.6	202.4	0.0644	0.4096	225.9	0.453	248.0	0.490
42	44.18	0.0171	2.290	29.9	202.6	0.0651	0.4093	226.1	0.452	248.3	0.489
44	45.89	0.0171	2.216	30.7	203.0	0.0666	0.4087	226.6	0.451	248.8	0.489
46	47.61	0.0171	2.141	31.4	203.3	0.0680	0.4081	227.0	0.451	249.4	0.488
48	49.32	0.0172	2.067	32.2	203.7	0.0695	0.4075	227.5	0.450	249.9	0.488
50	51.03	0.0172	1.992	32.9	204.1	0.0709	0.4069	227.9	0.449	250.5	0.487
52	53.00	0.0172	1.931	33.7	204.4	0.0724	0.4063	228.2	0.448	251.0	0.486
54	54.97	0.0173	1.870	34.4	204.7	0.0739	0.4056	228.6	0.448	251.5	0.486
56	56.94	0.0173	1.810	35.2	205.1	0.0754	0.4050	228.9	0.447	252.0	0.485
58	58.91	0.0173	1.749	35.9	205.4	0.0769	0.4043	229.3	0.447	252.5	0.485
60	60.88	0.0174	1.688	36.7	205.7	0.0784	0.4037	229.6	0.446	253.0	0.484
62	63.13	0.0174	1.638	37.4	206.0	0.0798	0.4030	229.9	0.445	253.5	0.483
64	65.37	0.0174	1.588	38.2	206.3	0.0812	0.4024	230.3	0.444	254.0	0.483
66	67.62	0.0175	1.539	38.9	206.6	0.0827	0.4017	230.6	0.443	254.5	0.482
68	69.86	0.0175	1.489	39.7	206.9	0.0841	0.4011	231.0	0.442	255.0	0.482
70	72.11	0.0176	1.439	40.4	207.2	0.0855	0.4004	231.3	0.441	255.5	0.481
72	74.66	0.0176	1.398	41.1	207.5	0.0869	0.3998	231.6	0.440	256.0	0.480
74	77.21	0.0177	1.357	41.9	207.7	0.0883	0.3992	232.0	0.439	256.5	0.480
76	79.76	0.0177	1.315	42.6	208.0	0.0898	0.3985	232.3	0.439	256.9	0.479
78	82.31	0.0178	1.274	43.4	208.2	0.0912	0.3979	232.7	0.438	257.4	0.479
80	84.86	0.0178	1.233	44.1	208.5	0.0926	0.3973	233.0	0.437	257.9	0.478
82	87.74	0.0178	1.199	44.8	208.7	0.0940	0.3967	233.3	0.436	258.4	0.478
84	90.62	0.0179	1.165	45.6	209.0	0.0953	0.3960	233.6	0.435	258.9	0.477
86	93.50	0.0179	1.130	46.3	209.2	0.0967	0.3954	233.9	0.435	259.4	0.477
88	96.38	0.0180	1.096	47.1	209.5	0.0980	0.3947	234.2	0.434	259.9	0.476
90	99.26	0.0180	1.062	47.8	209.7	0.0994	0.3941	234.5	0.433	260.4	0.476
92	102.49	0.0180	1.033	48.6	209.9	0.1008	0.3935	234.8	0.433	260.8	0.476
94	105.72	0.0181	1.005	49.3	210.2	0.1022	0.3929	235.1	0.432	261.2	0.475
96	108.94	0.0181	0.9764	50.1	210.4	0.1035	0.3922	235.4	0.432	261.6	0.475
98	112.17	0.0182	0.9478	50.8	210.7	0.1049	0.3916	235.7	0.431	262.0	0.474
100	115.40	0.0182	0.9193	51.6	210.9	0.1063	0.3910	236.0	0.431	262.4	0.474
102	119.00	0.0183	0.8952	52.3	211.1	0.1076	0.3903	236.4	0.430	262.8	0.474
104	122.60	0.0183	0.8712	53.1	211.3	0.1090	0.3897	236.8	0.430	263.2	0.473
106	126.20	0.0184	0.8471	53.8	211.4	0.1103	0.3890	237.1	0.429	263.5	0.473
108	129.80	0.0184	0.8231	54.6	211.6	0.1117	0.3884	237.5	0.429	263.9	0.472
110	133.40	0.0185	0.7990	55.3	211.8	0.1130	0.3877	237.9	0.428	264.3	0.472
112	137.42	0.0185	0.7786	56.1	212.0	0.1144	0.3871	238.1	0.427	264.6	0.471
114	141.44	0.0185	0.7583	56.8	212.2	0.1157	0.3864	238.3	0.427	264.8	0.470
116	145.46	0.0186	0.7379	57.6	212.4	0.1171	0.3858	238.6	0.426	265.1	0.470
118	149.48	0.0186	0.7176	58.3	212.6	0.1184	0.3851	238.8	0.426	265.3	0.469
120	153.50	0.0187	0.6972	59.1	212.8	0.1198	0.3845	239.0	0.425	265.6	0.468

TABLE 4. PROPERTIES OF CARBON DIOXIDE

Sat Temp F	Abs Press Lb per Sq In	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct	Entropy	Ht Ct	Entropy
0	305 5	0 01570	0 29040	18 8	138 9	0 0418	0 3024	153 7	0 3342	167 5	0 3612
2	315 9	0 01579	0 28030	19 8	138 8	0 0440	0 3014	153 7	0 3330	167 6	0 3600
4	326.5	0 01588	0 27070	20 8	138 8	0 0461	0 3005	153 7	0 3318	167 7	0 3588
5	332 0	0 01592	0 26610	21 3	138 8	0 0472	0 3000	153.7	0 3312	167 7	0 3582
6	337 4	0.01596	0 26140	21.8	138 7	0 0483	0.2994	153 7	0 3306	167 8	0.3576
8	348.7	0 01605	0 25260	22 9	138 7	0 0504	0 2982	153 7	0 3293	167 9	0.3563
10	360 2	0 01614	0 24370	24 0	138 7	0 0526	0 2970	153 7	0 3281	168 0	0.3550
12	371 9	0 01623	0 23540	25 0	138 6	0 0548	0 2958	153 7	0 3270	168 1	0 3538
14	383 9	0 01632	0 22740	26 1	138 6	0 0571	0 2946	153 7	0 3259	168 2	0 3526
16	396 2	0 01642	0 21970	27 2	138 5	0 0593	0 2933	153 7	0 3249	168.3	0 3513
18	408 9	0 01652	0 21210	28 3	138 4	0 0616	0 2921	153 7	0 3238	168 5	0 3501
20	421 8	0 01663	0 20490	29 4	138 3	0 0638	0 2909	153 7	0 3227	168 6	0 3489
22	434 0	0 01673	0 19790	30 5	138 2	0 0662	0 2897	153 7	0 3214	168 7	0 3479
24	448 4	0 01684	0 19120	31 7	138 1	0 0686	0 2885	153 7	0 3202	168 8	0 3470
26	462 2	0 01695	0 18460	32 9	138 0	0 0710	0 2873	153 7	0 3189	168 9	0.3460
28	476 3	0 01707	0 17830	34 1	137 9	0 0734	0 2861	153 7	0 3177	169 0	0 3451
30	490 8	0 01719	0 17220	35 4	137 8	0 0758	0 2849	153 7	0 3164	169 1	0 3441
32	505 5	0 01731	0 16630	36 7	137 7	0 0781	0 2834	153 7	0 3158	169 2	0 3431
34	522 6	0 01744	0 16030	37 9	137 4	0 0804	0 2820	153 7	0 3151	169 3	0 3421
36	536 0	0 01759	0 15500	39 1	137 2	0 0828	0 2805	153 7	0 3145	169 4	0 3411
38	551 7	0 01773	0 14960	40 4	136 9	0 0851	0 2791	153 7	0 3138	169 5	0.3401
39	559 7	0 01780	0 14700	41 0	136 8	0 0862	0 2783	153 7	0 3135	169 5	0 3396
40	567 8	0 01787	0 14440	41 7	136 7	0 0874	0 2776	153 7	0 3132	169 6	0 3391
41	576 0	0 01794	0 14185	42 3	136 5	0 0887	0 2768	153 7	0 3127	169 6	0 3386
42	584 3	0 01801	0 13930	42 9	136 3	0 0899	0 2761	153 7	0 3122	169 7	0 3381
44	601 1	0 01817	0 13440	44 3	136 1	0 0924	0 2745	153 7	0 3112	169 8	0 3371
46	618 2	0 01834	0 12970	45 6	135 7	0 0950	0 2730	153 7	0 3101	169 9	0 3362
48	635 7	0 01851	0 12500	47 0	135 4	0 0975	0 2714	153 7	0 3091	170 0	0 3352
50	653 6	0 01868	0 12050	48 4	135 0	0 1000	0 2699	153 7	0 3081	170 1	0 3342
52	671.9	0 01887	0 11610	49.8	134 5	0 1027	0 2681	153 7	0 3069	170 2	0 3333
54	690 6	0 01906	0 11170	51 2	133 9	0 1054	0 2663	153 7	0 3057	170 3	0 3324
56	709 5	0 01927	0 10750	52 6	133 4	0 1081	0 2644	153 7	0 3046	170 5	0 3315
58	728 8	0 01948	0 10340	54 0	132 7	0 1108	0 2626	153 7	0 3034	170 6	0 3306
60	748 6	0 01970	0 09940	55 5	132 1	0 1135	0 2608	153 7	0 3022	170 7	0 3297
62	769 0	0 01995	0 09545	57 0	131 3	0 1164	0 2584	153 7	0 3012	170 8	0 3289
64	789 4	0 02020	0 09180	58 6	130 6	0 1194	0 2560	153 7	0 3002	170 9	0 3281
66	810 3	0 02048	0 08800	60 2	129 7	0 1223	0 2535	153 7	0 2991	171 0	0 3273
68	831 6	0 02079	0 08422	61 9	128 7	0 1253	0 2511	153 7	0 2981	171 1	0 3265
70	853 4	0 02112	0 08040	63 7	127 5	0 1282	0 2487	153 7	0 2971	171 2	0 3257
72	875 8	0 02152	0 07654	65 5	126 0	0 1321	0 2450	153 7	0 2962	171 3	0 3250
74	898 2	0 02192	0 07269	67 3	124 5	0 1360	0 2414	153 7	0 2953	171 4	0 3242
76	921 3	0 02242	0 06875	69 4	122 8	0 1398	0 2377	153 7	0 2945	171 5	0 3235
78	944 8	0 02300	0 06473	71 6	120 9	0 1437	0 2341	153 7	0 2936	171 6	0 3227
80	968 7	0 02370	0 06064	73 9	118 7	0 1476	0 2304	153 7	0 2927	171 7	0 3220
82	993 0	0 02456	0 05648	76 4	116 6	0 1578	0 2195	153 7	0 2920	173 8	0 3215
84	1017 7	0 02553	0 05223	79 4	113 9	0 1679	0 2087	153 7	0 2914	176 0	0 3209
86	1043 0	0 02686	0 04789	83 3	110 4	0 1781	0 1978	153 7	0 2907	178 2	0.3204
87 8	1069 9	0 03454	0 03454	97 0	97 0	0 1880	0 1880	153 7	0 2901	180 1	0 3199

chloride, water, and monofluorotrichloromethane (F_{11}), for each of which a table is presented. Each table gives the principal physical properties of the saturated substance, and all are arranged in uniform fashion. In each case columns are included which give the heat content and entropy of the superheated vapor at two selected points. Tables 1, 2, 3 and 4 which include the refrigerants much used in reciprocating and rotary mechanical compression systems have a 2 F temperature interval. As water and F_{11} are much used in centrifugal compression systems the temperature interval in Tables 5 and 6 is 5 F.

TABLE 5. PROPERTIES OF MONOFLUOROTRICHLOROMETHANE (F_{11})

Sat Temp. F	Abs Press Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct	Entropy	Ht Ct.	Entropy
0	2.59	0.01020	13.700	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91.2	0.0200	0.1974	94.7	0.2047	98.2	0.2117
10	3.38	0.01028	10.700	9.82	92.0	0 0222	0.1973	95.5	0 2045	99.0	0.2114
15	3.85	0.01032	9.530	10.80	92.8	0 0243	0 1971	96.3	0.2043	99 8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0 0264	0 1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12.90	94.5	0 0286	0 1969	98 0	0.2039	101.5	0.2107
30	5.57	0.01045	6.770	13.90	95.3	0.0307	0.1969	98 8	0.2038	102.3	0.2105
35	6.27	0.01049	6.080	14.90	96.1	0.0328	0.1968	99.6	0 2037	103.1	0.2103
40	7.03	0 01053	5.460	16 00	96.8	0 0349	0.1968	100.3	0 2036	103 8	0 2101
45	7.88	0.01057	4.920	17.00	97.6	0 0370	0.1967	101.1	0.2035	104 6	0.2099
50	8.79	0.01062	4.440	18.10	98.4	0 0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4.020	19.10	99 2	0 0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.90	0 01071	3.640	20.20	100 0	0 0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21.30	100.8	0.0453	0 1967	104.3	0.2032	107 8	0.2094
70	13.40	0.01081	3.000	22.40	101.5	0 0473	0.1967	105.0	0 2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105 7	0.2031	109.2	0.2092
80	16.30	0.01091	2.500	24 50	102.9	0 0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.90	0 01096	2.280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110 6	0.2089
90	19.70	0.01101	2 090	26 70	104.4	0 0553	0.1966	107.9	0 2028	111.4	0.2088
95	21.60	0 01106	1.918	27.80	105.1	0 0573	0.1966	108 6	0.2028	112.1	0.2087
100	23.60	0.01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.90	0.01116	1.620	30.10	106.4	0.0613	0.1965	109.9	0.2026	113.4	0.2084

AIR DRYING AGENTS

Moisture may be removed from air, thus accomplishing dehumidification, by the use of any one of a number of substances if the moist air and these substances are brought together under suitable circumstances. One class of these substances are solids at ordinary conditions and have the power of adsorbing the moisture from the air. Another class of air drying agents are liquids under ordinary conditions and absorb the moisture from the air. Nearly all the drying agents in frequent commercial use in air conditioning installations are of one or the other of these two classes.

Adsorbents

These substances are characterized by a physical structure containing a great number of extremely small pores but still retaining sufficient mechanical strength to resist whatever wear and handling to which they are subjected. To be suitable for air drying purposes they must be widely available at economical cost, durable in use, stable in form and properties, and capable of withstanding the re-activation processes by which they are made ready for repeated use. They must also possess capacity for adsorbing and holding so sufficient a quantity of moisture that the dimensions of the beds necessary to accommodate them will be practical.

Aluminum Oxide, (Alumina), in a porous, amorphous form is a solid adsorbent frequently called by the common name *activated alumina*, and containing small amounts of hydrated aluminum oxide, and very small amounts of soda, and various metallic oxides. A good grade of activated

alumina will show 92 per cent of Al_2O_3, and its soda content will be combined with silica and alumina into an insoluble compound. This substance also has the property of adsorbing certain gases and certain vapors other than water vapor—a property which is sometimes useful in air conditioning installations. It is available commercially in granules ranging from a fine powder to pieces approximately 1.5 in. in diameter. It has high adsorptive capacity per unit of weight, and is non-toxic. It may be repeatedly re-activated after becoming saturated with adsorbed moisture without practical loss of its adsorptive ability. In the grade frequently used for air drying the re-activation may be accomplished at temperatures under 350 F. Specific gravity is 3.25 and the pores are reported to occupy 58 per cent of the volume of each particle. For most estimating purposes the volume-weight relation on a dry basis may be taken as 50 lb per cubic foot although in the smaller sizes the packed weight may be as much as 64 lb per cubic foot.

Silicon Dioxide, (Silica), in a special form obtained by suitably mixing sulphuric acid with sodium silicate, is another solid adsorbent and is commonly called *silica gel*. Its capillary structure is exceedingly small, so small that its exact structure has to be deduced rather than observed. The gel is available commercially in a wide variety of sizes of granules ranging from 4 to 300 mesh. It has high adsorptive capacity per unit of weight and is non-toxic, may be repeatedly re-activated without practical deterioration. Re-activation may be accomplished at temperatures of air up to 600 F although it is frequently accomplished with air or other gases at temperatures not over 350 F. Volume of the capillary pores is reported to be from 50 to 70 per cent of the total solid volume. For most estimating purposes the volume-weight relation can be assumed as from 38 to 40 lb per cubic foot on a dry basis.

Other substances having properties which make them available as

TABLE 6. PROPERTIES OF WATER

Sat Temp F	Abs. Press Lb per Sq In	Volume		Heat Content and Entropy Taken From +32 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct	Entropy	Ht Ct	Entropy
32	0.0887	0 01602	3296.0	0.00	1073.0	0.0000	2.1826	1096.9	2.2277	1120.8	2.2688
35	0.1000	0 01602	2941.0	3.02	1074.4	0.0062	2.1724	1098.3	2.2172	1122.2	2.2581
40	0.1217	0 01602	2441.0	8.05	1076.8	0.0163	2.1555	1100.6	2.2000	1124.5	2.2406
45	0.1475	0.01602	2034.0	13.07	1079.2	0.0262	2 1390	1102.9	2.1832	1126 7	2.2234
50	0.1780	0.01602	1702.0	18.08	1081.5	0.0361	2.1230	1105.2	2.1667	1129.0	2.2066
55	0.2140	0 01603	1430.0	23.08	1083.9	0 0459	2.1073	1107.5	2.1506	1131.3	2.1902
60	0.2561	0.01603	1206 0	28.08	1086.2	0 0556	2.0920	1109.8	2.1349	1133.5	2.1742
65	0.3054	0.01604	1021.0	33.08	1088.6	0 0652	2.0771	1112.2	2.1196	1135.8	2.1585
70	0.3628	0.01605	868 0	38.07	1090.9	0.0746	2.0625	1114.5	2.1046	1138.1	2.1432
75	0.4295	0.01606	740.0	43.06	1093.2	0.0840	2.0483	1116.7	2.0900	1140.3	2.1283
80	0.507	0.01607	632.9	48 05	1095.5	0.0933	2.0344	1119.0	2.0758	1142.5	2.1138
85	0.596	0.01609	543.3	53.04	1097.8	0.1025	2.0208	1121.2	2.0619	1144.7	2.0996
90	0.698	0.01610	467.9	58.03	1100.0	0.1116	2.0075	1123.4	2.0483	1146.8	2.0857
95	0.815	0 01612	404.2	63.01	1102.3	0.1206	1.9946	1125.6	2.0350	1148.9	2.0721
100	0.949	0 01613	350.3	68.00	1104.6	0.1296	1.9819	1127.9	2.0220	1151.1	2.0588
105	1.101	0.01615	304 4	72.98	1106 8	0.1384	1.9695	1130.2	2.0093	1153.2	2.0458

For properties of steam at high temperatures, see Page 29

solid adsorbents include lamisilate and charcoal but details of their physical properties are not available.

Nature of Adsorption Process

Adsorption is accomplished without chemical change between the air and the adsorbent substance. The adsorbent does not go into solution but water vapor is extracted from the air-vapor stream passing through the bed of adsorbent material and is caught and retained in the capillary pores. The exact nature of the process which goes on during adsorption is not known but it is stated that the action is brought about by surface condensation, and also by a difference between the vapor pressure of the water condensing inside the pores and the partial pressure of the water vapor in the air-vapor mixture. The adsorbing process in the bed can continue until the vapor pressures come into equilibrium. The amount of vapor adsorbed will depend on the adsorbent substances being used but for any single substance the amount depends on the temperature of the bed as well as on the partial pressure of the air-vapor mixture being passed over it.

As the process of adsorption goes on heat is liberated in the bed. The heat so liberated is the latent heat of the water vapor condensed together with the so-called heat of wetting. For a pound of water vapor at 60 F the latent heat released by condensation is approximately 1057 Btu. The heat of wetting for silica gel, for example, is about 200 Btu, making a total heat of adsorption of approximately 1257 Btu per pound of water adsorbed from the air-vapor mixture passing through the silica gel bed. The heat of wetting varies with the substance being used as the adsorbent while the latent heat of condensation depends only on the temperature and pressure of the water vapor.

Temperature-Pressure-Concentration Relations

Since the adsorptive ability of an adsorbent depends on the temperature of the bed and on the partial pressure difference between the pores and the air-vapor mixture it is important to know the pressures and temperatures at which pressure equilibrium is reached.

Evidently the equilibrium conditions represent the limits beyond which adsorption of vapor cannot continue. The relationship can be shown graphically and Fig. 1 is such a chart for silica gel. Charts of like nature can be plotted for other adsorbent materials.

Fig. 1 shows the equilibrium conditions for a gel bed maintained at constant temperature while the water vapor adsorption is allowed to continue until pressure equilibrium is reached. Each curve on the chart shows a certain dew-point temperature, and therefore a certain pressure, of the saturated water vapor.

As an example in the interpretation of the chart consider the case when moist air at a temperature of 80 F and a partial vapor pressure of 0.5 in. of mercury flows through a bed of silica gel which is at a temperature of 80 F. The chart indicates that the equilibrium of pressure between the air-vapor mixture and the bed is reached when the dry bed has adsorbed moisture to the extent of 31 per cent of the weight when dry. When this happens the bed can adsorb no more moisture unless its temperature is changed.

In practice however the adsorbent bed is seldom held at a steady temperature in air conditioning applications and neither is the adsorption process permitted to continue until moisture equilibrium is reached. Instead, the bed temperature varies and the bed is re-activated before equilibrium is approached. While charts of this kind can show the limiting properties of the substances they are seldom directly applicable to the solution of air conditioning problems unless considerable additional

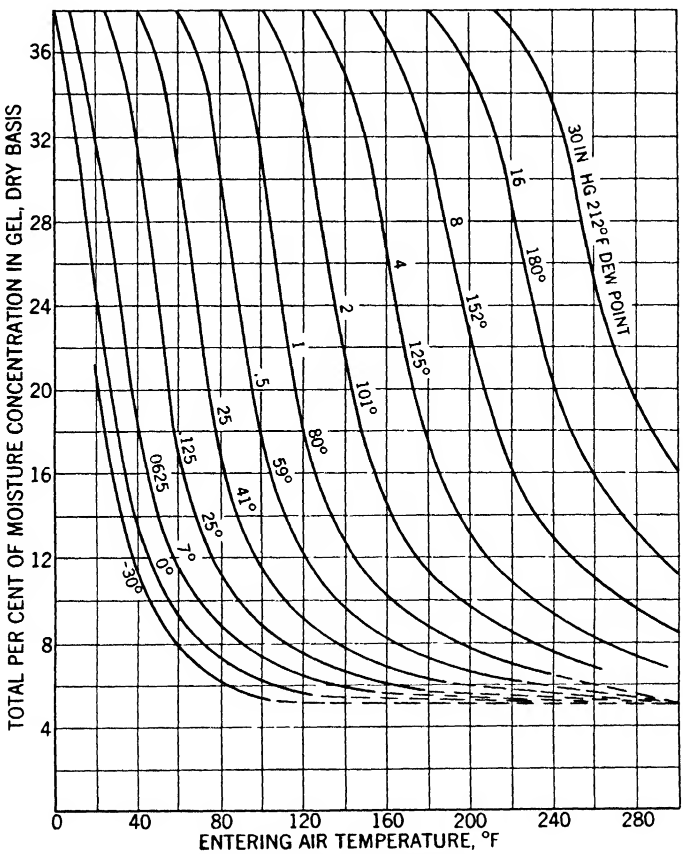

FIG. 1 TEMPERATURE—VAPOR PRESSURE—CONCENTRATION RELATION FOR A SILICA GEL BED AT CONSTANT TEMPERATURE

information is available. This takes the form of performance data covering the characteristics of the equipment in which the adsorbent bed is placed. Such performance data are beyond the scope of this chapter.

Liquid Absorbents

Any absorbent substance may be used as an air drying agent if it has a vapor pressure lower than the vapor pressure in the air-vapor mixture from which the moisture is to be removed. Absorbents are characteristically water solutions of materials in which the vapor pressure is

TABLE 7. DEW-POINT OF AIR IN EQUILIBRIUM WITH LITHIUM CHLORIDE SOLUTIONS
CONCENTRATION IN GRAM MOLS OF LITHIUM CHLORIDE PER 1000 GRAINS WATER

DEW-POINT AT ZERO CONC	CONCENTRATION OF LITHIUM CHLORIDE														
	2 0	4 0	6 0	8 0	10 0	12 0	14 0	16 0	18 0	20 0	22 0	24 0	26 0	28 0	30 0
320	315.2	308.7	299.9	290.2	279.7	269.4	259.6	251.5	244.1	236.5	230.0	223.8	218.6	214.5	210.3
300	295.4	289.1	280.5	270.9	260.6	250.5	240.8	232.6	225.4	218.0	211.8	205.8	200.8	196.9	192.8
280	275.6	269.5	261.1	251.7	241.5	231.6	222.2	214.0	206.7	199 7	193.5	187.8	183.2	179.3	175.2
260	255.8	250.0	241.9	232.6	222.7	212.8	203.5	195.5	188.4	181.7	175.4	170.0	165.6	162.0	158.4
240	236.0	230.4	222.5	213.5	203.8	194.2	185.0	177.1	170.0	163.6	157.5	152.2	148.3	144.6	140.5
220	216.2	210.8	203.2	194.4	184.9	175 5	166.4	158.6	151.6	145.3	139.6	134 6	130.7	127 3	124 2
200	196.4	191.2	183.9	175.4	166.1	156.7	148.0	140.3	133.5	127.3	121.9	117.0	113 3	110 1	
180	176.6	171.6	164.7	146.4	147.3	138.1	129.6	122.1	115.5	109.4	104.2	99.6	96 0		
160	156.8	152.1	145.4	137.4	128.6	119.7	111.3	103.9	97.4	91 6	86.6	82.2			
140	137.0	132.6	126.1	118.4	109.9	101.3	93 1	85.9	79.5	73.8	69.0				
120	117.2	113.0	106.8	99 4	91.1	82.7	74.7	67.8	61.5	56 0					
110	107.3	103.2	97.2	89.9	81.9	73.5	65.6	58.8	52.6	47 1					
100	97.4	93.4	87.5	80.5	72.7	64.4	56.6	49.8	43.7	38 2					
90	87.5	83.6	77.9	71.0	63.3	54.2	47.6	40.8	34.8	29.3					
80	77.6	73.8	68.4	61.6	54.0	46.1	38.5	31.8	25.9	20 6					
70	67.7	64.0	58.7	52.2	44.8	37.0	29.5	22 9	17.2	12.0					
60	57.8	54.3	49.1	42.7	35.5	27.9	20 5	14.0	8.3						
40	38.0	34.7	29 9	23 9	16.9	9 6	2 4	−3 9							
20		15.1	10.7	5.0	−1 7	−8.7	−15 4								
0		−4 5	−8.6	−13.9	−20 2	−27.0	−33.3								

reduced to a suitable level by governing the concentration of the solution. In addition to having a suitable low vapor pressure, a practical absorbent must also be widely available at economical cost, be non-corrosive, odorless, non-toxic, chemically inert against any impurities in the air stream, stable over the range of use and especially it must not precipitate out at the lowest temperature to which the apparatus is exposed. It must have low viscosity and be capable of being economically regenerated or concentrated after having been diluted by absorbing moisture.

TABLE 8. DENSITY OF LITHIUM CHLORIDE SOLUTIONS

CONCENTRATION MOLS *LiCl* PER 1000 GRAINS WATER	TEMPERATURE DEG F						
	0	50	100	150	200	250	300
0							
2		1 045	1 037	1 026	1 012		
4	1 090	1 085	1 076	1 064	1 052		
6	1 124	1 119	1 111	1 100	1 087		
8	1 156	1 150	1 143	1 132	1 122		
10	1 188	1 181	1 172	1 162	1 152	1 142	
12	1 217	1 209	1 199	1 188	1 178	1 168	
14	1 242	1 235	1 225	1 214	1 203	1 192	
16		1 257	1 248	1 236	1 226	1 215	
18		1 279	1 270	1 259	1 248	1 237	
20			1 291	1 280	1 279	1 568	
22				1 310	1 289	1 278	1 267
24				1 317	1 307	1 296	1 286
26					1 313	1 312	1 302
28					1 338	1 327	1 318
30						1 34	1 33
32							1.35

TABLE 9. VISCOSITY OF LITHIUM CHLORIDE SOLUTIONS (MILLIPOISE)

TEMP DEG F	CONCENTRATION IN MOLAL												
	0	2	4	6	8	10	12	14	16	18	20	22	24
0			56 75	72.44	97.05	136.8	199.5						
20		28.91	37.07	47.42	63.09	84.94	123.3	178 6					
40	15.45	19.91	25.53	32.58	43 05	58.48	81.10	116.1	165.6				
60	11.02	14.26	18.37	23.55	30.90	41.40	56 62	79.80	111.2	156.3			
80	8.61	11.19	14.42	18.62	24.32	32.28	43 45	60.26	82.04	113.8			
100	6 82	8.89	11.48	14.94	19.36	25 59	33 96	46.13	61.52	84.72	118.3		
120	5.60	7.31	9.51	12.30	15 92	20 99	27.67	36 64	48.31	65.77	89.95		
140	4.70	6 15	8.07	10.42	13.43	17 66	22 96	30 06	38 99	52.48	71.12	95.94	
160	4 01	5.25	6.92	8.93	11 51	15 00	19.36	25.06	32.14	42.76	56.89	75.86	106.2
180	3.48	4 56	6.01	7.78	10.00	12.91	16 56	21.28	27.10	35.48	46.45	60.67	84.33
200	3.05	4 01	5.28	6 86	8 79	11.22	14.32	18.28	23 12	29.92	38.55	50.70	67.92
220	2.72	3.58	4.72	6.14	7 83	9.93	12 59	16 00	20.14	25 64	32.96	43 05	56 49
240	2.43	3.21	4.25	5.50	7.02	8.83	11 12	14.09	17.62	22.18	28.31	36 98	47.42
260	2.19	2.90	3.84	4.94	6 46	7.91	9 91	12.47	15.50	19.36	24 60	31 92	40.55
280	2 00	2 66	3.52	4.51	5 75	7 19	8 97	11.27	14.00	17.22	21.78	28 05	35.56
300	1.86	2.48	3.28	4.17	5.32	6.67	8.28	10.38	12.82	15.70	19 68	25.12	31.92
320	1.74	2.32	3.08	3.89	4.94	6 19	7.73	9 64	11.86	14.45	18.03	22.80	29.11

Water solutions, or brines, of the chlorides of various inorganic elements such as calcium chloride and lithium chloride are the absorbents most frequently used in connection with air conditioning applications and detailed attention is confined to these two in this chapter.

Nature of Absorption Process

The application consists of bringing the air-vapor stream into intimate contact with the absorbent, permissably by passing the air stream through a finely divided spray of the brine but more generally by passing the air over a metal surface coil where the liquid absorbent presents a large surface to the air stream. The difference in vapor pressure causes some

TABLE 10 PROPERTIES OF LITHIUM CHLORIDE SOLUTIONS

CONCENTRATION MOLS (42 4 GRAMS *LiCl* PER 1000 GRAMS WATER)	PARTIAL HEAT OF MIXING AT 0 F BTU PER LB	TEMPERATURE COEF OF PARTIAL HEAT OF MIXING BTU PER LB PER F	SPECIFIC HEAT AT 70 F	BOILING POINT F (AT 760 MM HG)	FREEZING POINT	SUBSTANCE THAT FIRST SEPARATES OUT ON FREEZING
0	0 0	0 0	0 998	212 0	32	Ice
2	2 04	−0 014	0 901	215 8	16 3	Ice
4	7 24	−0 036	0 831	221 5	−5 8	Ice
6	16 7	−0 069	0 778	228 9	−34 2	Ice
8	31 9	−0 109	0 739	238 1	−69	Ice
10	51 1	−0 143	0 710	248 4	−90	Ice
12	75 7	−0 160	0 687	258 8	−40	$LiCl\text{-}3H_2O$
14	90 8	−0 167	0 666	268 9	1	$LiCl\text{-}3H_2O$
16	124 8	−0 176	0 647	277 9	36 5	$LiCl\text{-}2H_2O$
18	145	−0 186	0 631	285 8	58 1	$LiCl\text{-}2H_2O$
20	162	−0 194	0 617	293 2	86 4	$LiCl\text{-}H_2O$
22	171	−0 20	0 604	300 2	133	$LiCl\text{-}H_2O$
24	177	−0 20	0 59	307	156	$LiCl\text{-}H_2O$
26	182	−0 21	0 58	313	180	$LiCl\text{-}H_2O$
28	191	−0 21	0 575	318	190	$LiCl\text{-}H_2O$
30	194	−0 21	0 57	323	195	$LiCl\text{-}H_2O$
32	198	−0 22	0 56	328	280	$LiCl$

of the vapor in the air-vapor mixture to migrate into the brine. Here it condenses into liquid water and decreases the concentration of the absorbent. In order that the process be continuous means must be provided for counteracting the diluting effect of the extracted moisture and also for maintaining the temperature of the brine sufficiently low to hold the desired vapor pressure.

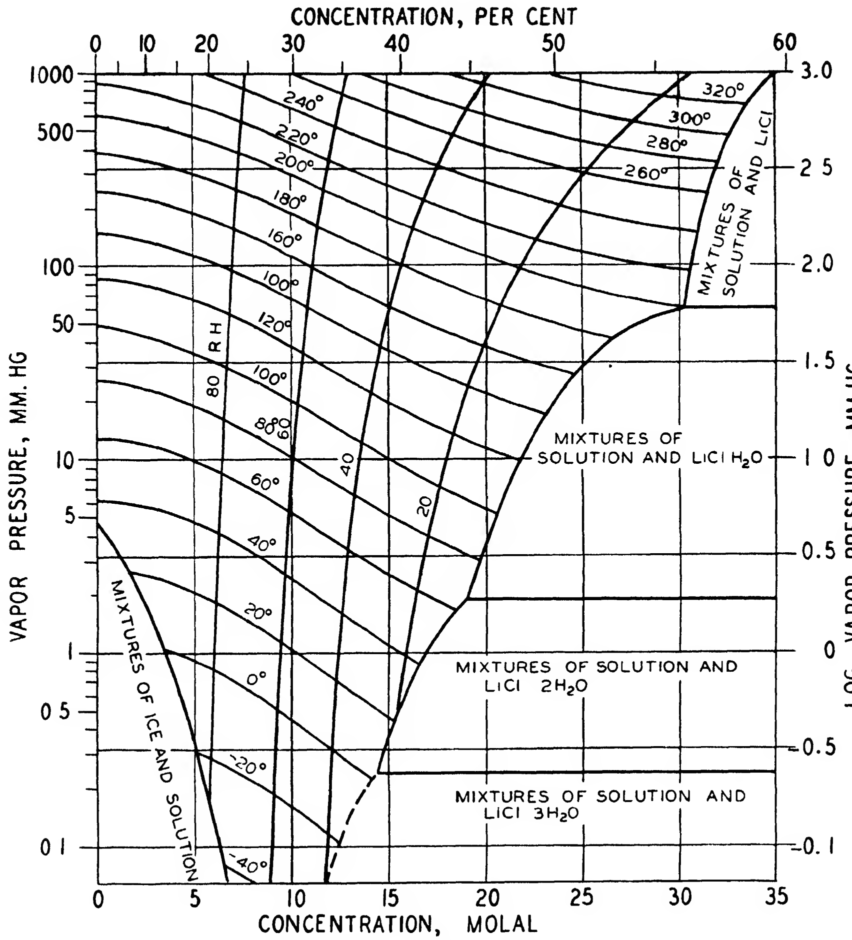

FIG. 2. TEMPERATURE—PRESSURE—CONCENTRATIONS FOR LITHIUM CHLORIDE

As the water vapor is added to the absorbent and condenses, it gives up its latent heat of condensation which tends to raise the temperature of both the absorbent and the moist air stream. For every pound of water absorbed and condensed the heat added to the air stream and the brine combined is obtainable from steam tables. For instance, at 60 F the amount of this heat is about 1057 Btu. In addition to this heat there is involved also the so-called heat of mixing which is frequently considerable.

Temperature-Pressure-Concentration Relations

Since the absorption process can continue only as long as there is a difference in vapor pressure between the absorbent and the air-vapor mixture and since at a given temperature of the absorbent the vapor pressure depends on the concentration of the solution, evidently there must be a relation between these quantities which if known would state the limits of the process. The relationship would also depend on the absorbent being used, and would have to be determined for each substance used as an absorbent. Fig. 2 shows this relationship graphically for lithium chloride. It will be noted that this chart is essentially similar to that shown in Fig. 1 and its direct usefulness is limited by much the same considerations.

In order to permit numerical calculations of air conditioning problems it is desirable to have tables for use instead of a chart like Fig. 2, and Tables 7, 8, 9 and 10 can be used in making calculations for lithium chloride.

Instead of tabulating the vapor pressure of the solution of lithium chloride it is preferable to tabulate the dew-point of air in equilibrium with lithium chloride, since it is easy to interpolate between values of the dew-point and not so easy to interpolate accurately between values of vapor pressure. The values for dew-point may be converted to vapor pressures, relative humidity, and wet-bulb of air in equilibrium by means of the usual psychrometric chart or formula.

In Tables 7, 8, 9 and 10 the unit of concentration is the *mol*. A 'M' molal solution is defined as a solution containing $M \times 42.37$ grains of anhydrous lithium chloride per 1000 grains of water. The formula connecting concentration in mols with weight in per cent is equivalent to: $[100 \times M \times 42.37] \div [1000 + (M \times 42.37)]$.

PROBLEMS IN PRACTICE

1 ● What is the heat content above −40 F of 2.5 lb of ammonia when under a pressure of 92.9 lb per square inch gage and a temperature of 160 F?

First determine the condition of the ammonia at the stated temperature and pressure. Do this by finding the absolute pressure which in this case is 92 9 lb gage plus 14.7 or 107 6 lb per square inch absolute. From Table 1 note that the saturation temperature at this pressure is 60 F. Therefore, the ammonia is superheated 100 F, and the total heat per pound can be read directly from the Table as 689 9 Btu In the 2 5 lb of ammonia there are 2 5 × 689 9, or 1724.75 Btu.

2 ● What volume is necessary to accommodate 0.27 lb of saturated F_{12} vapor when compressed to 99.6 lb gage?

The absolute pressure is 99 6 plus 14 7 or 114 3 lb. In Table 2 find that one pound of F_{12} vapor saturated occupies 0 368 cu ft. Then the 0 27 lb would occupy 0 27 × 0.368, or 0.099 cu ft.

3 ● How much heat would be removed from air passing over a coil through which 2 lb of methyl chloride per minute is forced? The coil is under a gage pressure of 64 lb per square inch and the liquid refrigerant is completely vaporized in passing through the coil.

Find that the absolute pressure is 64 plus 14.7 or 78.7 lb per sq in From Table 3 note that the saturation temperature at this pressure is 75 F (nearly) and that the heat

content per pound of the vapor is 207.8 Btu. Also that the heat content of the liquid is 42.2 Btu per pound. Subtract 42.2 from 207.8 and 165.6 Btu per pound is the heat necessary to change the liquid refrigerant to a vapor (latent heat). As the heat to accomplish this change comes from the air around the coil, the heat removed from the air is 165.6 Btu per pound of methyl chloride evaporated in the coil. When the refrigerant is circulated at 2 lb per minute, 2 × 165.6, or 331.2 Btu per minute are removed from the air, or refrigerating effect is produced at the rate of 331.2 ÷ 200, or 1.65 tons.

4 ● Calculate the dew-point, wet-bulb, relative humidity and absolute humidity of air in equilibrium at 100 F with pure lithium chloride solution of density 1.270.

From Table 8 the concentration of a solution of density 1.270 at 100 F is 18.0 *M*. From Table 7 the dew-point of 18 *M* lithium chloride at 100 F is 43 7 F. From Table 6, Chapter 1, the partial pressure of water over the solution is 0.2858 in. of Hg, the absolute humidity is 42.00 grains per pound dry air, and the wet-bulb is 65 8 F. The relative humidity is 14 0 per cent.

5 ● Calculate the boiling point, and freezing point of 18 *M* lithium chloride solutions.

From Table 10, boiling point (standard) is 285.8 F, freezing point is 58.1 F. The salt precipitated on cooling to this temperature has the composition $LiCl \cdot 2H_2O$.

6 ● Calculate the heat of vaporization of 1 lb of water from a large amount of lithium chloride solution at the boiling point.

The heat of boiling is equal to the heat of mixing plus the heat of boiling pure water at the same temperature. The heat of mixing from Table 10 at 18 *M* and 285 8 F is (145 − 0.186 × 285.8) = 92 Btu per pound. The heat of vaporization of water from steam tables at 285.8 F is 920 Btu per pound. Therefore the heat of vaporization of water from the solution is 920 + 92 = 1012 Btu per pound.

7 ● One thousand pounds of air per minute at 100 F dry-bulb with a dew-point of 70 F and a relative humidity of 39 per cent is passed over 18 *M* lithium chloride solution. The rate of flow of the solution is 200 gpm and the entering temperature is 80 F. The air leaves the absorber at 85 F dry-bulb and dew-point of 35 F. Calculate (*a*) the heat to be removed from the lithium chloride solution to maintain these conditions, and (*b*) the temperature rise of the solution in passing through the absorber.

a. The heat content of the entering air: From Table 6, Chapter 1, weight of vapor at 70 F dew-point is 0.01574 lb times heat content of steam at 100 F dry-bulb is 1104.2 (Table 8, Chapter 1) equals 17.41 Btu per pound plus heat content of dry air at 100 F is 24.0 (Table 6, Chapter 1) resulting in heat content of mixture as 41.41 Btu per pound.

Similarly, the heat content of the leaving air: Weight of vapor at 35 F dew-point is 0.004262 × 1097.5 = 4.68 Btu per pound plus heat content of dry air at 85 F is 20.39 resulting in heat content of mixture as 25.07 Btu per pound.

Heat to be extracted from air is 1000 × (41.41 − 25.07) = 16,340 Btu per minute. Add to this the heat of mixing of 18*M* lithium chloride at 80 F equals 145 − (0.186 × 80) = 130 Btu per pound (Table 10) or for 1000 lb of air × (0.01574 − 0.00426) × 130 = 1494 Btu per minute. Heat to be removed from solution is 16,340 + 1494 = 17,834 Btu per minute.

b. The weight of solution circulated is 200 × 1.275 (Table 8) × 8.33 = 2124 lb per minute. Its heat capacity is 2124 × 0.631 (Table 10) = 1340 Btu per minute per degree Fahrenheit. The temperature rise is 17,834 ÷ 1340 = 13.31 F.

Chapter 3

PHYSICAL AND PHYSIOLOGICAL PRINCIPLES OF AIR CONDITIONING

Vitiation of Air, Heat Regulation in Man, Effects of Heat, Effects of Cold and Temperature Changes, Acclimatization, Effective Temperature Index of Warmth, Optimum Air Conditions, Winter and Summer Comfort Zone, Optimum Humidity, Air Quality and Quantity, Air Movement and Distribution, Natural and Mechanical Ventilation, Heat and Moisture Losses, Ultra-Violet Radiation and Ionization, Recirculation and Ozone, Ventilation Standards

VENTILATION is defined in part as "the process of supplying or removing air by natural or mechanical means to or from any space." (See Chapter 45). The word in itself implies quantity but not necessarily quality. From the standpoint of comfort and health, however, the problem is now considered to be one of securing air of the proper quality rather than of supplying a given quantity.

The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. More particularly, it includes the simultaneous control of temperature, humidity, movement, and purity of the air. The term is broad enough to embrace whatever other additional factors may be found desirable for maintaining the atmosphere of occupied spaces at a condition best suited to the physiological requirements of the human body.

VITIATION OF AIR

Under the artificial conditions of indoor life, the air undergoes certain physical and chemical changes which are brought about by the occupants themselves. The oxygen content is somewhat reduced, and the carbon dioxide slightly increased by the respiratory processes. Organic matter, which is usually perceived as odors, comes from the nose, mouth, skin and clothing. The temperature of the air is increased by the metabolic processes, and the humidity raised by the moisture emitted from the skin and lungs. There is also a marked decrease in both positive and negative ions in the air of occupied rooms but the significance of this factor is still questionable[1].

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are much too small even under the worst conditions. The amount of carbon dioxide in air is

[1]A S.H V E RESEARCH REPORT No 921—Changes in Ionic Content in Occupied Rooms Ventilated by Natural and Mechanical Methods, by C P Yaglou, L C Benjamin and S P Choate (A S H V.E. TRANSACTIONS, Vol 38, 1932, p. 191)

often used in ventilation work as an index of odors of human origin, but the information it affords rarely justifies the labor involved in making the observation[2, 3]. Little is known of the identity and physiological effects of the organic matter given off in the process of respiration. The former belief that the discomfort experienced in confined spaces was due to some toxic volatile matter in the expired air is now limited, in the light of numerous researches, to the much less dogmatic view that the presence of such a substance has not been demonstrated. The only certain fact is that expired and transpired air is odorous and offensive, and it is capable of producing loss of appetite and a disinclination for physical activity. These reasons, whether esthetic or physiological, call for the introduction of a certain minimum amount of clean outdoor air to dilute the odoriferous matter to a concentration which is not objectionable.

A certain part of the dissemination of disease in confined spaces is caused by the emission of pathogenic bacteria from infected persons. Droplets sprayed into the air in talking, coughing, sneezing, etc., do not all fall immediately to the ground within a few feet from the source, as it was formerly believed. The large droplets do, of course, but minute droplets less than 0.1 mm in diameter evaporate to dryness before they fall the height of a man. Nuclear residues from such sources, which may contain infective organisms drift long distances with the air currents and the virus may remain alive long enough to be transmitted to other persons in the same room or building. Wells[4] recovered droplet nuclei from cultures of resistant micro-organisms a week after inoculation into a tight chamber of 300 cu ft capacity. Typical organisms of infections of the upper respiratory tract (pneumococcus type I. B. diphtheriae, Streptococcus hemolyticus, and Streptococcus viridans) were found to die out quite soon when exposed to light and air, and could be recovered from the air in small numbers only 48 hours after inoculation. Organisms typical of the intestinal tract (B. coli, B. typhosus, B. paratyphosus, A. and B. dysenteriae) were not recovered 12 hours after inoculation.

The significant factors in infection are believed to be the numbers of infective organisms encountered, the frequency of exposure, and the resistance of the individual including the degree of acquired immunity. The probability of encountering a sufficient number of organisms to break down the natural body defense is related to the air space per person and the quantity of clean air supplied. Except in badly ventilated rooms, the danger is believed to be "much contracted in space, limited in time and restricted to comparatively few diseases."[5]

Practical possibilities in sterilizing air supplies by the use of ultra-violet light are now being studied[6].

The primary factors in air conditioning work, in the absence of any specific contaminating source, are temperature, radiation, drafts and

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[2]A S H V E RESEARCH REPORT No 959—Indices of Air Change and Air Distribution, by F C Houghten and J L Blackshaw (A S H V E TRANSACTIONS, Vol 39, 1933, p 261)

[3]A S H V E RESEARCH REPORT No 1031—Ventilation Requirements, by C P Yaglou, E C Riley and D I Coggins (A S H V E TRANSACTIONS, Vol 42, 1936, p 133)

[4]Air-Borne Infection and Sanitary Air Control, by W F Wells, (*Journal Industrial Hygiene*, November 1935)

[5]Preventive Medicine and Hygiene, by Milton J Rosenau (6th edition, pp 909-917, D Appleton-Century Co, N Y, 1935).

[6]Viability of B Coli Exposed to Ultra-Violet Radiation in Air, by W F Wells, and G M Fair (*Science*, 1935 82 p 280)

body odors. As compared with these physical factors, the chemical factors are, as a general rule, of secondary importance.

HEAT REGULATION IN MAN

The importance of the thermal factors arises from the profound influence which they exert upon body temperature, comfort and health. Body temperature depends on the balance between heat production and heat loss. The heat resulting from the combustion of food within the body maintains the body temperature well above that of the surrounding air. At the same time, heat is constantly lost from the body by radiation, conduction and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss. In healthy persons this takes place automatically by the action of the heat regulating mechanism.

According to the general view, special areas in the skin are sensitive to heat and cold. Nerve courses carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The mechanisms of adjustment are complex and little understood at the present time. Coordination of these different mechanisms seems to vary greatly with different air conditions.

With rising air temperatures up to 75 F or 80 F, metabolism, or internal heat production, decreases slightly[7], probably by an inhibitory action on heat producing organs, especially the adrenal glands, which seem to exert the major influence on basic combustion processes in the body. The blood capillaries in the skin become dilated by reflex action of the vasomotor nerves, allowing more blood to flow into the skin, and thus increase its temperature and consequently its heat loss. The increase in peripheral circulation is at the expense of the internal organs. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin, where it is evaporated. This method of cooling is the most effective of all, as long as the humidity of the air is sufficiently low to allow for evaporation. In high humidities, where the difference between the dew-point temperature of the air and body temperature is not sufficient to allow rapid evaporation, equally good results may be obtained by increasing the air movement, and hence the heat loss by conduction and evaporation.

In cold environments, in order to keep the body warm there is an actual increase in metabolism brought about partly by voluntary muscular contractions (shivering) and partly by an involuntary reflex upon the heat producing organs. The surface blood vessels become constricted, and the blood supply to the skin is curtailed by vasomotor shifts to the internal organs in order to conserve body heat. The sweat glands become inactive.

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[7]A S H V E RESEARCH REPORT No 830—Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F C Houghten, W W Teague, W E Miller and W P Yant (A S H V E TRANSACTIONS, Vol 35, 1929, p 245)

EFFECTS OF HEAT

Although the human organism is capable of adapting itself to variations in environmental conditions, its ability to maintain heat equilibrium is limited. The upper limit of effective temperature to which the human organism is capable of adapting itself without serious discomfort or injury to health is 90 deg ET for men at rest and between 80 and 90 deg ET for men at work depending upon the rate of work. Within these limits a new equilibrium is established at a higher body temperature level through a chain of physiological adjustments. The heat regulating center fails, when the external temperature is so abnormally high that bodily heat cannot be eliminated as fast as it is produced. Part of it is retained in the body, causing a rise in skin and deep tissue temperature, an increase in the heart rate, and accelerated respiration. (See Table 1). In extreme heat, the metabolic rate is markedly increased owing to the excessive rise in body temperature[8], and a vicious cycle results which may eventually lead to serious physiologic damage.

TABLE 1. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp	Actual Cheek Temp (Deg Fahr)	Men at Rest			Men at Work 90,000 ft-lb of Work per Hour			
		Rise in Rectal Temp (Deg Fahr per Hour)	Increase in Pulse Rate (Beats per Min per Hour)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt by Perspiration (Lb per Hr)
60	....	....	......	--- .	225,000	0.0	6	0.5
70	....	0.0	0	0 2	225,000	0 1	7	0.6
80	96.1	0.0	0	0 3	209,000	0.3	11	0.8
85	96 6	0 1	1	0 4	190,000	0 6	17	1.1
90	97.0	0 3	4	0 5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1 7	67,000	4.0	103[b]	2.7
105	104.7	4.0	83	2 7	49,000	6 0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b]	4.0[b]	37,000	8 5[b]	237[b]	4.4[b]

[a]Data by A S H V E Research Laboratory
[b]Computed value from exposures lasting less than one hour

Examples of this are met with in unusually hot summer weather and in hot industries where the radiant heat from hot objects renders heat loss from the body by radiation and convection impossible. Consequently, the workers depend entirely on evaporation for the elimination of body heat. They stream with perspiration and drink liquids abundantly to replace the loss.

One of the deleterious effects of high temperatures is that the blood is diverted from the internal organs to the surface capillaries, in order to serve in the process of cooling. This affects the stomach, heart, lungs and other vital organs, and it is suggested that the feeling of lassitude and discomfort experienced is due in part to the anæmic condition of the brain. The stomach loses some of its power to act upon the food, owing to a

[8]A.S H V E RESEARCH REPORT No 719—Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities, by W J McConnell, C P Yaglou and W B Fulton (A.S H V E TRANSACTIONS, Vol 31, 1925, p 123)

diminished secretion of gastric juice, and there is a corresponding loss in the antiseptic and antifermentive action which favors the growth of bacteria in the intestinal tract[9]. These are considered to be the potent factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather. The victim may lose appetite and suffer from indigestion, headache and general enervation, which may eventually lead to a premature old age.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, Moss[10] recommends the addition of 6 grams of sodium chloride and 4 grams of potassium chloride to a gallon of water.

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show clearly that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality rates increase progressively as the temperature rises.

EFFECTS OF COLD AND TEMPERATURE CHANGES

The action of cold on human beings is not well known. Cold affects the human organism in two ways: (1) through its action on the body as a whole, and (2) through its action on the mucous membranes of the upper respiratory tract. Little exact information is available on the latter.

On exposure to cold, the loss of heat is increased considerably and only within certain limits is compensation possible by increased heat production and decreased peripheral circulation. The rectal temperature often rises upon exposure to cold but the pulse rate and skin temperature fall. The blood pressure increases, owing to constriction in the peripheral vessels. Just how cold affects health is not well understood. It imposes an extra load upon the heat-producing organs to maintain body temperature. The strain falls largely upon digestion, metabolism, blood circulation, and the kidneys, and indirectly upon the nervous system[11].

Although the seasonal increase in morbidity and mortality sets in with the approach of cold weather and subsides in the warm summer months, little is known of the specific causative factors and their mechanism of action. Over-crowding of buildings, overheated rooms, lack of ventilation, and close personal contacts are frequently held responsible for our winter ills, but the evidence is not conclusive.

In extremely cold atmospheres compensation by increased metabolism becomes inadequate. The body temperature falls and the reflex irritability of the spinal cord is markedly affected. The organism may finally pass into an unconscious state which ends in death.

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[9]Influence of Effective Temperature upon Bactericidal Action of Gastro-Intestinal Tract, by Arnold and Brody (*Proceedings Society Exp Biol Med*, Vol 24, 1927, p. 832).

[10]Some Effects of High Air Temperatures upon the Miner, by K. N. Moss (*Transactions Institute of Mining Engineers*, Vol 66, 1924, p 284).

[11]Loc Cit Note 5

Cannon showed that excessive loss of heat is associated with increased activity of the adrenal medulla[12]. The extra output of adrenin hastens heat production which protects the organism against cooling. Bast[13] found a degeneration of thyroid and adrenal glands upon exposure to cold.

A moderate amount of variability in temperature is known to be beneficial to health, comfort, and the performance of physical and mental work. On the other hand, extreme changes in temperature, such as those experienced in passing from a warm room to the cold air out-of-doors, appear to be harmful to the tissues of the nose and throat which are the portals for the entry of respiratory diseases.

Experiments show that chilling causes a constriction of the blood vessels of the palate, tonsils, throat, and nasal mucosa, which is accompanied by a fall in the temperature of the tissues. On re-warming, the palate and throat do not always regain their normal temperature and blood supply. This anæmic condition favors bacterial activity and it probably plays a part in the disposition of common cold and other respiratory diseases. It is believed that the lowered resistance is due to a diminution in the number and phagocytic activity of the leucocytes (white blood cells) brought about by exposure to cold and by changes in temperature.

Sickness records in industries seem to strengthen this belief. The Industrial Fatigue Research Board of England[14] found that in workers exposed to high temperatures and to changes in temperature, namely, steel melters, puddlers, and general laborers, there is an excess of all sickness, the excess among the puddlers being due chiefly to respiratory diseases and rheumatism. The causative factor was not the heat itself but the sudden changes in temperature to which the workers were exposed. The tin-plate millmen who were not exposed to chills, since they work almost continuously throughout the shift, had no excess of rheumatism and respiratory diseases. On the other hand, the blast-furnacemen, who work mostly in the open, showed more respiratory sickness than the steel workers. This experience in British factories is well in accord with the findings in American industries[15, 16]. According to these data the highest pneumonia death rate is associated with dust, extreme heat, exposure to cold, and to sudden changes in temperature.

ACCLIMATIZATION

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

Some persons regard the unnecessary endurance of cold as a virtue.

[12]Studies on the Condition of Activity of Endocrine Glands, by W B Cannon, A Guerido, S W Britton and E M Bright (*American Journal of Physiology*, Vol 79, 1926, p 466)

[13]Studies in Exhaustion Due to Lack of Sleep, by T H Bast, J S Supernaw, B Lieberman and J Munro (*American Journal of Physiology*, Vol. 85, 1928, p 135)

[14]Fatigue and Efficiency in the Iron and Steel Industry, by H M Vernon (*Industrial Fatigue Research Board*, Report No 5, 1920, London)

[15]Iron Foundry Workers Show Highest Percentage of Deaths from Pneumonia (*Statistical Bulletin*, Metropolitan Life Insurance Company, 1928)

[16]The Pneumonia Problem in the Steel Industry, by D K Brundage and J J. Bloomfield, (*Journal of Industrial Hygiene*, 14, December, 1932)

They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of the present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and it interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature averaging 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. All this seems to indicate that adaptation to an environment averaging between 60 and 80 F is a very primitive trait[17].

Within these limits, however, there does occur a definite adaptation to external temperature level. People and animals raised under conditions of tropical moist heat have a lower rate of heat production than do those who grow up in cooler environments. This causes them to stand chilling poorly as they are unable to quickly increase internal combustion to keep up the body temperature. For this reason they have trouble standing the cold, stormy weather of the temperate zones, and when exposed to it are very susceptible to respiratory infections. Likewise, people living in cool climates suffer greatly in the moist heat of the tropics until their adrenal activity has slowed down. Within a couple of years, however, they find themselves standing the heat much better and disliking cold. They become acclimated by a definite change in the combustion level within the body[18].

In certain individuals the psychologic factor is more powerful than acclimatization. A fresh air fiend may suffer in a room with windows closed regardless of the quality of the air. As a matter of fact, instances are known in which paid subjects refused to stay in a windowless but properly conditioned experimental chamber because the atmosphere felt suffocating to them upon entering the room.

EFFECTIVE TEMPERATURE INDEX OF WARMTH

Sensations of warmth or cold depend, not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler.

On the other hand, in cold environments an increase in humidity produces a cooler sensation. The dividing line at which humidity has no effect upon warmth varies with the air velocity and is about 46 F (dry-bulb) for still air and about 51, 56 and 59 F for air velocities of 100, 300 and 500 fpm, respectively. Radiation from cold or warm surfaces is another important factor under certain conditions.

Combinations of temperature, humidity and air movement which induce the same feeling of warmth are called thermo-equivalent con-

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[17]Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1924

[18]Air Conditioning in its Relation to Human Welfare, by C A Mills (A S H V E TRANSACTIONS, Vol 40, 1934, p 289).

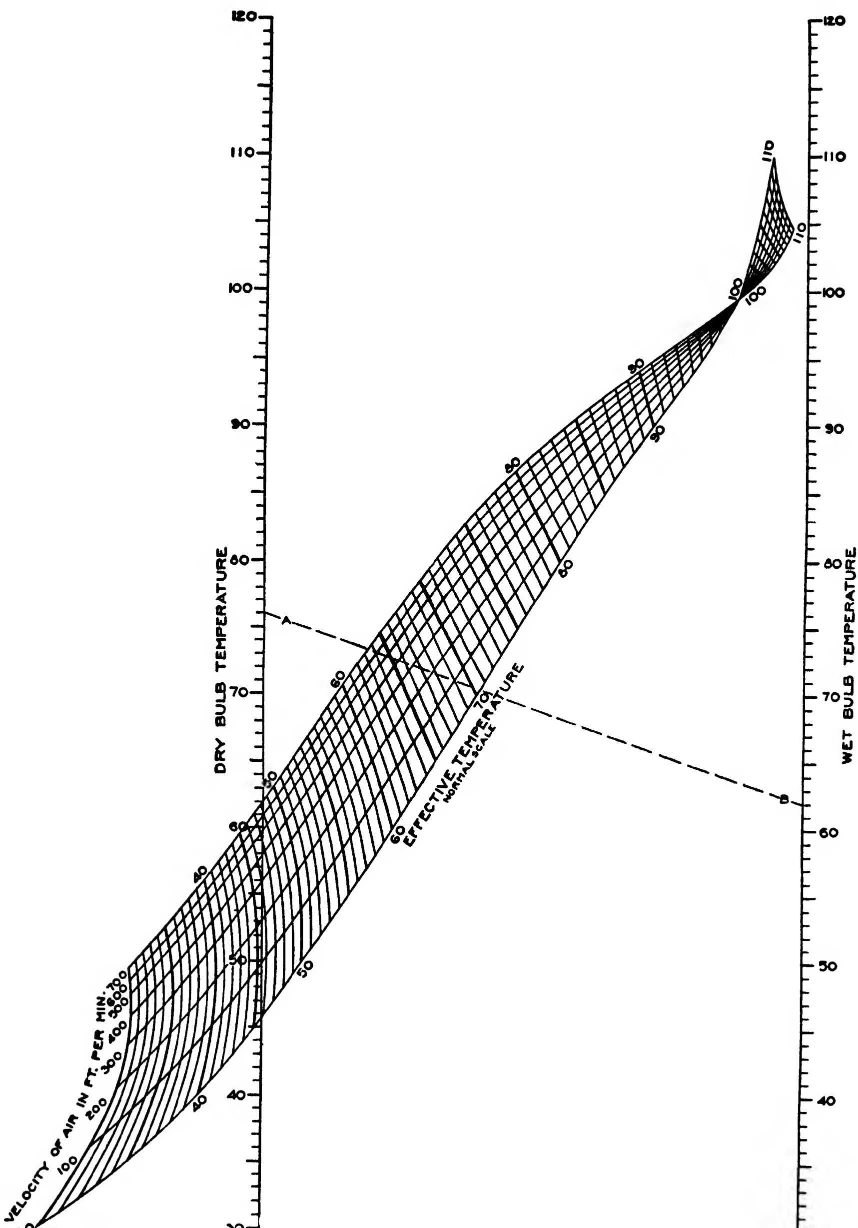

FIG. 1. EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE. APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS:

A Clothing. Customary indoor clothing. *B Activity:* Sedentary or light muscular work. *C. Heating Methods* Convection type, *i e.* warm air, direct steam or hot water radiators, plenum systems.

ditions. A series of tests[19, 20, 21, 22] at the A.S.H.V.E. Research Laboratory, Pittsburgh, established the equivalent conditions met with in general air conditioning work. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also determines the physiological *effects* on the body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

Effective temperature is not in itself an index of comfort, except under ordinary humidity conditions (30 to 60 per cent) when the individual is least conscious of humidity. Moist air at a comparatively low temperature, and dry air at a higher temperature may each feel as warm as air of an intermediate temperature and humidity, but the *comfort* experienced in the three air conditions would be different, although the effective temperature is the same.

Air of proper warmth may, for instance, contain excessive water vapor, and in this way interfere with the normal physiologic loss of moisture from the skin, leading to damp skin and clothing and producing more or less discomfort; or the air may be excessively dry, producing appreciable discomfort to the mucous membrane of the nose and to the skin which dries up and becomes chapped from too rapid loss of moisture.

The numerical value of the effective temperature index for any given air condition is fixed by the temperature of calm (15 to 25 fpm air movement) and saturated air which induces a sensation of warmth or cold like that of the given condition. Thus, any air condition has an effective temperature of 60 deg, for instance, when it induces a sensation of warmth like that experienced in calm air at 60 deg saturated with moisture. The effective temperature index cannot be measured directly but it is computed from the dry- and wet-bulb temperature and the velocity of air using charts (see Figs. 1 or 2, 3 and 4) or tables. The accuracy in estimating effective temperature is ± 0.5 F, because the human organism cannot perceive smaller temperature differences. Therefore, there is no justification in trying to read chart values closer than 0.5 F, as this implies fictitious accuracy.

The charts shown in Figs. 1, 2, 3 and 4 apply to average normal and healthy persons adapted to American living and working conditions. Application is limited to sedentary or light muscular activity, and to rooms heated by the usual American convection methods (warm air, central fan and direct hot water and steam heating systems) in which the difference between the air and wall surface temperatures may not be too

[19]A S H V E RESEARCH REPORT No 673—Determining Lines of Equal Comfort, by F C Houghten and C P Yaglou (A S H V E TRANSACTIONS, Vol 29, 1923, p. 361).

[20]A S H V E RESEARCH REPORT No 691—Cooling Effect on Human Beings by Various Air Velocities, by F C Houghten and C P Yaglou (A S H V E TRANSACTIONS, Vol 30, 1924, p 193).

[21]A S H V E RESEARCH REPORT No. 717—Effective Temperature with Clothing, by C P. Yaglou and W. E Miller (A S H.V E TRANSACTIONS, Vol 31, 1925, p 89)

[22]A S H V E RESEARCH REPORT No 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C Houghten, W W Teague and W E Miller (A S H V E. TRANSACTIONS, Vol 32, 1926, p 315)

great. The charts do not apply to rooms heated by radiant method such as British panel system, open coal fires and similar usages. They will probably not apply to races other than the white or perhaps to inhabitants of other countries where the living conditions, climate, heating methods, and clothing are materially different than those of the subjects employed in experiments at the Research Laboratory.

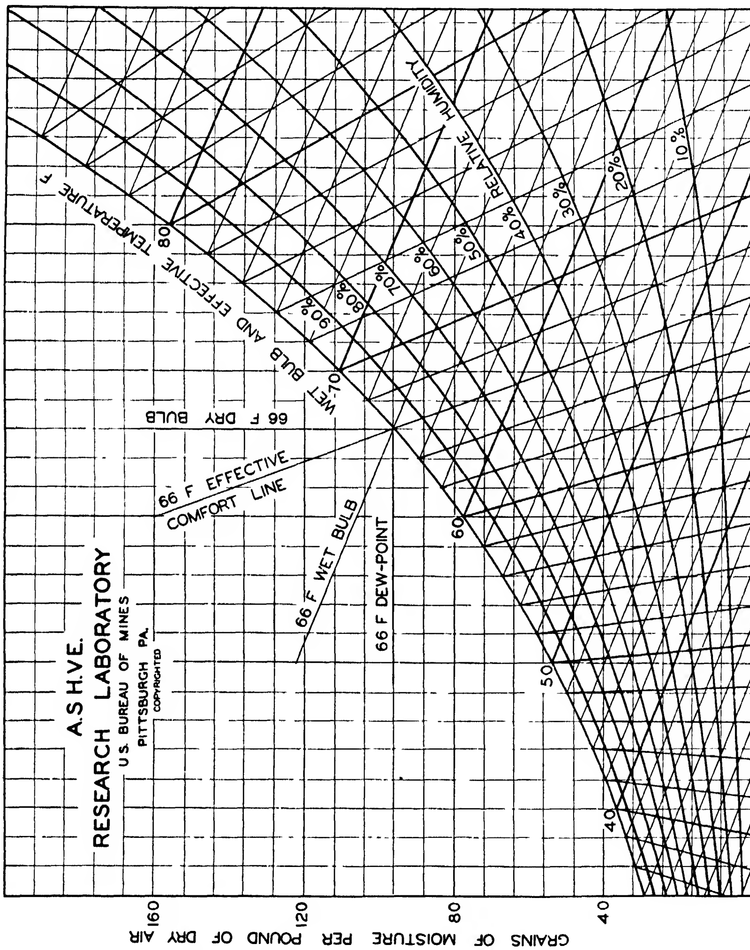

FIG 2. PSYCHROMETRIC CHART, PERSONS AT REST, NORMALLY CLOTHED, IN STILL AIR

In rooms in which the average wall surface temperature is considerably below or above air temperature, a correction must be applied to the readings of the dry-bulb thermometer to allow for such negative or positive radiation. In Fig. 5 is given the cooling effect of cold walls as determined at the A.S.H.V.E. Research Laboratory[23] by trained subjects

[23]A S H V E RESEARCH REPORT No 946—Cold Walls and Their Relation to the Feeling of Warmth, by F C Houghten and Paul McDermott (A S H V E TRANSACTIONS, Vol 39, 1933, p 83)

passing back and forth from a small experimental room having three cold walls, to a control room with walls and air at the same temperature.

It can be seen in Fig. 5 that with air and walls at 70 F in the control (warm wall room), the cooling effect of three cold walls at 55 F of the experimental room was 4 F. Therefore, for the same feeling of warmth,

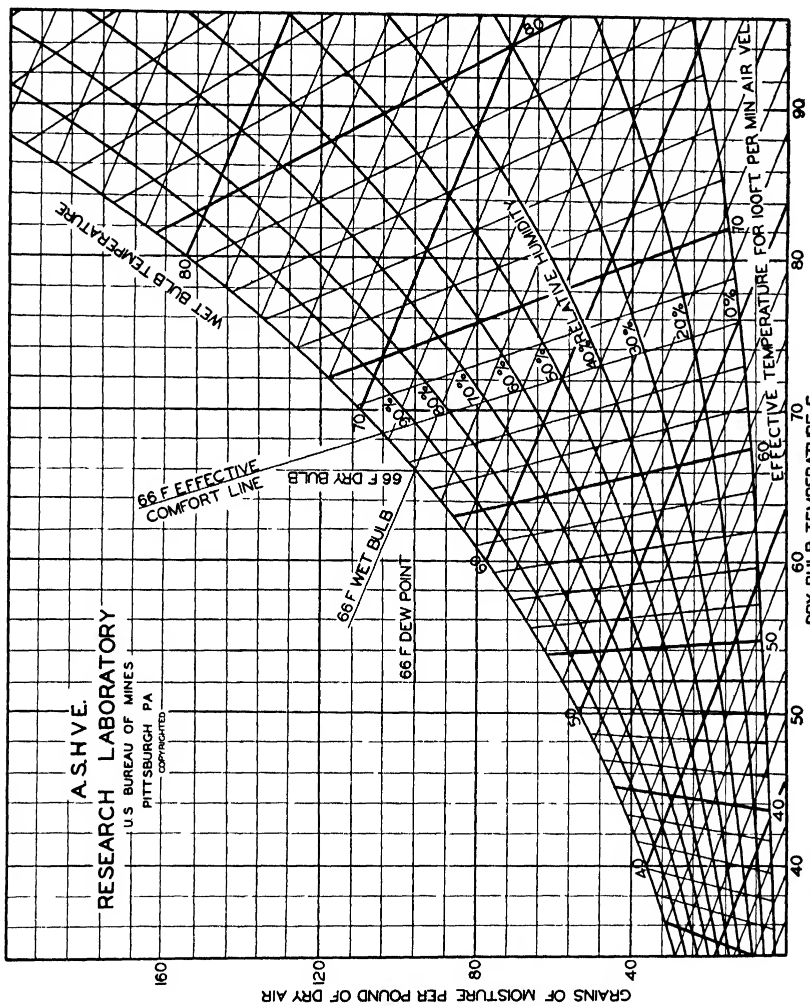

FIG. 3. PSYCHROMETRIC CHART, PERSONS NORMALLY CLOTHED AT REST, IN 100 FPM AIR VELOCITY

the temperature in the experimental room should be increased to 74 F. The reverse would hold in rooms with high-wall surface temperature; a lower air temperature would be required to compensate for positive radiations to the occupants.

OPTIMUM AIR CONDITIONS

No single comfort standard can be laid down which would meet every need. There is an inherent individual variation in the sensation of warmth or comfort felt by persons when exposed to an identical atmospheric condition. The state of health, age, sex, clothing, activity, and

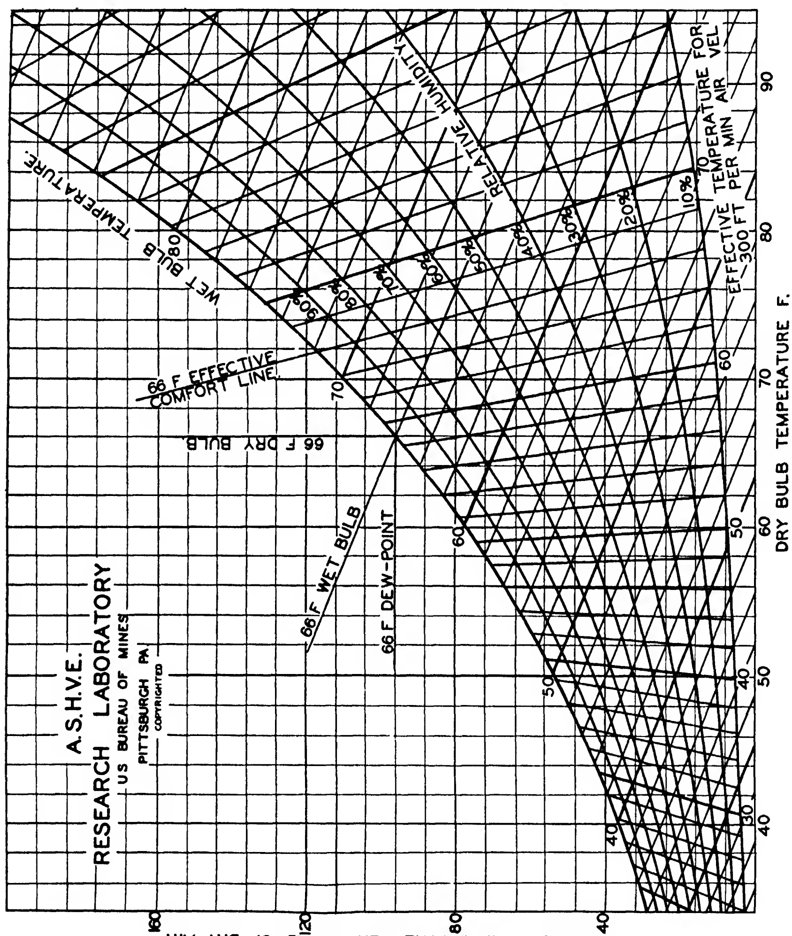

FIG. 4. PSYCHROMETRIC CHART, PERSONS NORMALLY CLOTHED AT REST, IN 300 FPM AIR VELOCITY

the degree of acquired adaptation seem to be the important factors affecting the comfort standards.

Since the prolonged effects of temperature, humidity and air movement on health are not known to the same extent as their effects on comfort, the optimum conditions for health may not be identical with those for comfort. On general physiologic grounds, however, the two do not differ greatly since this is in accordance with the efficient operation of the

heat regulating mechanism of the body. This belief is strengthened by results of studies on premature infants over a four-year period[24]. By adjusting the temperature and humidity so as to stabilize the body temperature of these infants, the incidence of diarrhœa and mortality was decreased, gains in body weight increased and infections were reduced to a minimum.

Winter Comfort Zone and Comfort Line

In Fig. 6 is shown the A.S.H.V.E. winter comfort zone which was determined experimentally with large groups of men and women subjects wearing customary indoor winter clothing. The extreme comfort zone includes conditions between 60 and 74 deg ET in which one or more of the experimental subjects were comfortable. The average comfort zone

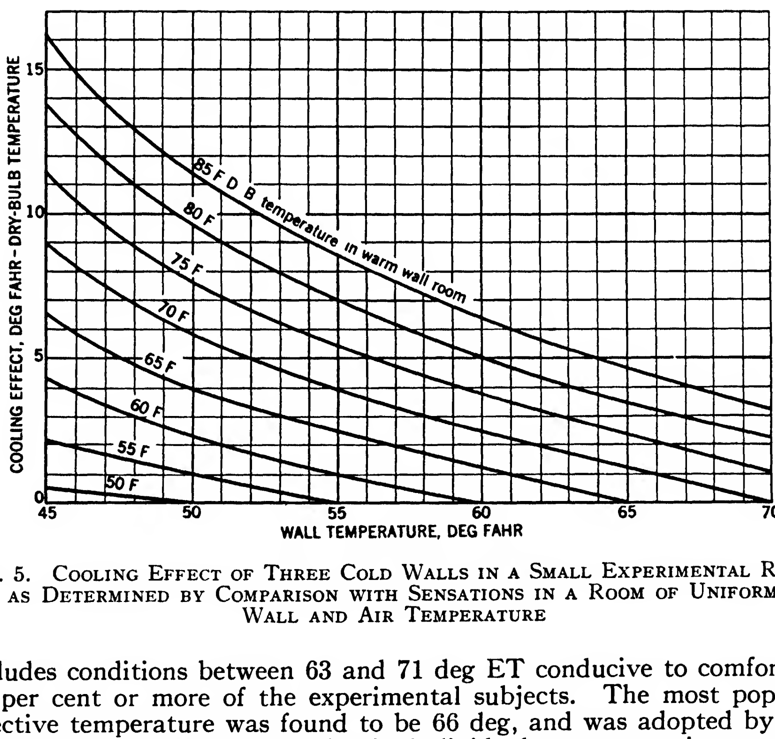

FIG. 5. COOLING EFFECT OF THREE COLD WALLS IN A SMALL EXPERIMENTAL ROOM, AS DETERMINED BY COMPARISON WITH SENSATIONS IN A ROOM OF UNIFORM WALL AND AIR TEMPERATURE

includes conditions between 63 and 71 deg ET conducive to comfort in 50 per cent or more of the experimental subjects. The most popular effective temperature was found to be 66 deg, and was adopted by the Society[25] as the *winter comfort* line for individuals at rest wearing customary winter clothing.

The comfort line separates the cool air conditions to its left from the warm air conditions to its right. Under the air conditions existing along or defined by the comfort line, the body is able to maintain thermal

[24]Application of Air Conditioning to Premature Nurseries in Hospitals, by C P Yaglou, Philip Drinker and K. D Blackfan (A S H V E TRANSACTIONS, Vol 36, 1930, p 383).

[25]How to Use the Effective Temperature Index and Comfort Charts (A S H V E TRANSACTIONS, Vol 38, 1932, p 410)

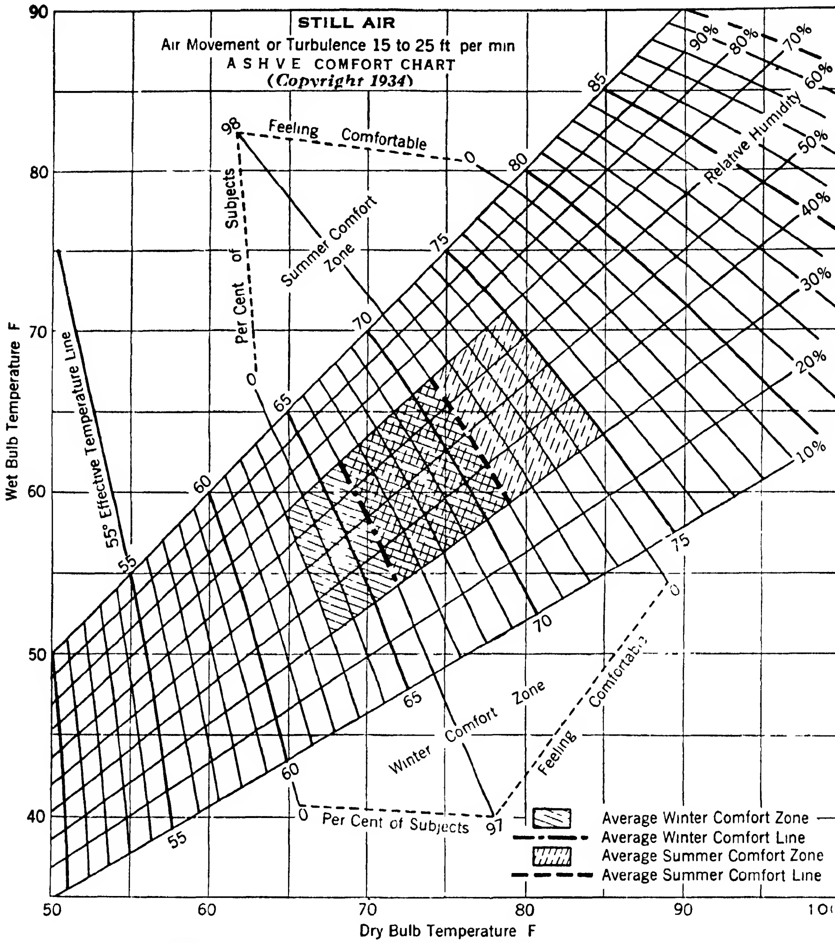

FIG. 6. A.S.H.V.E. COMFORT CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)[26, 27]

Note—Both summer and winter comfort zones apply to inhabitants of the United States only Application of winter comfort line is further limited to rooms heated by central station systems of the convection type The line does not apply to rooms heated by radiant methods Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours

equilibrium with its environment with the least conscious sensation to the individual, or with the minimum physiologic demand on the heat regulating mechanism. This environment involves not only the condition of the air with respect to temperature and humidity, but also the condition of the surrounding objects and wall surfaces. The comfort zone tests were

[26]A S H V E RESEARCH REPORT No 673—Determination of the Comfort Zone With Further Verification of Effective Temperatures Within This Zone, by F C Houghten and C P Yaglou (A S H V E TRANSACTIONS, Vol 29, 1923, p 361)

[27]The Summer Comfort Zone; Climate and Clothing, by C P Yaglou and Philip Drinker (A S H V E TRANSACTIONS, Vol 35, 1929, p 269)

made in rooms with wall surface temperatures approximately the same as the room dry-bulb temperature. For walls of large area having unusually high or low surface temperatures, however, a somewhat lower or higher range of effective temperature is required to compensate for the increased gain or loss of heat to or from the body by radiation as shown in Fig. 5. (See also Chapter 41).

The average winter comfort line (66 deg ET) applies to average American men and women living inside the broad geographic belt across the United States in which central heating of the convection type is generally used during four to eight months of the year. It does not apply to rooms heated by radiant energy, rooms with excessive glass area or rooms with poorly insulated or cold walls. Even in the warm south and southwestern climates, and in the very cold north-central climate of the United States, the comfort chart would probably have to be modified according to climate, living and working conditions, and the degree of acquired adaptation.

In densely occupied spaces, such as classrooms, theaters and auditoriums, somewhat lower temperatures may be necessary than those indicated by the comfort line on account of counter-radiation between the bodies of occupants in close proximity[28].

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated by the comfort line, it should not be expected that all the occupants of a room will feel perfectly comfortable. When the winter comfort line is applied in accordance with the foregoing recommendations, the majority of the occupants will be perfectly comfortable, but there will always be a few who would feel *a bit too cool* and a few *a bit too warm*. These individual differences among the minority should be counteracted by suitable clothing.

Air conditions lying outside the average comfort zone but within the extreme comfort zone may be comfortable to certain persons. In other words, it is possible for half of the occupants of a room to be comfortable in air conditions outside the *average* comfort zone, but in the majority of cases, if not in all, these conditions will be well within the extreme comfort zone as determined experimentally.

The comfort chart (Fig. 6) applies to adults between 20 and 70 years of age living in the northeastern parts of the United States. For prematurely born infants, the optimum temperature varies from 100 F to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent[29]. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 F to 68 F with natural indoor humidities. For school children, the studies of the *New York State Commission on Ventilation* place the optimum air conditions at 66 F to 68 F temperature with a moderate humidity (not specified) and a moderate but not excessive amount of air movement (not specified)[30].

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[28]Loc Cit Note 27
[29]Loc Cit Note 24
[30]Ventilation (Report N Y State Commission on Ventilation E P Dutton and Co, N Y., 1923)

Satisfactory comfort conditions for men at work are found to vary from 40 deg to 70 deg ET, depending upon the rate of work and amount of clothing worn[31]. In hot industries, 80 deg ET is considered the upper limit compatible with efficiency, and, whenever possible, this should be reduced to 70 deg ET or less.

Summer Comfort Zones

The summer comfort zone is much more difficult to fix than the winter zone owing to the complicating factor of sweating in warm weather. A given air condition which is comfortable for persons with dry skin and clothing may prove too cold for those perspiring, as is the case, for instance, with employees and customers in a cooled store, restaurant, or theater, on a warm summer day. The conditions to be maintained in different types of public buildings depend to a large extent upon the occupants' length of stay and upon the prevailing outdoor condition.

In Fig. 6 is shown the summer comfort zone for exposures of 3 hours or more, after adaptation has taken place. The average zone extends from 66 to 75 deg ET, with a comfort line at 71 deg ET, as determined at the Harvard School of Public Health[32]. These effective temperatures average about 4 deg higher than those found in winter when customary winter clothing was worn. The variation from winter to summer is probably due partly to adaptation to seasonal weather and partly to differences in the clothing worn in the two seasons.

The best effective temperature (for exposures lasting 3 hours or more) was found to follow the average monthly outdoor temperature more closely than the prevailing outdoor temperature. It remained at approximately the same value in July, August and September, and although the average monthly temperature did not vary much, the prevailing outdoor temperature ranged from 70 F to 99.5 F. A decrease in the optimum temperature became apparent only when the prevailing outdoor temperature fell to 66 F, which is below the customary room temperature in the United States for summer and winter.

Crowding the experimental chamber lowered the comfortable effective temperature from 70.8 deg when the gross floor area per occupant was 44 sq ft and the air space 380 cu ft, to 69.4 deg when the floor area was reduced to 14 sq ft and the air space to 120 cu ft per occupant.

The basic summer comfort zone, shown in Fig. 6 has more academic than practical significance. It prescribes conditions of choice for continuous exposures, as in homes, offices, etc., without regard to costs, prevailing outdoor air conditions, and temperature contrasts upon entering or leaving the cooled space. A great number of persons seem to be content with a higher plane of indoor temperature, particularly when the matter of first cost and cost of operation of the cooling plant is given due consideration.

According to previous investigations[33], an indoor temperature of about 80 F with relative humidities below 55 per cent, or 74.5 deg ET and lower, result in satisfactory comfort conditions in the living quarters of a residence,

[31]Loc Cit Note 22
[32]Loc Cit Note 27
[33]A S H V E RESEARCH REPORT No 1012—Study of Summer Cooling in the Research Residence for the Summer of 1934, by A P Kratz, S Konzo, M. K Fahnestock and E L Broderick (A S H V E TRANSACTIONS, Vol 41, 1935, p 207)

and while this condition is not representative of optimum comfort it provides for sufficient relief in hot weather to be acceptable to the majority of users. Experience in a number of air conditioned office buildings, including the New Metropolitan Life Building in New York[34], indicates that a temperature of about 80 F with a relative humidity between 45 and 55 per cent (73 to 74.5 deg ET) is generally satisfactory in meeting the requirements of the employees.

In artificially cooled theaters, restaurants, and other public buildings where the period of occupancy is short, the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the temperature and humidity to be maintained. The object of cooling such places in the summer is to provide sufficient relief from the heat without causing sensations of chill or intense heat on entering and leaving the building.

Effective temperatures as high as 75 F at times have been found satisfactory in very warm weather. There are two schools of thought concerning the relation between temperature and humidity to be maintained. For a given effective temperature some engineers including the operators of cooling plants favor a comparatively low temperature with a high humidity as this results in a reduction of refrigeration requirements. Preliminary experiments at the A.S.H.V.E. Laboratories[35] would seem to indicate no appreciable impairment of comfort with relative humidities as high as 80 per cent, provided the effective temperature is between 70 and 75 deg.

The second school favors a higher dry-bulb temperature, according to the prevailing outdoor dry-bulb, with a comparatively low humidity (well below 50 per cent); the main purpose being to reduce temperature contrasts upon entering and leaving the *cooled* space and to keep the clothing and skin dry. This second scheme requires more refrigeration with the present conventional type of apparatus.

Current practice in theatres, restaurants, etc., follows a schedule similar to that shown in Table 2. This schedule should be used with considerable judgment depending on the occupancy and local climatic conditions. There are some indications that a definite indoor effective temperature may be applicable throughout the cooling season, but other observations seem to show that changing indoor conditions are desirable with violently changing outdoor weather conditions. It is, in fact, questionable whether entirely satisfactory air conditions could be adduced for practical use to meet the changing requirements of patrons from the time they enter to the time they leave a cooled space. Too many uncontrollable variables enter into the problem. Work now going on at the A.S.H.V.E. Laboratory and other interested institutions may throw considerable light on this complex problem.

For cooled banks and stores where the customers come and go spending

[34]The Air Conditioned System of the New Metropolitan Building—First Summer's Experience, by W. J. McConnell and I B. Kagey (A.S.H V E TRANSACTIONS, Vol 40, 1934, p. 217)

[35]A.S.H V.E RESEARCH REPORT No 1035—Comfort Standards for Summer Air Conditioning, by F C. Houghten and Carl Gutberlet (A S H V E TRANSACTIONS, Vol 42, 1936, p 215) A S H V E. RESEARCH PAPER—Cooling Requirements for Summer Air Conditioning, by F. C Houghten, F E Giesecke, Cyril Tasker and Carl Gutberlet (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1936, p 681).

TABLE 2. DESIRABLE INSIDE CONDITIONS IN SUMMER CORRESPONDING TO OUTSIDE TEMPERATURES[a]

Occupancy Over 40 Min

Outside Dry-Bulb Deg F	Inside Air Conditions				
	Effective Temperature	Dry-Bulb Deg F	Wet-Bulb Deg F	Dew-Point Deg F	Relative Humidity Per Cent
100	75	83	66	56	40
	75	82	67	59	45
	75	81	68	61	51
	75	80	70	65	60
95	74	82	64	53	36
	74	81	66	57	44
	74	80	67	60	51
	74	79	68	62	57
	74	78	70	66	68
90	73	81	63	52	36
	73	80	64	54	41
	73	79	66	59	50
	73	78	67	61	56
85	72	80	61	48	32
	72	79	63	53	41
	72	78	64	56	46
	72	77	66	60	56
80	71	78	61	49	36
	71	77	63	54	45
	71	76	64	57	52
	71	75	66	61	61

[a]Applicable to individuals engaged in sedentary or light muscular activity

but a few minutes in the cooled space, observations[36] indicate a schedule about 1 deg dry-bulb or effective temperature higher than that shown in Table 2. Laboratory experiments with exposures of 2 to 10 min indicate temperatures 2 to 10 F higher than those in Table 2 but with much lower relative humidities.

It should be kept in mind that southern people, with their more sluggish heat production and lack of adaptability, will demand a comfort zone several degrees higher than that for the more active people of northern climates. Instead of the summer comfort line standing at 71 deg as here given, it was found to be much higher for foreigners in Shanghai where climatic conditions are similar to those of our gulf states. This difference in adaptability of people forms a very real problem for air conditioning engineers. Cooling of theaters, restaurants, and other public buildings in southern climates cannot be based on northern standards without considerable modification.

Optimum Humidity

Just what the optimum range of humidity is, is a matter of conjecture. There seems to exist a general opinion, supported by some experimental and statistical data, that warm, dry air is less pleasant than air of a

[36]How Cool? Inside Temperature Should Depend Upon Type of Occupancy, by J H Walker (*Heating and Ventilating*, October, 1932).

moderate humidity, and that it dries up the mucous membranes in such a way as to increase susceptibility to colds and other respiratory disorders [37, 38, 39]. Owing to the cooling effect of evaporation, higher temperatures are necessary, and this condition may lead to discomfort and lassitude. Moist air, on the other hand, interferes with the normal evaporation of moisture from the skin, and again may cause a feeling of oppression and lassitude, especially when the temperature is also high. For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth[40] until the infants reach a weight of about 5 lb. No such clear-cut evidence exists in the case of adult persons. In the comfort zone experiments of the A.S.H.V.E. Research Laboratory, the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the Harvard School of Public Health, the majority of the subjects were unable to detect sensations of humidity (*i.e.*, too high, too low, or medium) when the relative humidity was between 30 per cent and 60 per cent with ordinary room temperatures. This is in accord with studies by Howell[41], Miura[42] and others.

The limitation of the comfort zones in Fig. 6 with respect to humidity must not be taken too seriously. Relative humidities below 30 per cent may prove satisfactory from the standpoint of comfort, so long as extremely low humidities are avoided. In mild weather comparatively high relative humidities are entirely feasible, but in cold or sub-freezing weather they are objectionable on account of condensation and frosting on the windows. They may even cause serious damage to certain building materials of the exposed walls by condensation and freezing of the moisture accumulating inside these materials. Unless special precautions are taken to properly insulate the affected surfaces, it will be necessary to reduce the degree of artificial humidification in sub-freezing weather to less than 40 per cent, according to the outdoor temperature. Information on the prevention of condensation on building surfaces is given in Chapter 7 The principles underlying humidity requirements and limitations are discussed more fully elsewhere[43].

The purpose of artificial humidification may be easily defeated by failure to change the spray water of the humidifier at least daily. Where this condition occurs, the air is characterized by a *lack of freshness*, and under extreme conditions by a musty, sour odor in the conditioned space.

[37]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surface, by Mudd, Stuart, et al (*Journal Experimental Medicine*, 1921, Vol 34, p 11)

[38]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surfaces, by A Goldman, et al, and Concurrent Study of Bacteriology of Nose and Throat (*Journal Infectious Diseases*, 1921, Vol 29, p 151)

[39]The Etiology of Acute Inflammations of the Nose, Pharynx and Tonsils, by Mudd, Stuart, et al (Am Otol , Rhinol , and Laryngol , 1921)

[40]Loc Cit Note 24

[41]Humidity and Comfort, by W H Howell (*The Science Press*, April, 1931)

[42]Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U Miura (*American Journal of Hygiene*, Vol 13, 1931, p 432)

[43]Humidification for Residences, by A P Kratz, University of Illinois (*Engineering Experiment Station Bulletin* No 230, July 28, 1931)

AIR QUALITY AND QUANTITY

Air Quality

In occupied spaces in which the vitiation is entirely of human origin, the chemical composition of the air, the dust, and often the bacteria content may be dismissed from consideration so that the problem consists in maintaining a suitable temperature with a moderate humidity, and in keeping the atmosphere free from objectionable odors. Such unpleasant odors, human or otherwise, can be easily detected by persons entering the room from clean, odorless air.

In industrial rooms where the primary consideration is the control of air pollution (dusts, fumes, gases, etc.), or contamination not removable at the source of production, the clean air supply must be sufficient to dilute the polluting elements to a concentration below the physiological threshold (see Chapters 4 and 26).

Air Quantity

The air supply to occupied spaces must always be adequate to satisfy the physiological requirements of the occupants. It must be sufficient to maintain the desired temperature, humidity, and purity with reasonable uniformity and without drafts. In many practical instances there are two air quantities to be considered, (a) outdoor air supply, and (b) total air supply. The difference between the two gives the amount of air to be recirculated.

When the only source of contamination is the occupant, the minimum quantity of outdoor air needed appears to be that necessary to remove objectionable body odors, or tobacco smoke. The concentration of body odor in a room, in turn, depends upon a number of factors, including socio-economic status of occupants, outdoor air supply, air space allowed per person, odor adsorbing capacity of air conditioning processes, temperature, and other factors of secondary importance. With any given group of occupants and type of air conditioner the intensity of body odor perceived upon entering a room from relatively clean air was found to vary inversely with the logarithm of outdoor air supply and the logarithm of the air space allowed per person.

The minimum outdoor air supply necessary to remove objectionable body odors under various conditions, as determined experimentally at the Harvard School of Public Health[44], is given in Table 3.

Outdoor air requirements for the removal of objectionable tobacco smoke odors have yet to be determined. Practical values in the field vary from 5 to 15 cfm per person; this air quantity may and should be a part of that necessary for other requirements, *i.e.*, removal of body odors, heat, moisture, etc.

The total quantity of air to be circulated through an enclosure is governed largely by the needs for controlling temperature and air distribution when either heating or cooling is required. The factors which determine total air quantity include the type and nature of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and last but not least, the method of distribution.

Serious difficulties are often encountered in attempting to cool a room

[44]Loc. Cit. Note 3.

with a poor distribution system or with an air supply which is too small to result in uniform distribution without drafts. Some systems of distribution produce drafts with but a few degrees temperature rise, while other systems operate successfully with a temperature rise as high as 35 F. The total air quantity introduced in any particular case is inversely proportional to the temperature rise, and depends largely upon the judgment and ingenuity of the engineer in designing the most suitable system for the particular conditions.

TABLE 3. MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS

(Provisional values subject to revision upon completion of work)

TYPE OF OCCUPANTS	AIR SPACE PER PERSON CU FT	OUTDOOR[a] AIR SUPPLY CFM PER PERSON
Heating season with or without recirculation	*Air not conditioned*	
Sedentary adults of average socio-economic status	100	25
Sedentary adults of average socio-economic status	200	16
Sedentary adults of average socio-economic status	300	12
Sedentary adults of average socio-economic status	500	7
Laborers	200	23
Grade school children of average class	100	29
Grade school children of average class	200	21
Grade school children of average class	300	17
Grade school children of average class	500	11
Grade school children of poor class	200	38
Grade school children of better class	200	18
Grade school children of best class	100	22
Heating season Air humidified by means of centrifugal humidifier Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary Adults	200	12
Summer season Air cooled and dehumidified by means of a spray dehumidifier Spray water changed daily Total air circulation 30 cfm per person.		
Sedentary Adults	200	<4

[a]Impressions upon entering room from relatively clean air at threshold odor intensity.

The changes in moisture content resulting from occupation in the atmosphere of a room supplied with various volumes of outside air is shown in Fig. 7. Data are given for an adult, 5 ft 8 in. in height, weighing 150 lb and having a body surface of 19.5 sq ft and for a child, 12 years of age, 4 ft 7 in. in height weighing 76.6 lb and having a body surface area of 12.6 sq ft. Also given in Fig. 7 is the temperature of incoming air necessary to maintain a room temperature of either 70 or 80 F as indicated assuming that there is no heat gain or loss to the room by transmission through the walls, solar radiation or other sources.

AIR MOVEMENT AND DISTRIBUTION

Stagnant warm air, no matter how pure, is not stimulating and it detracts to some extent from the quality of air. Experience, and recent field studies by the A.S.H.V.E. Research Laboratory[45] place the desirable air movement between 15 and 25 fpm under ordinary room temperatures during the heating season. Objectionable drafts are likely to occur when the velocity of the air current is 40 fpm and the temperature of the air current 2 F or more below the customary winter room temperature. Higher velocities are not objectionable in the summer time when the air

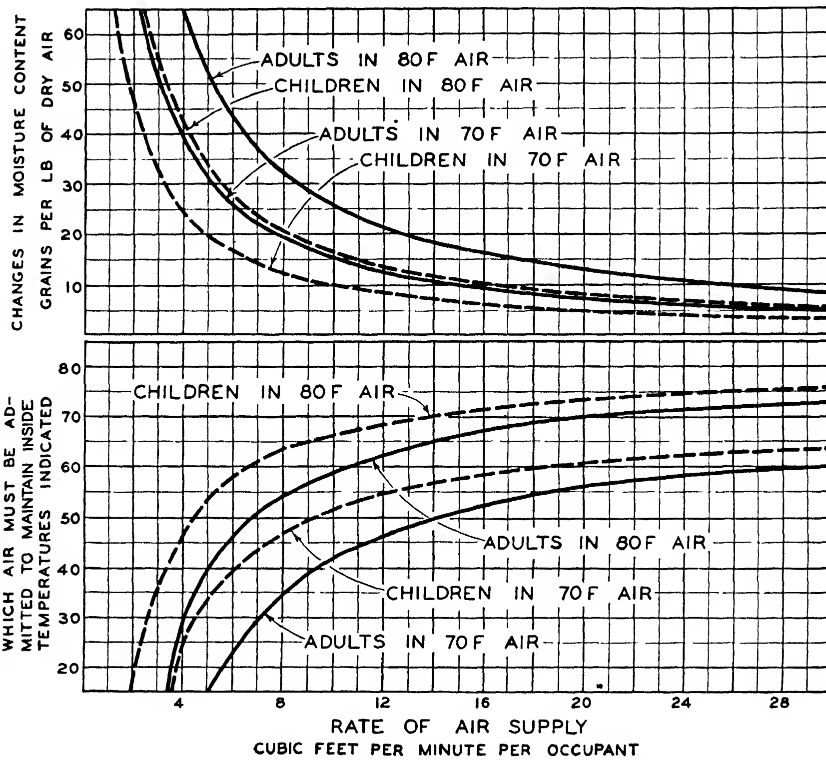

FIG. 7. RELATION AMONG RATE OF AIR CHANGE PER OCCUPANT, MOISTURE CONTENT OF ENCLOSURE, AND DRY-BULB TEMPERATURE OF INCOMING AIR

temperature exceeds 80 F. Variations in air movement and temperature in different parts of occupied rooms are often indicative of relative air distribution. The work of the A.S.H.V.E. Research Laboratory indicates that an air movement between 15 and 25 fpm with a temperature variation of 3 F or less in different parts of a room, 36 in. above floor, insure satisfactory distribution. Considerable evidence was obtained in these tests to show that measurements of carbon dioxide are not essential for the study of air distribution, or for indirect measurements of outdoor air

[45]A S H V E RESEARCH REPORT No 1016—Classroom Drafts in Relation to Entering Air Stream Temperature, by F C Houghten, H H Trimble, Carl Gutberlet and M F Lichtenfels (A S H V E TRANSACTIONS, Vol 41, 1935, p 268)

TABLE 4. RELATION BETWEEN METABOLIC RATE AND ACTIVITY[a]

ACTIVITY	HOURLY METABOLIC RATE FOR AVG PERSON OR TOTAL HEAT DISSIPATED, BTU PER HOUR	HOURLY SENSIBLE HEAT DISSIPATED, BTU PER HOUR	HOURLY LATENT HEAT DISSIPATED, BTU PER HOUR	MOISTURE DISSIPATED, PER HOUR	
				GRAINS	LB
Average Person Seated at Rest[1]	384	225	159	1070	0.153
Average Person Standing at Rest[1]	431	225	206	1390	0.199
Tailor[2]	482	225	257	1740	0 248
Office Worker Moderately Active	490	225	265	1790	0 256
Clerk, Moderately Active, Standing at Counter	600	225	375	2530	0 362
Book Binder[2]	626	225	401	2710	0 387
Shoe Maker[2]; Clerk, Very Active Standing at Counter	661	225	436	2940	0 420
Pool Player	680	230	450	3040	0 434
Walking 2 mph[3, 4], Light Dancing	761	250	511	3450	0 493
Metal Worker[2]	862	277	585	3950	0 564
Painter of Furniture[2]	876	280	596	4020	0 575
Restaurant Serving, Very Busy	1000	325	675	4560	0 651
Walking 3 mph[3]	1050	346	704	4750	0 679
Walking 4 mph[3, 4], Active Dancing, Roller Skating	1390	452	938	6330	0 904
Stone Mason[2]	1490	490	1000	6750	0 964
Bowling	1500	490	1010	6820	0 974
Man Sawing Wood[2]	1800	590	1210	8170	1 167
Slow Run[4]	2290	.....	.....	.....	.....
Walking 5 mph[3]	2330	.....	.....	.....	.....
Very Severe Exercise[5]	2560	.....	.....	.....	.....
Maximum Exertion Different People[4]	3000 to 4800	.....	.....	.....	.....

[a]Metabolism rates noted based on tests actually determined from the following authoritative sources [1]A S H V E Research Laboratory, [2]Becker and Hamalainen, [3]Douglas, Haldane, Henderson and Schneider; [4]Henderson and Haggard, and [5]Benedict and Carpenter Metabolic rates for other activities estimated Total heat dissipation integrated into latent and sensible rates by actual tests for metabolic rates up to 1250 Btu per hour, and extrapolated above this rate Values for total heat dissipation apply for all atmospheric conditions in a temperature range from approximately 60 to 90 F dry-bulb Division of total heat dissipation rates into sensible and latent heat holds only for conditions having a dry-bulb temperature of 79 F For lower temperatures, sensible heat dissipation increases and latent heat decreases, while for higher temperatures the reverse is true

TABLE 5 DEGREES OF PERSPIRATION FOR PERSONS SEATED AT REST UNDER VARIOUS ATMOSPHERIC CONDITIONS

DEGREE OF PERSPIRATION[a]	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E T	D B	W B	E T	D B	W B
Forehead clammy	**73.0**	73.6	72 4	**75.0**	87.0	60.7
Body clammy	**73.0**	73 6	72 4	**75.0**	87.0	60.7
Body damp	**79.0**	79.7	78.4	**81.0**	97.5	67.5
Beads on forehead	**80.0**	80.8	79.4	**87.0**	109 4	75.2
Body wet	**84.5**	85.4	84.0	**86.5**	108.5	74 6
Perspiration on forehead runs and drips	**88.0**	89.0	87.6	**94.0**	125.2	85.4
Perspiration runs down body	**88.5**	89 5	88.1	**90.0**	116.0	79.5

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated

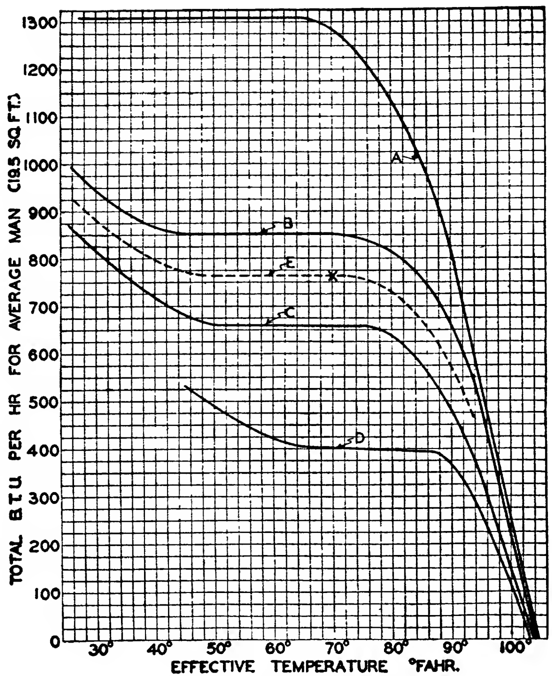

FIG. 8. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,160 ft-lb per hour Curve *B*—Men working 33,075 ft-lb per hour Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest Curves *A* and *C* drawn from data at an effective temperature of 70 deg only and extrapolating the relation between curves *B* and *D*, which were drawn from data at many temperatures

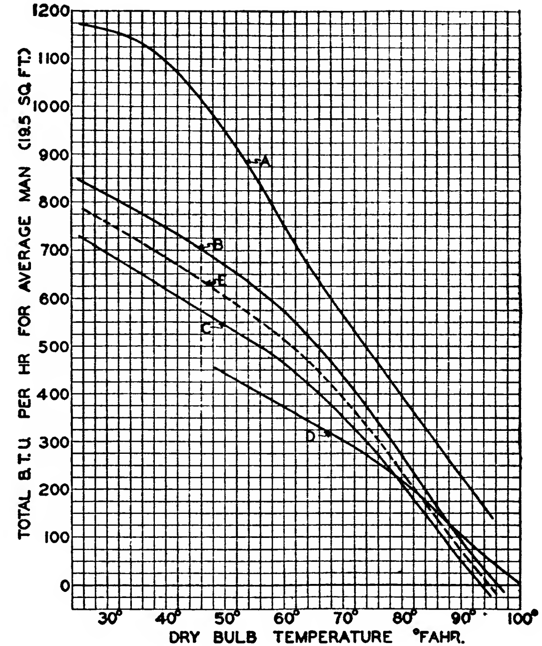

FIG. 9 RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour Curve *B*—Men working 33,075 ft-lb per hour Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest Curves *A* and *C* drawn from data at a dry-bulb temperature of 81 3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures

TABLE 6. DEGREES OF PERSPIRATION FOR PERSONS AT WORK UNDER VARIOUS ATMOSPHERIC CONDITIONS

Work Rate 33,000 Ft Lb per Hour

DEGREE OF PERSPIRATION[a]	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E T.	D B.	W. B	E. T.	D. B	W. B
Forehead clammy	**59.0**	59.4	58.3	**69.5**	80.5	56.5
Body clammy	**50.0**	50.2	49.3	**57.0**	61.6	44.2
Body damp	**60.0**	60.3	59.3	**62.5**	69 6	49.5
Beads on forehead	**68.0**	68.5	67.5	**76.0**	91 0	63.4
Body wet	**69.0**	69.6	68.5	**71.0**	82.8	53 0
Perspiration on forehead runs and drips	**78.5**	79.3	78 0	**82.0**	100.5	70.2
Perspiration runs down body	**79.0**	79.8	78.5	**81.0**	99 8	69.0

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated

supply, which can be obtained more conveniently from the increase in moisture content of the ventilating current.

HEAT AND MOISTURE GIVEN UP BY HUMAN BODY

In conditioning air for comfort and health it is necessary to know the rate of sensible and latent heat liberation, from the human body, which in conjunction with other heat loads (see Chapters 5 and 7) determine the capacity of the conditioner. The data in common use are those of the A.S.H.V.E. Research Laboratory[46] shown in Figs. 8, 9, 10 and 11. Other useful data are given in Tables 4, 5 and 6, which are self-explanatory.

ULTRA-VIOLET RADIATION AND IONIZATION

In spite of the rapid advances in the field of air conditioning during the past few years, the secret of reproducing indoor atmospheres of as stimulating qualities as those existing outdoors under ideal weather conditions, has not as yet been found. Extensive studies have failed to elucidate the cause of the stimulating quality of country air, qualities which are lost when such air is brought indoors and particularly when it is handled by mechanical means. Ultra-violet light and ionization have been suggested but the evidence so far is inconclusive or negative[47].

[46]Thermal Exchanges between the Bodies of Men Working and the Atmospheric Environment, by F C Houghten W W Teague, W. E Miller and W P Yant (*American Journal of Hygiene*, Vol XIII, No. 2, March, 1931 pp 415-431)

[47]A S H V E RESEARCH REPORT No 921—Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C P Yaglou, L C Benjamin and S P Choate (A S H V E TRANSACTIONS, Vol 38, 1932, p 191) A S H V E RESEARCH REPORT No 965—Physiologic Changes During Exposure to Ionized Air, by C P Yaglou, A D Brandt and L C Benjamin (A S H V E TRANSACTIONS, Vol 39, 1933, p 357) A S H V E RESEARCH REPORT No 985—Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C P Yaglou and L C Benjamin (A S H V E TRANSACTIONS, Vol 40, 1934, p 271) The Nature of Ions in Air and Their Possible Physiological Effects, by L B Loeb A S H V E TRANSACTIONS, Vol 41, 1935, p 101) The Influence of Ionized Air upon Normal Subjects by L P Herrington (*Journal Clinical Investigation*, 14, January, 1935) The Effect of High Concentrations of Light Negative Atmospheric Ions on the Growth and Activity of the Albino Rat, by L P Herrington and Karl L Smith (*Journal Ind Hygiene*, 17, November, 1935) Subjective Reactions of Human Beings to Certain Outdoor Atmospheric Conditions, by C-E A Winslow and L P. Herrington (A S H V E TRANSACTIONS, Vol 42, 1936, p 119)

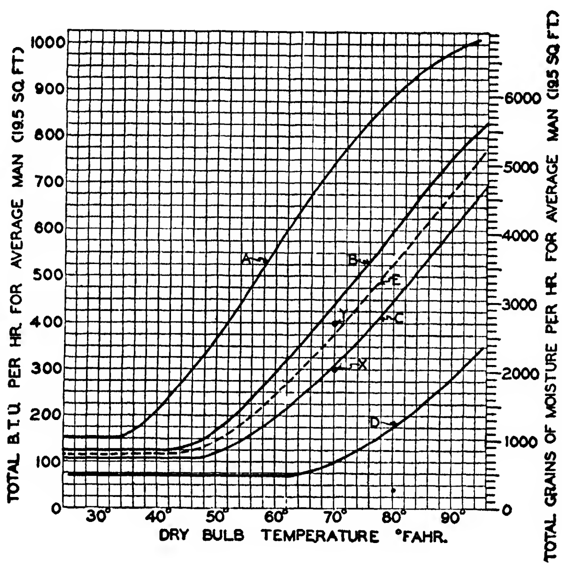

FIG. 10. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest Curves *A* and *C* drawn from data at a dry-bulb temperature of 81 3 F only and extrapolating the relation between Curves *B* and *D* which were drawn from data at many temperatures

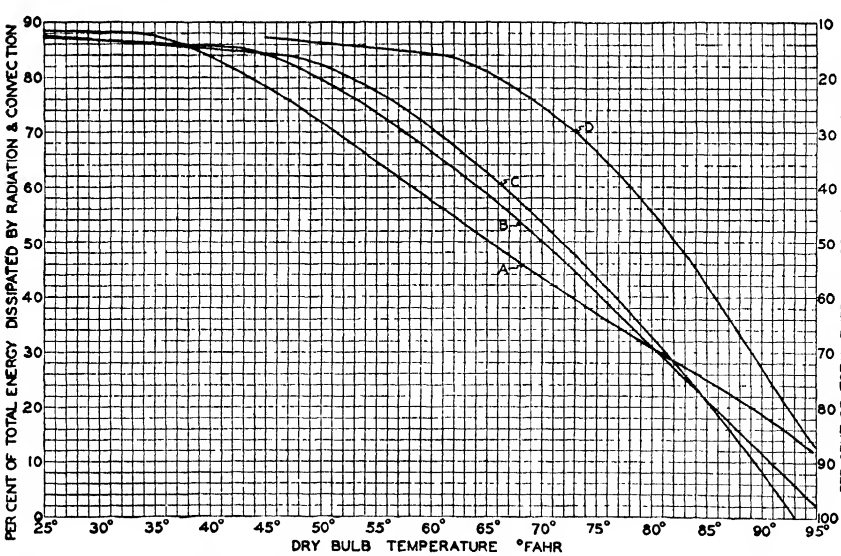

FIG. 11. HEAT LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION AND CONVECTION IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour Curve *C*—Men working 16,538 ft-lb per hour Curve *D*—Men seated at rest Curves *A* and *C* drawn from data at a dry-bulb temperature of 81 3 F only and extrapolating the relation between Curves *B* and *D* which were drawn from data at many temperatures

NATURAL AND MECHANICAL VENTILATION

Under favorable conditions natural ventilation methods properly combined with means for heating may be sufficient to provide for the foregoing objectives in homes, uncrowded offices, small stores, etc.

In large offices, large school rooms, and in public and industrial buildings, natural ventilation is uncertain and makes heating difficult. The chief disadvantage of natural methods is the lack of control; they depend largely on weather and upon the velocity and direction of the wind. Rooms on the windward side of a building may be difficult to heat and ventilate on account of drafts, while rooms on the leeward side may not receive an adequate amount of air from out-of-doors. The partial vacuum produced on the leeward side under the action of the wind may even reverse the flow of air so that the leeward half of the building has to take the *drift* of the air from the rooms of the windward half. Under such conditions no outdoor air would enter through a leeward window opening, but room air would pass out.

In warm weather natural methods of ventilation afford little or no control of indoor temperature and humidity. Outdoor smoke, dust and noise constitute other limitations of natural methods.

RECIRCULATION AND OZONE

The amount of recirculated air may be varied to suit changes in weather and seasonal requirements, so as to conserve heat in winter and refrigeration in summer, but the saving in operating cost should not be obtained at the expense of air quality.

Ozone has been used for deodorizing recirculated air by oxidation or *masking*. Under favorable conditions some success is possible but from the practical standpoint it is difficult to regulate the ozone output so as to just neutralize undesirable odors at all times during the occupancy of a room. The difficulties appear to be mainly due to a wide variability in the rate of ozone disappearance in different rooms, or in the same room at different times, according to the characteristics of a room, the absolute humidity, impurities in the air, number and type of occupants, and probably other factors which require considerable study before ozone can be safely and economically applied.

The allowable concentrations in the breathing zone are very small, between 0.01 to 0.05 parts of O_3 per million parts of air. These are much too small to influence bacteria. Higher concentrations are associated with a pungent unpleasant odor and considerable discomfort to the occupants. One part per million causes respiratory discomfort in man, headaches and depression, lowers the metabolism, and may even lead to coma[48].

Toilets, kitchens, and similar rooms, in buildings using recirculation, should be ventilated separately by mechanical exhaust in order to prevent objectionable odors from diffusing into other parts of the building.

[48]*The British Medical Journal.* Editorial, June 25, 1932, p. 1182. See also Loc. Cit. Note 5

PROBLEMS IN PRACTICE

1 ● What are the most comfortable air conditions?

Comfort standards are not absolute, but they are greatly affected by the physical condition of the individual, and the climate, season, age, sex, clothing, and physical activity. For the northeastern climate of the United States, the conditions which meet the requirements of the majority of people consist of temperatures between 68 and 72 F in winter and between 70 and 85 F in summer, the latter depending largely upon the prevailing outdoor temperature The most desirable relative humidity range seems to be between 30 and 60 per cent.

2 ● Are the optimum conditions for comfort identical with those for health?

There are no absolute criteria of the prolonged effects of various air conditions on health For the present it can be only inferred that bodily discomfort may be an indication of adverse conditions leading to poor health.

3 ● Given dry-bulb and wet-bulb temperatures of 76 F and 62 F, respectively, and an air velocity of 100 fpm, determine: (1) effective temperature of the condition; (2) effective temperature with calm air; (3) cooling produced by the movement of the air.

(1) In Fig. 1 draw line *AB* through given dry- and wet-bulb temperatures. Its intersection with the 100 ft velocity curve gives 69 deg for the effective temperature of the condition. (2) Follow line *AB* to the right to its intersection with the 20 fpm velocity line, and read 70 4 deg for the effective temperature for this velocity or so-called still air. (3) The cooling produced by the movement of the air is 70 4 − 69 = 1.4 deg ET.

4 ● Assume that the design of an air conditioning system for a theater is to be based on an outdoor dry-bulb temperature of 95 F and a wet-bulb temperature of 78 F with an indoor relative humidity of 50 per cent. According to Table 2, the dry-bulb temperature in the auditorium should be 80 F. Estimate the sensible and latent heat given up per person.

The sensible heat given up per person per hour may be obtained from Fig 9 With an abscissa value of 80 F, Curve *D* for men seated at rest gives a value (on the ordinate scale) of 220 Btu per person per hour as the sensible heat loss. The latent heat given up by a person seated at rest may be obtained from Fig. 10. With an abscissa value of 80 F, Curve *D* indicates a latent heat loss of 175 Btu per hour (left hand scale) or a moisture loss of 1190 grains per hour (right hand scale).

5 ● Neglecting the gain or loss of heat by transmission or infiltration through walls, windows and doors, how many cubic feet of outside air, with dry- and wet-bulb temperatures of 65 F and 59 F, respectively, (63.1 deg ET) must be supplied per hour to an auditorium containing 1000 people in order that the inside temperature shall not exceed 75 F dry-bulb and 65 F wet-bulb?

Figs. 9 and 10 give 265 Btu sensible heat and 905 grains of moisture per person with a dry-bulb temperature of 75 F in the auditorium Therefore, 265,000 Btu of sensible heat and 905,000 grains of moisture will be added to the air in the auditorium per hour.

Taking 0 24 as the specific heat of air, 2 4 Btu per pound of air will be absorbed in raising the dry-bulb temperature from 65 to 75 F, and 265,000 ÷ 2 4 = 110,400 lb of air or 110,400 × 13 4 = 1,479,000 cfh of air will be required. This is equivalent to 1,479,000 ÷ (1000 × 60) = 24.7 cfm per person.

The moisture content of the inside air is 76 grains per pound of dry air and that of the outside condition is 65 grains From a psychrometric chart the increase in moisture content will therefore be 11 grains per pound of dry air. Hence 905,000 ÷ 11 0 = 82,300 lb of air at the specified condition will be required. This is equivalent to 82,300 × 13.4 = 1,103,000 cfh of air or 1,103,000 ÷ (1000 × 60) = 18 4 cfm of air per person.

The higher volume of 24.7 cfm per person will be required to keep the dry-bulb temperature from rising above the 75 F specified. The wet-bulb temperature will therefore not rise to the maximum of 65 F.

Chapter 4

AIR POLLUTION

Classification of Air Impurities, Dust Concentrations, Air Pollution and Health, Occlusion of Solar Radiation, Smoke and Air Pollution Abatement, Dust and Cinders, Nature's Dust Catcher

THE particulate impurities which contribute to atmospheric pollution include carbon from the combustion of fuels, particles of earth, sand, ash, rubber tires, leather, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silks, bits of animal and vegetable matter, and pollen. Microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon.

CLASSIFICATION OF AIR IMPURITIES

The most conspicuous sources of atmospheric pollution may be classified arbitrarily according to the size of the particles as dusts, fumes, and smoke. *Dusts* consist of particles of solid matter varying from 1.0 to 150 microns in size. (micron = 0.001 millimeter or 1/25,000 in.) *Fumes* include particles resulting from chemical processing, combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size. The word fumes may be applied also to mixtures of mists (liquid droplets) and gases as *acid mists*. *Smoke* is composed of fine soot or carbon particles, usually less than 0.1 micron in size, which result from incomplete combustion of carbonaceous materials, such as coal, oil, tar, and tobacco. In addition to carbon and soot, smoke contains unconsumed hydrocarbon gases, sulphur dioxide, carbon monoxide, and other industrial gases capable of injuring property, vegetation, and health.

The lines of demarcation in these three classifications are neither sharp nor positive, but the distinction is descriptive of the nature and origin of the particles, and their physical action. Dusts settle without appreciable agglomeration, fumes tend to aggregate, smoke to diffuse. Particles which approach the common bacteria in size—about 1 micron—are difficult to remove from air and are apt to remain in suspension unless they can be agglomerated by artificial means. The term *fly-ash* is usually applied to the microscopic glassy spheres which form the principal solid constituent of the effluent gases from powdered-coal fired furnaces. *Cinders* denote the larger solid constituents which may be entrained by furnace gases.

It is well established that particles larger than about 1 micron are unlikely to remain suspended in air currents of moderate strength. Only

violent air motion will sustain them in air long enough for them to be breathed. This means that, in hygienic problems, the engineer is concerned mostly with suspensions of particles comparable to the common

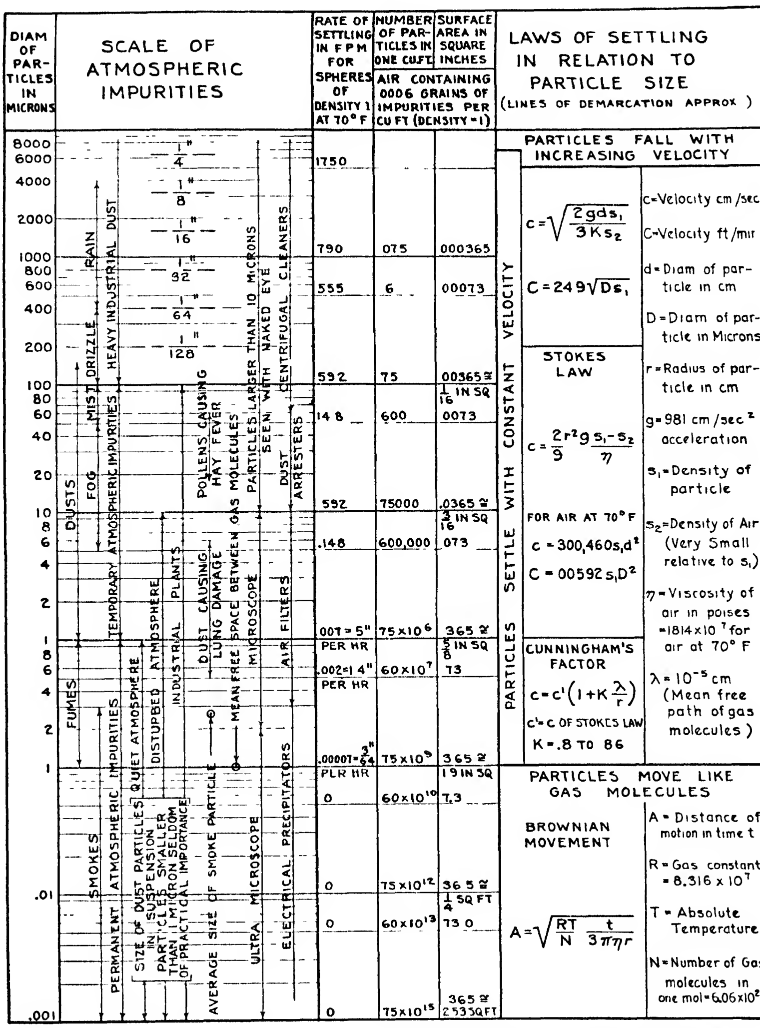

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FIG. 1. SIZES AND CHARACTERISTICS OF AIR-BORNE SOLIDS

bacteria in size. A notable exception to this size limitation is the common hay-fever producing pollen such as that from rag-weed. Pollen grains may be anything from fragments 15 microns in diameter to whole pollens 25 microns or more in size. Since the lower limit of visability to the

TABLE 1. APPROXIMATE LIMITS OF INFLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a]

GAS OR VAPOR	LOWER LIMIT VOLUME IN PER CENT	HIGHER LIMIT VOLUME IN PER CENT
Hydrogen	4 1	74 0
Ammonia	16 0	27 0
Hydrogen sulphide	4 3	46 0
Carbon disulphide	1 0	50 0
Carbon monoxide	12 5	74 0
Methane	5 3	14 0
Methane (turbulent mixture)	5 0	15 0
Ethane	3 2	12 5
Propane	2 4	9 5
Butane	1 9	8 5
Pentane	1 45	7 5
Ethylene	3 0	29 0
Acetylene	3 0	...
Acetylene (turbulent mixture)	2 3	...
Benzene	1 4	7 0
Toluene	1 4	7 0
Cyclohexane	1 3	8 3
Methyl cyclohexane	1 2	...
Methyl alcohol	7 0	...
Ethyl alcohol	4 0	19 0
Ethyl ether	1 7	26 0
Benzine	1 1	...
Gasoline	1 4	6 0
Water gas	6 to 9	55 to 70
Ethylene oxide	3 0	80 0
Acetaldehyde	4 0	57 0
Furfural (125 C)	2 0	...
Acetone	3 0	11 0
Acetone (turbulent mixture)	2 5	...
Methyl ethyl ketone	2 0	12 0
Methyl formate	6 0	20 0
Ethyl formate	3 5	16 5
Methyl acetate	4 1	14 0
Ethyl acetate	2 5	11 5
Propyl acetate	2 0	...
Butyl acetate (30 C)	1 7	...
Ethyl nitrite	3 0	...
Methyl chloride	8 0	19 0
Methyl bromide	13 5	14 5
Ethyl chloride	4 0	15 0
Ethyl bromide	7 0	11 0
Ethylene dichloride	6 0	16 0
Dichlorethylene	10 0	13 0
Vinyl chloride	4 0	22 0
Pyridine (70 C)	1 8	12 5
Natural gas	4 8	13 5
Illuminating gas	5 3	31 0
Blast-furnace gas	35 0	74 0

[a]Limits of Inflammability of Gases and Vapors, by H F Coward and G W Jones, (*U S Bureau of Mines, Bulletin* No 279, 1931)

average eye is around 50 microns all air floated material of this kind is too small to identify without the aid of the microscope.

Mineral particles, such as grains of sand, bits of rock, volcanic ash, or fly-ash, can be transported long distances under unusual circumstances. Thus, the dust storms of 1935 in the Kansas district resulted in vast amounts of fine top soil being thrown high into the air. Solar illumination as far east as Boston was affected noticeably and particles as large as 40 to 50 microns were actually carried half way across the continent before they settled out. In similar manner volcanic ash has been carried even further. It is not surprising, therefore, that fly-ash from furnace gases, cement dust and the like, can be carried for considerable distances and occasionally the engineer is confronted with the problem of removing such material before the air in question is suitable for use in building ventilation.

The physical properties of the particulate impurities of air are summarized conveniently in the chart of Fig. 1.

In the case of gases the physical property which is probably of most importance is inflammability. The best data available at present on this subject are given in Table 1.

Dust Concentrations

It is customary to report dust concentrations as grains per 1000 cu ft or milligrams per cubic meter. Gas concentrations are commonly recorded as milligrams per cubic meter or as parts per million or as per cent by volume. Typical ranges in dust concentrations as now found in practical applications are given in Table 2.

TABLE 2. DUST CONCENTRATION RANGES IN PRACTICAL APPLICATIONS[a]

APPLICATION	GRAINS PER 1000 CU FT	MGS PER CU M
Rural and suburban districts	0 2 to 0 4	0 4 to 0 8
Metropolitan districts	0 4 to 0 8	0 9 to 1 8
Industrial districts	0 8 to 1 5	1 8 to 3 5
Dusty factories or mines	4 0 to 80 0	10 to 200
Explosive concentrations (as of flour or soft coal)	4000 to 8000	10,000 to 20,000

[a]1 grain per 1000 cu ft = 2 3 mgs per cubic meter, 1 oz per cubic foot = 1 gram per liter

The engineer frequently desires information regarding the effects of various concentrations of gases or dusts upon man, as the success of a particular installation may depend upon the maintenance of air which is adequately clean. At the present time there are a number of organizations working on this problem all of them publishing literature of various kinds.[1] References to books covering the hygienic significance, determination and control of dust are listed at the end of this chapter.

[1]National Institute for Health, U S Public Health Service; Division of Labor Standards, U S Department of Labor; University of Toronto Medical School, Canada, Saranac Laboratories, Saranac Lake, N Y ; Air Hygiene Foundation, Inc , Pittsburgh Pa , Harvard School of Public Health Boston, Mass ; Haskell Laboratory, Wilmington, Del ; and the Departments of Health and of Labor in the United States and in various provinces of Canada

AIR POLLUTION AND HEALTH

The prevention of various diseases which result from exposure to atmospheric impurities is an engineering problem. It is important for the engineer to insure, by proper ventilation, suitable environments for working or for general living. If the equipment used is to be successful, it must operate automatically as in the modern air conditioned theatre or railroad train.

In Table 3 are given data on permissable concentrations of various substances, gases and dusts, which occur in industry. The prudent

TABLE 3. TOXICITY OF GASES AND FUMES IN PARTS PER 10,000 PARTS OF AIR[a]

Vapor or Gas	Rapidly Fatal	Maximum Concentration for from ½ to 1 Hour	Maximum Concentration for 1 Hour	Maximum Allowable for Prolonged Exposure
Carbon monoxide	40	15–20	10	1
Carbon dioxide	800–1000	....	....	....
Hydrocyanic acid	30	1½	½	⅕
Ammonia	50–100	25	3	1
Hydrochloric acid gas	10–20	½	....	1/10
Chlorine	10	½	....	1/100
Hydrofluoric acid gas	2	1/10	....	1/33
Sulphur dioxide	4–5	½–1	....	1/10
Hydrogen sulphide	10–30	5–7	2–3	1
Carbon bisulphide	....	11	5	½
Phosphene	20	4–6	1–2	....
Arsine	2½	½	½	....
Phosgene	Over ¼	¼	....	1/100
Nitrous fumes	2½–7½	1–1½	....	⅓
Benzene	190	....	31–47	1½–3
Toluene and xylene	190	....	31–47	....
Aniline	....	....	1–1½	1/10
Nitrobenzene	....	....	1/100	1/500
Carbon tetrachloride	480	240	40	1
Chloroform	250	140	50	2
Tetrachlorethane	73	....	....	1/10
Trichlorethylene	370	....	....	....
Methyl chloride	1500–3000	200–400	70	5–10
Methyl bromide	200–400	20–40	10	2
Lead dust	....	....	....	0.15 mg/cu m
Quartz dust	....	....	....	1 mg/cu m

[a]Adapted from Y Henderson and H Haggard (See *Noxious Gases, 1927,* and *Lessons Learned from Industrial Gases and Fumes, Institute of Chemistry of Great Britain and Ireland,* London, 1930)

engineer will design equipment using these bench marks as the upper limits of pollution. In general it is good practice to avoid recirculation of air which contains originally toxic substances. Obviously there may be exceptions to this rule, but it is one which is generally being followed in current practice.

Bronchitis is the chief condition associated with exposure to thick dust, and follows upon inhalation of practically any kind of insoluble and non-colloidal dust. Atmospheric dust in itself cannot be blamed for causing tuberculosis, but it may aggravate the disease once it has started.[2]

[2]Physiological Response of the Peritoneal Tissue to Dusts Introduced as Foreign Bodies, by Miller and Sayers (*U. S. Public Health Reports,* 49 80, 1934).

The sulphurous fumes and tarry matter in smoke are more dangerous than the carbon. In foggy weather the accumulation of these substances in the lower strata may be such as to cause irritation of the eyes, nose, and respiratory passages, leading to asthmatic breathing and bronchitis and, in extreme cases, to death. The Meuse Valley fog disaster will probably become a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulphur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons.

Carbon monoxide from automobiles and from chimney gases constitutes another important source of aerial pollution in busy cities. During heavy traffic hours and under atmospheric conditions favorable to concentration, the air of congested streets is found to contain enough *CO* to menace the health of those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. In open air under ordinary conditions the concentration of *CO* in city air is insufficient to affect the average city dweller or pedestrian.

Occlusion of Solar Radiation

The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[3] by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. In New York City[4] a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

Recent studies[5] in Pittsburgh indicate that heavy smoke pollution is definitely unhealthful. Heretofore adequate proofs on this point were lacking.

The aesthetic and economic objections to air pollution are so definite, and the effect of air-borne pollen can be shown so readily as the cause of hay fever and other allergic diseases, that means and expenses of prevention or elimination of this pollution are justified.

SMOKE AND AIR POLLUTION ABATEMENT

Successful abatement of atmospheric pollution requires the combined efforts of the combustion engineer, the public health officer, and the public itself. The complete electrification of industry and railroads, and the separation of industrial and residential communities would aid materially in the effective solution of the problem.

In the large cities where the nuisance from smoke, dust and cinders is the most serious, limited areas obtain some relief by the use of district heating. The boilers in these plants are of large size designed and operated to burn the fuel without smoke, and some of them are equipped with dust catching devices. The gases of combustion are usually discharged at

[3]Effects of Atmospheric Pollution Upon Incidence of Solar Ultra-Violet Light, by J H Shrader, M H Coblentz and F A Korff (*American Journal of Public Health*, p 7, Vol 19, 1929)

[4]Studies in Illumination, by J E Ives (*U S Public Health Service Bulletin* No 197, 1930).

[5]Pneumoconiosis in the Pittsburgh district, Based on a Study of 2,500 Post Mortem Examinations made in Pittsburgh Hospitals, by Schnurer et al (*Journal Industrial Hygiene*, 17 294, March, 1935)

a much higher level than is possible in the case of buildings that operate their own boiler plants.

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals. (See Chapter 9.)

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher gravity oil always produces less smoke than a high volatile coal or low gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is necessary.

Legislative measures at the present time are largely concerned with the smoke discharged from the chimneys of boiler plants. Practically all of the ordinances limit the number of minutes in any one hour that smoke of a specified density, as measured by comparison with a Ringelmann Chart (Chapter 44), may be discharged.

These ordinances do not cover the smoke discharged at low levels by automobiles, and, although they have been instrumental in reducing the smoke emitted by boiler plants, they have, in many instances, increased the output of chimney dust and cinders due to the use of more excess air and to greater turbulence in the furnaces.

Legislative measures in general have not as yet covered the noxious

gases, such as sulphur dioxide and sulphuric acid mist, which are discharged with the gases of combustion. Where high sulphur coals are burned, these sulphur gases present a serious problem.

DUST AND CINDERS

The impurities in the air other than smoke come from so many sources that they are difficult to control. Only those which are produced in large quantities at a comparatively few points, such as the dust, cinders and fly-ash discharged to the atmosphere along with the gases of combustion from burning solid fuel, can be readily controlled.

Dusts and cinders in flue gas may be caught by various devices on the market, such as fabric filters, dust traps, settling chambers, centrifugal separators, electrical precipitators, and gas scrubbers, described in Chapter 26.

The cinder particles are usually larger in size than the dust particles; they are gray or black in color, and are abrasive. Being of a larger size, the range within which they may annoy is limited.

The dust particles are usually extremely fine; they are light gray or yellow in color, and are not as abrasive as cinder particles. Being extremely fine, they are readily distributed over a large area by air currents.

The nuisance created by the solid particles in the air is dependent on the size and physical characteristics of the individual particles. The difficulty of catching the dust and cinder particles is principally a function of the size and specific gravity of the particles.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal hasbeen removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coals when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

NATURE'S DUST CATCHER

Nature has provided means for catching solid particles in the air and depositing them upon the earth. A dust particle forms the nucleus for each rain drop and the rain picks up dust as it falls from the clouds to the earth. In fact, without dust in the air to form the nuclei for rain drops it would never rain, and the earth would be continually enveloped in a cloud of vapor. However, it was found in recent studies[6] that rain was not a good air cleaner of the material below about 0.7 micron.

[6]Atmospheric Pollution of American Cities for the years 1931-1933, by J. E. Ives et al (*U. S. Public Health Bulletin* No. 224, March, 1936)

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PROBLEMS IN PRACTICE

1 ● Classify the detrimental aspects of air pollution as it affects large industrial communities.

Air pollution may be classified (*a*) medical, as it affects the physiological functions of people, (*b*) botanical, as it affects vegetation, trees, plants, shrubs and flowers, and (*c*) physical, as it affects the discoloration and deterioration of buildings, and the nuisance of soiled interior furnishings, clothes, merchandise, etc.

2 ● Distinguish between dusts, fumes, and smokes.

Solid particles ranging in size from 1.0 micron to 150 microns are called *dusts* (micron = $\frac{1}{25,000}$ in.).

Particles resulting from sundry chemical reactions and ranging from 0.1 to 1.0 micron in size are called *fumes.*

Carbon particles less than 0.1 micron in size which generally arise from the incomplete combustion of such materials as coal, oil, or tobacco are called *smokes.*

3 ● What are some of the more important physical properties of these various groups of foreign bodies which are of importance in ventilation?

In slowly moving air, dusts tend to settle out by gravity without agglomerating to form larger particles, fumes have the tendency to form larger particles which will settle when they attain the size of approximately 1.0 micron; while smokes tend to diffuse and remain in the air as permanent impurities

4 ● Why is atmospheric pollution an important engineering problem?

a. Certain impurities, when present in too great concentrations, cause ill health or even death.

b. High concentrations of solids occlude solar radiations

c. Some materials cause permanent injury to parts of buildings, as sulphur fumes corrode exposed metal.

d. Extra cleaning expense is incurred in dusty localities.

e. Internal combustion engines are damaged by abrasive dusts.

5 ● How may the hazards of dust-producing industrial operations best be curtailed?

By providing mechanical exhaust ventilation sufficient to keep dust concentration at a safe level (see Table 3) and then removing foreign bodies to reduce the pollution of outside air.

6 ● How may the pollution of the atmosphere be lessened?

By compelling industrial plants to install dust catching and smoke controlling devices In many cities the domestic heating plant is one of the most serious offenders, but these plants are too small to justify the installation of dust catchers. Public education in improved firing methods would be of considerable help in this field.

7 ● What size particles are detrimental to health?

While fairly large particles may enter the upper air passages, those found in the lungs are seldom more than 10 microns in size, and comparatively few of them are more than 5 microns. It is agreed that particles between ½ and 2 microns may be harmful, some authorities place the upper limit at about 5 microns, and some incline to extend the lower limit to 0.1 of a micron.

8 ● Is the shape of the particle of any significance?

Hard particles with sharp corners or edges have a cutting effect on the delicate mucous membranes of the upper respiratory tract which may lower the resistance of the nose and throat to acute infections. This is aggravated by the irritating effects of some chemical compounds which may be taken in with the air and which act to reduce resistance.

9 ● What are the principal meteorological effects of smoke and dust?

a. The reduction in the amount of light received. Measurements have shown that visible light may be as much as 50 per cent less intense in a smoky section of a city than in a section that is free from smoke Ultra-violet light is reduced as much or more, and in some cases is cut out entirely for a time.

b. Smoke and dust aid in the formation and prolongation of fogs. City fogs accumulate smoke and become darker in color and very objectionable. The sun requires a longer time to disperse them, and when the water is evaporated, there is a rain of smoke and soot particles that have been entrained.

10 ● Why has not smoke abatement been more effective?

Because communities have not been made sufficiently aware of the possibilities of burning high volatile fuels smokelessly and of separating cinder and ash from the stack gases to a degree that will prevent a nuisance.

11 ● Is the abatement of dust and cinders important?

Yes. Only a small percentage of the solid emission from stacks is smoke, in the accepted popular sense, the remainder is fly-ash and cinders. While black smoke is disagreeable and its tarry matter and carbon particles soil anything with which they come in contact, the cinders and some of the ash are hard and destructive. They also, together with dusts from industrial processes, make up the irritating, air-borne solids that are breathed by individuals not working in a dusty mill or factory.

12 ● Are air-borne impurities causative factors in hay fever, bronchial asthma, and allergic disorders?

Yes Recent medical investigations indicate that 90 per cent of seasonal hay fever and 40 per cent of bronchial asthma are caused by air-borne pollens, tree dusts, and other allergic irritants.

13 ● Name some essential requirements for the smokeless combustion of fuels.

Time, temperature, and turbulence. A study of these factors is usually of value in overcoming a smoke nuisance.

14 ● What is the Ringelmann Chart Method of comparing smoke densities?

See Chapter 44. The Ringelmann Chart consists of four cards ruled with lines having different degrees of blackness. These cards, together with a white card and a black one, are hung in a horizontal row 50 ft from the observer At this distance the lines become invisible and the cards appear to be different shades of gray, ranging from white to black. The observer, by matching the cards against the shades of smoke coming from a stack, is able to estimate the blackness of the smoke as compared with the chart.

Chapter 5

HEAT TRANSMISSION COEFFICIENTS AND TABLES

Methods of Heat Transfer, Coefficients, Conductivity of Homogeneous Materials, Surface Conductance Coefficients, Air Space Conductance, Practical Coefficients, Table of Conductivities and Conductances, Tables of Over-all Coefficients of Heat Transfer for Typical Building Construction, Combined Coefficients of Transmission

IN order to maintain comfortable living temperatures within a building it is necessary to supply heat at the same rate that it is lost from the building. The loss of heat occurs in two ways, by direct transmission through the various parts of the structure and by air leakage or filtration between the inside and outside of the building. The purpose of this chapter is to show methods of calculation and to give practical transmission coefficients which may be applied to various structures to determine the heat loss by direct transmission. The amount lost by air filtration is determined by different methods, as outlined in Chapter 6, and must be added to that lost by direct transmission to obtain the total heating plant requirements.

METHODS OF HEAT TRANSFER

Heat transmission between the air on the two sides of a structure takes place by three methods, namely, radiation, convection and conduction. In a simple wall built up of two layers of homogeneous materials separated to give an air space between them, heat will be received from the high temperature surface by radiation, convection and conduction. It will then be conducted through the homogeneous interior section by conduction and carried across to the opposite surface of the air space by radiation, conduction and convection. From here it will be carried by conduction through to the outer surface and leave the outer surface by radiation, convection and conduction. The process of heat transfer through a built-up wall section is complicated in theory, but in practice it is simplified by dividing a wall into its component parts and considering the transmission through each part separately. Thus the average wall may be divided into external surfaces, homogeneous materials and interior air spaces. Practical heat transmission coefficients may be derived which will give the total heat transferred by radiation, conduction and convection through any of these component parts and if the selection and method of applying these individual coefficients is thoroughly understood it is usually a comparatively simple matter to calculate the over-all heat transmission coefficient for any combination of materials.

HEAT TRANSFER COEFFICIENTS

The symbols representing the various coefficients of heat transmission and their definitions are as follows:

U = thermal transmittance or over-all coefficient of heat transmission, the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 deg F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

k = thermal conductivity; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 deg F between the two surfaces of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

C = thermal conductance; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a non-homogeneous material for the thickness or type under consideration for a difference in temperature of 1 deg F between the two surfaces of the material Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plaster board and hollow clay tile.

f = film or surface conductance, the amount of heat expressed in Btu transmitted by radiation, conduction and convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg F between the surface and the surrounding air. To differentiate between inside and outside wall (or floor, roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance.

a = thermal conductance of an air space, the amount of heat expressed in Btu transmitted by radiation, conduction and convection in one hour through an area of 1 sq ft of an air space for a temperature difference of 1 deg F. The conductance of an air space depends on the mean absolute temperature, the width, the position and the character of the materials enclosing it.

R = resistance or resistivity which is the reciprocal of transmission, conductance, or conductivity, *i.e.:*

$\frac{1}{U}$ = over-all or air-to-air resistance.

$\frac{1}{k}$ = internal resistivity.

$\frac{1}{C}$ = internal resistance.

$\frac{1}{f}$ = film or surface resistance.

$\frac{1}{a}$ = air-space resistance.

As an example in the application of these coefficients assume a wall with over-all coefficient U. Then,

$$H = A\,U\,(t - t_o) \tag{1}$$

where

H = Btu per hour transmitted through the material of the wall, glass, roof or floor.

A = area in square feet of wall, glass, roof, floor, or material, taken from building plans or actually measured. (Use the net inside or heated surface dimensions in all cases).

$t - t_o$ = temperature difference between inside and outside air, in which t must always be taken at the proper level. Note that t may not be the *breathing-line* temperature in all cases.

If the heat transfer between the air and the inside surface of the wall is being considered, then,

$$H = A f_i (t - t_i) \tag{2}$$

where

f_i = inside surface conductance.

t and t_i = the temperatures of the inside air and the inside surface of the wall respectively.

In practice it is usually the over-all heat transmission coefficient that is required. This may be determined by a test of the complete wall, or it may be obtained from the individual coefficients by calculation. The simplest method of combining the coefficients for the individual parts of the wall is to use the reciprocals of the coefficients and treat them as resistance units. The total over-all resistance of a wall is equal numerically to the sum of the resistances of the various parts, and the reciprocal of the over-all resistance is likewise the over-all heat transmission coefficient of the wall. For a wall built up of a single homogeneous material of conductivity k and x inches thick the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} \tag{3}$$

If the coefficients f_i, f_o and k, together with the thickness of the material x are known, the over-all coefficient U may be readily calculated as the reciprocal of the total heat resistance.

For a compound wall built up of three homogeneous materials having conductivities k_1, k_2 and k_3 and thicknesses x_1, x_2 and x_3 respectively, and laid together without air spaces, the total resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} \tag{4}$$

For a wall with air space construction consisting of two homogeneous materials of thicknesses x_1 and x_2 and conductivities k_1 and k_2, respectively, separated to form an air space of conductance a, the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o} \tag{5}$$

Likewise any combination of homogeneous materials and air spaces can be put into the wall and the over-all resistance of the combination may be calculated by adding the resistances of the individual sections of the wall. In certain special forms of construction such as tile with irregular air spaces it is necessary to consider the conductance C of the unit as built instead of the unit conductivity k, and the resistance of the section is equal to $\frac{1}{C}$. The method of calculating the over-all heat transmission coefficient for a given wall is comparatively simple, but the selection of the proper coefficients is often complicated. In some cases the construction of the wall is such that the substituting of coefficients in the accepted formula will give erroneous results. This is the case with irregular cored

out air spaces in concrete and tile blocks, and walls in which there are parallel paths for heat flow through materials having different heat resistances. In such cases it is necessary to resort to test methods to check the calculations, and in practically all cases it has been necessary to determine fundamental coefficients by test methods.

Conductivity of Homogeneous Materials

The thermal conductivity of homogeneous materials is affected by several factors. Among these are the density of the material, the amount of moisture present, the mean temperature at which the coefficient is determined, and for fiberous materials the arrangement of fiber in the material. There are many fiberous materials used in building construction and considered as homogeneous for the purpose of calculation, whereas they are not really homogeneous but are merely considered so as a matter of convenience. In general, the thermal conductivity of a material increases directly with the density of the material, increases with the amount of moisture present, and increases with the mean temperature at which the coefficient is determined. The rate of increase for these various factors is not the same for all materials, and in assigning proper coefficients one should make certain that they apply for the conditions under which the material is to be used in a wall. Failure to do this may result in serious errors in the final coefficients.

Surface Conductance Coefficients

Heat is transmitted to or from the surface of a wall by a combination of radiation, convection and conduction. The coefficient will be effected by any factor which has an influence on any one of these three methods of transfer. The amount of heat by radiation is controlled by the character of the surface and the temperature difference between it and the surrounding objects. The amount of heat by conduction and convection is controlled largely by the roughness of the surface, by the air movement over the surface and by the temperature difference between the air and the surface. Because of these variables the surface coefficients may be subject to wide fluctuations for different materials and different conditions. The inside and outside coefficients f_i and f_o are in general affected to the same extent by these various factors and test coefficients determined for inside surfaces will apply equally well to outside surfaces under like conditions. Values for f_i in still and moving air at different mean temperatures have been determined for various building materials at the University of Minnesota under a cooperative agreement with the Society.[1]

The relation obtained between surface conductances for different materials at mean temperatures of 20 F is shown in Fig. 1. These values were obtained with air flow parallel to the surface and from other tests in which the angle of incident between the direction of air flow and the surface was varied from zero to 90 deg it would appear that these values might be lowered approximately 15 per cent for average conditions. While for average building materials there is a difference due to mean temperature, the greatest variation in these coefficients is caused by the character of the surface and the wind velocity. If other surfaces, such as

[1]A S H V E RESEARCH REPORT No 869—Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F B Rowley, A B Algren and J L Blackshaw (A S H V E TRANSACTIONS, Vol 36, 1930, p 429)

aluminum foil with low emissivity coefficients were substituted, a large part of the radiant heat would be eliminated. This would reduce the total coefficient for all wind velocities by about 0.7 Btu and would make but very little difference for the higher wind velocities. In many cases in building construction the heat resistance of the internal parts of the wall is high as compared with the surface resistance and the surface factors become of small importance. In other cases such as single glass windows the surface resistances constitute practically the entire resistance of the structure, and therefore become important factors. Due to the wide variation in surface coefficients for different conditions their selection for

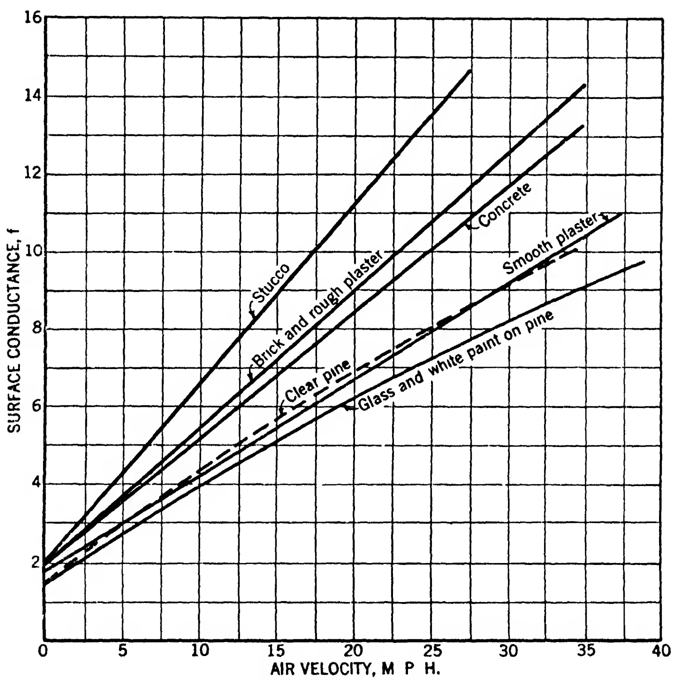


FIG. 1. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT 20 F MEAN TEMPERATURE

a practical building becomes a matter of judgment. In calculating the over-all coefficients for the walls of Tables 3 to 12, 1.65 has been selected as an average inside coefficient and 6.0 as an average outside coefficient for a 15-mile wind velocity. In special cases where surface coefficients become important factors in the over-all rate of heat transfer more selective coefficients may be required.

Air Space Conductance

Heat is conducted across an air space by a combination of radiation, conduction and convection. The amount of heat by radiation is governed largely by the nature of the surface and the temperature difference between the boundary surfaces of the air space. Conduction and con-

vection are controlled largely by the width and shape of the air space and the roughness of the boundary surfaces. The thermal resistances of air spaces bounded by extended parallel surfaces perpendicular to the direction of heat flow and at different mean temperatures have been determined for average building materials at the University of Minnesota in a cooperative research program with the Society.

The values given in Table 1 show the results of this study and apply to air spaces bounded by such materials as paper, wood, plaster, etc., having emissivity coefficients of from 0.9 to 0.95. The conductivity coefficients decrease with air space width until a width of about ¾ in. has been reached, after which the width has but very little effect. In these

TABLE 1. CONDUCTANCES OF AIR SPACES[a] AT VARIOUS MEAN TEMPERATURES

Mean Temp Deg Fahr	Conductances of Air Spaces for Various Widths in Inches						
	0 128	0 250	0 364	0 493	0.713	1 00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1 070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1 152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2 730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1 270
90	2.908	1.757	1.547	1.433	1.340	1.320	1 310
100	2.990	1.813	1.600	1 486	1 380	1.362	1.350
110	3 078	1 870	1 650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3 250	1 980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1 680	1 550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

[a]Thermal Resistance of Air Spaces by F B Rowley and A B Algren (A S H V E. TRANSACTIONS, Vol 35, 1929, p 165).

coefficients radiation is a large factor, and if surfaces with low emissivity coefficients are substituted for ordinary building materials the total amount of radiant heat will be reduced. The reduction in radiant heat caused by the low emissivity surface is independent of width of air space. Air spaces properly formed in combination with metallic surfaces such as aluminum foil, coated sheet steel, and other materials having a reflective surface, possess heat repelling characteristics. Values of air spaces lined with aluminum foil on one or both sides for widths of ⅜ in. and ¾ in. are shown in Table 2 of conductivities. A low emissivity coefficient is dependent on the permanency of the reflective surface. If a bright clean surface is covered with a thin layer of corrosive material its reflectivity is appreciably reduced[2].

In comparing the conductance coefficients for air spaces with and without bright metallic surface lining it should be noted that the reduction in heat transfer is substantially as great when one surface is lined as it is when both surfaces are lined. The reason for this is that practically 95 per cent of the total radiant heat is intercepted by one surface lining

[2]Aluminum Foil Insulation (*National Bureau of Standards Letter Circular* No. LC465, June, 1936).

and there is but a small amount left to be stopped by the second surface lining. The effect of any low emissivity surface in stopping the transmission of radiant heat is the same regardless of whether it is on the high or low temperature side of the air space. For materials such as aluminum paint or bronze paint which stop only a small percentage of radiant heat there is a greater percentage of gain by addition of a second surface lining.

PRACTICAL COEFFICIENTS

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types of construction without the necessity of making tests on the individual material or combination of materials. In Table 2 coefficients are given for a group of materials which have been selected from various sources. Wherever possible the properties of material and conditions of tests are given. However, in selecting and applying these values to any construction a reasonable amount of caution is necessary; variations will be found in the coefficients for the same materials, which may be partly due to different test methods used, but which are largely due to variations in materials. The recommended coefficients which have been used for the calculation of over-all coefficients as given in Tables 3 to 12 are marked by an asterisk.

It should be recognized in these tables of calculated coefficients that space limitations will not permit the inclusion of all the combinations of materials that are used in building construction and the varied applications of insulating materials to these constructions. Typical examples are given of combinations frequently used, but any special construction not given in Tables 3 to 12 can generally be computed by using the conductivity values given in Table 2 and the fundamental heat transfer formulae. For example, the tabulation of all of the values for multiple layers of insulating materials would present extensive and detailed problems of calculations for the varied application combinations, but the engineer having the fundamental conductivity values can quickly obtain the proper coefficients.

Attention is called to the fact that the conductivity values per inch of thickness do not afford a true basis for comparison between insulating materials as applied, although they are frequently used for that purpose. The value of an insulating material is measured in terms of its heat resistance, which not only depends upon the thermal conductivity coefficient per inch but also upon the thickness as installed and the manner of installation. For instance the material having a coefficient of 0.50 and 1 in. thick is equal in value to a material having a coefficient of 0.25 and a thickness of ½ in. Certain types of blanket installations are designed to be installed between the studs of a frame building in such manner as to give two air spaces. In order to get the full value of such materials they should be so installed that each air space is approximately 1 in. or more in thickness and the air spaces should be sealed at the top and bottom to prevent the circulation of air from one space to the other. Another common error in installing such a material is to nail the blanket on the outside of the studs underneath the sheathing, in which case one air space is lost and also the thickness of the insulating material is materially reduced at the studs. There are certain other types of insulation which are very porous, allowing air circulation within the material if not

properly installed. The architect or engineer must carefully evaluate the economic considerations involved in the selection of an insulating material as adapted to various building constructions. Lack of good judgment in the intelligent choice of an insulating material, or its improper installation, frequently represents the difference between good or unsatisfactory results. Refer to Chapter 7 for a discussion of wall condensation.

Computed Transmission Coefficients

Computed heat transmission coefficients of many common types of building construction are given in Tables 3 to 13, inclusive, each construction being identified by a serial number. For example, the coefficient of transmission (U) of an 8-in. brick wall and ½ in. of plaster is 0.46, and the number assigned to a wall of this construction is *1-B*, Table 3.

Example 1. Calculate the coefficient of transmission (U) of an 8-in brick wall with ½ in. of plaster applied directly to the interior surface, based on an outside wind exposure of 15 mph. It is assumed that the outside course is of hard (high density) brick having a conductivity of 9 20, and that the inside course is of common (low density) brick having a conductivity of 5 0, the thicknesses each being 4 in The conductivity of the plaster is assumed to be 3 3, and the inside and outside surface coefficients are assumed to average 1.65 and 6 00, respectively, for still air and a 15 mph wind velocity.

Solution. k (hard high density brick) $= 9.20$, $x = 4.0$ in ; k (common low density brick) $= 5.0$, $x = 4.0$ in., k (plaster) $= 3.3$, $x = \frac{1}{2}$ in , $f_i = 1.65$, $f_o = 6.0$ Therefore,

$$U = \frac{1}{\frac{1}{6.0} + \frac{4.0}{9.20} + \frac{4.0}{5.0} + \frac{0.5}{3.3} + \frac{1}{1.65}}$$

$$= \frac{1}{0.167 + 0.435 + 0.80 + 0.152 + 0.606}$$

= 0 46 Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides.

The coefficients in the tables were determined by calculations similar to those shown in Example 1, using Fundamental Formulae 3, 4 and 5 and the values of k (or C), f_i, f_o and a indicated in Table 2 by asterisks. In computing heat transmission coefficients of floors laid directly on the ground (Table 10), only one surface coefficient (f_i) is used. For example, the value of U for a 1-in. yellow pine floor (actual thickness, 25/32 in.) placed directly on 6-in. concrete on the ground, is determined as follows:

$$U = \frac{1}{\frac{1}{1.65} + \frac{0.781}{0.80} + \frac{6.0}{12.0}}$$ = 0 48 Btu per hour per square foot per degree difference in temperature between the ground and the air immediately above the floor.

Rigid insulation refers to the so-called board form which may be used structurally, such as for sheathing. Flexible insulation refers to the blankets, quilts or semi-rigid types of insulation.

Actual thicknesses of lumber are used in the computations rather than nominal thicknesses. The computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Since no reliable figures are available concerning the conductivity of Spanish and French clay roofing tile, of which there are many varieties, the figures for such types of roofs were taken the same as for slate roofs, as

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness unless otherwise indicated

Material	Description					Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
	Cement	Fine Aggregate 0-No 4	Coarse Aggregate No 4-1½	Slump	Per Cent Voids					
Sand and Gravel, Concrete	1	2 00	2 75	0	11 5	144 7	75 06	13 10	0 08	(4)
	1	2 75	4 50	0	10 9	145 7	74 77	12 90	0 08	(4)
	1	3 50	5 50	0	11 2	144 5	75 00	13 20	0 08	(4)
	1	2 00	2 75	5	13 9	142 5	75 50	12 10	0 08	(4)
	1	2 00	2 75	5	13 9	142 5	74 74	12 40	0 08	(4)
	1	2 75	4 50	5	14 6	141 1	73 30	12 40	0 08	(4)
	1	2 75	4 50	5	14 6	141 1	74 89	12 10	0 08	(4)
	1	3 50	5 50	5	14 7	139 2	74 50	12 85	0 08	(4)
	1	3 50	5 50	5	14 7	139 2	75 15	12 50	0 08	(4)
	Avg Value for Sand and Gravel Concrete					142 3	...	12 62	.	..
Limestone Concrete	1	2 00	2 75	0	16 6	135 3	74 87	11 20	0 09	(4)
	1	2 75	4 50	0	15 4	137 8	75 18	12 00	0 08	(4)
	1	3 50	5 50	0	16 3	136 4	74 75	11 50	0 09	(4)
	1	2 00	2 75	3	20 9	130 1	74 85	10 50	0 10	(4)
	1	2 75	4 50	3	23 4	126 0	74 45	10 00	0 10	(4)
	1	3 50	5 50	3	23 4	127 3	75 26	9 79	0 10	(4)
	Avg Value for Limestone Concrete .					132 15	.	10 83		
Cinder Concrete	1	2 00	2 75	0	18 2	103 6	75 26	4 63	0 22	(4)
	1	2 75	4 50	0	19 9	98 7	75 71	4 30	0 23	(4)
	1	3 50	5 50	0	21 4	92 0	75 72	3 73	0 27	(4)
	1	2 00	2 75	3	22 8	101 4	74 95	4 89	0 20	(4)
	1	2 75	4 50	3	26 0	94 0	75 20	4.38	0 23	(4)
	1	3 50	5 50	3	24 4	94 4	75 55	4.24	0 24	(4)
	Avg Value for Cinder Concrete					97 35	.	4 86	....	
Haydite	1	2 00	2 75	0	18 0	80 7	74 82	4 15	0 25	(4)
	1	2 75	4 50	0	19 8	75 0	75 75	3 78	0 26	(4)
	1	3 50	5 50	0	21 8	71 7	74 82	3 67	0 27	(4)
	1	2 00	2 75	4	21 2	78 8	74 76	4 38	0 23	(4)
	1	2 75	4 50	4	22 2	72 4	75 39	3 89	0 26	(4)
	1	2 75	4 50	4	22 2	72 4	75 49	3 86	0 26	(4)
	1	3 50	5 50	4	23 9	71 0	75.46	4 00	0 25	(4)
	Avg Value for Haydite . . .					74 57		3 96		

AUTHORITIES

[1]U S Bureau of Standards, tests based on samples submitted by manufacturers

[2]A C Willard, L C Lichty, and L A Harding tests conducted at the University of Illinois

[3]J C Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers

[4]F B Rowley, tests conducted at the University of Minnesota

[5]A S H V E Research Laboratory

[6]E A Allcut, tests conducted at the University of Toronto

[7]Lees and Charlton

[8]G B Wilkes and C M F Peterson, tests conducted at the Massachusetts Institute of Technology

*Recommended conductivities and conductances for computing heat transmission coefficients

†For thickness stated or used on construction, not per 1-in thickness

[a]For additional conductivity data see Chapters 3 and 15, 1937 *A S R E Data Book*

[b]If outside surface of block is painted with an impervious coat of paint add 0 07 to resistance for sand and gravel blocks Add 0 18 to resistance for cinder blocks Add 0 17 to resistance for haydite blocks

[c]Recommended value See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.

[d]See A S H V E RESEARCH REPORT No 915—Conductivity of Concrete, by F C Houghten and Carl Gutberlet (A S H V E TRANSACTIONS, Vol 38, 1932, p 47)

[e]The 6-in, 8-in, and 10-in hollow tile figures are based on two cells in the direction of heat flow The 12-in hollow tile is based on three cells in the direction of heat flow The 16-in hollow tile consists of one 10-in and one 6-in tile, each having two cells in the direction of heat flow

[f]Not compressed

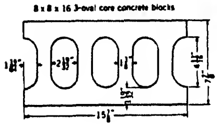
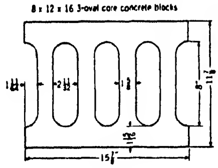
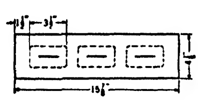
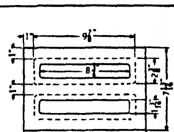
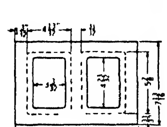
[g]Roofing, 0 15-in thick (1 34 lb per sq ft), covered with gravel (0 83 lb per sq ft), combined thickness assumed 0 25

[h]Values for air spaces or surfaces having an effective emissivity of e = 0 83 and for a temperature difference of 15 F

[i]Values for air spaces or surfaces having an effective emissivity of e = 0 05 and for a temperature difference of 15 F

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

Material	Description: Cement	Fine Aggregate 0-No 4	Coarse Aggregate No 4-½	Slump	Per Cent Voids	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
EXPANDED BURNED CLAY	1	8.00			18 4	57.9	75.57	2 28	0.44	(4)
STEAM TREATED LIMESTONE SLAG	1	7 00			27 1	74 6	74 49	2 27	0 44	(4)
PUMICE MINED IN CALIF	1	8 00			26.5	65.0	74 68	2 42	0 41	(4)
BY-PRODUCT OF MANUFACTURE OF PHOSPHATES	1	8 00	Fineness Modulus 3 75		25 5	86 6	74 62	3 19	0.31	(4)
	1	8.00			21.1	91 1	74 43	3 42	0.29	(4)
HAYDITE	1	8 50			21.8	67 1	75 89	2 89	0 35	(4)
	1	8.50			21.8	67 1	74 60	2 815	0.34	(4)
8 x 8 x 16 3-oval core concrete blocks 	Sand and gravel aggregate					126 4	40	0 900†	1 11	(4)
	Sand and gravel aggregate used for calculations							1 000†*	1 00	(4)
	Cores filled with 5 14 lb density cork						40	0 560†*	1 79	(4)
	Crushed limestone aggregate					134 3	40	0 856†*	1 17	(4)
	Cinder aggregate					86 2	40	0 577†	1 73	(4)
	Cinder aggregate used for calculations							0 600†*	1 66	(4)
	Cores filled with 69 7 lb density cinders						40	0 390†	2 56	(4)
	Cores filled with 5 12 lb density cork						40	0 250†*	4 00	(4)
	Cores filled with 14 2 lb density rock wool						40	0 266†*	3.76	(4)
	Haydite aggregate					67 7	40	0 495†*	2 02	(4)
	Cores filled with 5 06 lb density cork						40	0 206†	4 85	(4)
8 x 12 x 16 3-oval core concrete blocks 	Sand and gravel aggregate					124 9	40	0 777†	1 29	(4)
	Sand and gravel aggregate used for calculations							0 800†*	...	(4)
	Cinder aggregate					86 2	40	0 531†*	1 88	(4)
	Cores filled with 5.24 lb density cork						40	0 237†*	4.22	(4)
	Haydite aggregate					76 7	40	0 468†	2 13	(4)
	Cores filled with 5.6 lb density cork						40	0 168†*	5.94	(4)
	Cinder aggregate					100 0	40	1 00?†	1 00	(4)
	Double wall with 1 in air space between					100 0	40	0 358†	2 70	(4)
	1 in space filled with 9 97 lb density rock wool					100.0	40	0.204†	4.90	(4)
	5 x 8 x 12 block sand and gravel aggregate					133.7	40	0.380†	2.63	(4)
	5 x 8 x 12 block sand and gravel aggregate[b]					134 0	40	0 947†	1.06	(4)

For notes see Page 95.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS						
BRICK	Low density			5 00*	0 20	
	High density			9 20*	0 11	
BRICKWORK	Damp or wet			5 00[c]	0 20	(2)
CEMENT MORTAR	Typical			12 00*	0 08	
CONCRETE	Typical			12 00*	0 08	
	Various ages and mixes[d]			11 35 to 16 36		(5)
	Cellular	40 0	75	1 06	0 94	(3)
	Cellular	50 0	75	1 44	0 69	(3)
	Cellular	60 0	75	1 80	0 56	(3)
	Cellular	70 0	75	2 18	0 46	(3)
	Typical fiber gypsum, 87 5% gypsum and 12 5% wood chips	51 2	74	1 66*	0 60	(4)
	Special concrete made with an aggregate of hardened clay—1-2-3 mix	101 0	70	3 98	0 25	(3)
STONE	Typical			12 50*	0 08	
STUCCO	Typical			12 00*	0 08	
TILE	Typical hollow clay (4 in)			1 00†*	1 00	
	Typical hollow clay (6 in)[e]			0 64†*	1 57	
	Typical hollow clay (8 in)[e]			0 60†*	1 67	
	Typical hollow clay (10 in)[e]			0 58†*	1 72	
	Typical hollow clay (12 in)[e]			0 40†*	2 50	
	Typical hollow clay (16 in)[e]			0 31†*	3 23	
	Hollow clay (2 in) ½-in plaster both sides	120 0	110	1 00†	1 00	(2)
	Hollow clay (4 in) ½-in plaster both sides	127 0	100	0 60†	1 67	(2)
	Hollow clay (6 in) ½-in plaster both sides	124 3	105	0 47†	2 13	(2)
	Hollow gypsum (4 in)			0 46†*	2 18	
	Solid gypsum	51 8	70	1 66	0 60	(4)
	Solid gypsum	75 6	76	2 96	0 34	(4)
TILE OR TERRAZZO	Typical flooring			12 00*	0 08	
INSULATION—BLANKET OR FLEXIBLE TYPES						
FIBER	Typical			0 27*	3 70	
	Chemically treated wood fibers held between layers of strong paper[f]	3 62	70	0 25	4 00	(3)
	Eel grass between strong paper[f]	4 60	90	0 26	3 85	(1)
	" " " " "	3 40	90	0 25	4 00	(1)
	Flax fibers between strong paper[f]	4 90	90	0 28	3 57	(1)
	Chemically treated hog hair between kraft paper[f]	5 76	71	0 26	3 85	(3)
	Chemically treated hog hair between kraft paper and asbestos paper[f]	7 70	71	0 28	3 57	(3)
	Hair felt between layers of paper[f]	11 00	75	0 25	4 00	(3)
	Kapok between burlap or paper[f]	1 00	90	0 24	4 17	(1)
	Jute fiber[f]	6 70	75	0 25	4 00	(3)
	Ground paper between two layers, each ⅜-in thick made up of two layers of kraft paper (sample ¾-in thick)	12 1	75	0 40†	2 50	(4)
INSULATION—SEMI-RIGID TYPE						
FIBER	Felted cattle hair[f]	13 00	90	0.26	3 84	(1)
	" " "	11 00	90	0 26	3 84	(1)
	Flax[f]	12 10	70	0 30	3 33	(3)
	Flax and rye[f]	13 60	90	0 32	3 12	(1)
	Felted hair and asbestos[f]	7 80	90	0 28	3 57	(1)
	75% hair and 25% jute[f]	6 30	90	0 27	3 70	(1)
	50% hair and 50% jute[f]	6 10	90	0 26	3 85	(1)
	Jute[f]	6 70	75	0 25	4 00	(3)
	Felted jute and asbestos[f]	10 00	90	0 37	2 70	(1)
	Compressed peat moss	11 00	70	0.26	3 84	(3)
INSULATION—LOOSE FILL OR BAT TYPE						
FIBER	Made from ceiba fibers[f]	1 90	75	0 23	4.35	(3)
	" " " "	1 60	75	0 24	4.17	(3)

For notes see page 95

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
INSULATION—LOOSE FILL OR BAT TYPE—Continued						
FIBER	Fibrous material made from dolomite and silica	1.50	75	0.27	3.70	(3)
	Fibrous material made from slag	9.40	103	0.27	3.70	(1)
GLASS WOOL	Fibrous material 25 to 30 microns in diameter, made from virgin bottle glass	1.50	75	0.27	3.70	(3)
GRANULAR	Made from combined silicate of lime and alumina	4.20	72	0.24	4.17	(3)
	Made from expanded aluminum-magnesium silicate	6.32	86	0.29	3.45	(3)
GYPSUM	Cellular, dry	30.00	90	1.00	1.00	(1)
	“ “	24.00	90	0.77	1.30	(1)
	“ “	18.00	90	0.59	1.69	(1)
	“ “	12.00	90	0.44	2.27	(1)
	Flaked, dry and fluffy[f]	34.00	90	0.60	1.67	(1)
	“ “ “ “	26.00	90	0.52	1.92	(1)
	“ “ “ “	24.00	75	0.48*	2.08	(3)
	“ “ “ “	19.80	90	0.35	2.86	(1)
	“ “ “ “	18.00	75	0.34	2.94	(3)
MINERAL WOOL	All forms, typical	.	..	0.27*	3.70	.
REGRANULATED CORK	About 5/16-in particles	8.10	90	0.31	3.22	(1)
ROCK WOOL	Fibrous material made from rock	21.00	90	0.30	3.33	(1)
	“ “ “ “ “	18.00	90	0.29	3.45	(1)
	“ “ “ “ “	14.00	90	0.28	3.57	(1)
	“ “ “ “ “	10.00	90	0.27*	3.70	(1)
	Rock wool with a binding agent	14.50	77	0.33	3.03	(1)
	Rock wool with flax straw pulp, and binder	14.50	75	0.38	2.63	(3)
	Rock wool with vegetable fibers	11.50	72	0.31	3.22	(3)
SAWDUST	Various	12.00	90	0.41	2.44	(1)
SHAVINGS	Various from planer	8.80	90	0.41	2.44	(1)
	From maple beech and birch (coarse)	13.20	90	0.36	2.78	(1)
	Redwood bark	3.00	90	0.31	3.22	(1)
	Redwood bark	5.00	75	0.26	3.84	(3)
INSULATION—RIGID						
CORKBOARD	Typical	.	..	0.30*	3.33	...
	No added binder	14.00	90	0.34	2.94	(1)
	“ “ “	10.60	90	0.30	3.33	(1)
	“ “ “	7.00	90	0.27	3.70	(1)
	“ “ “	5.40	90	0.25	4.00	(1)
	Asphaltic binder	14.50	90	0.32	3.12	(1)
FIBER	Typical		..	0.33*	3.03	
	Chemically treated hog hair covered with film of asphalt	10.00	75	0.28	3.57	(3)
	Made from corn stalks	15.00	71	0.33	3.03	(3)
	“ “ exploded wood fibers	17.90	78	0.32	3.12	(4)
	“ “ hard wood fibers	15.20	70	0.32	3.12	(3)
	Insulating plaster 9/10-in thick applied to 3/8-in plaster board base	54.00	75	1.07†	0.93	(3)
	Made from licorice roots	16.10	81	0.34	2.94	(3)
	“ “ 85% magnesia and 15% asbestos	19.30	86	0.51	1.96	(1)
	“ “ shredded wood and cement	24.20	72	0.46	2.17	(3)
	“ “ sugar cane fiber	13.50	70	0.33	3.03	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.80	70	0.30	3.33	(3)
	Made from wheat straw	17.00	68	0.33	3.03	(3)
	“ “ wood fiber	15.90	72	0.33	3.03	(3)
	“ “ “ “	15.00	70	0.33	3.03	(3)
	“ “ “ “		52	0.33	3.03	(6)
	“ “ “ “	8.50	72	0.29	3.45	(3)
	“ “ “ “	15.20		0.33	3.03	(3)
	“ “ “ “	16.90	90	0.34	2.94	(1)
BUILDING BOARDS						
ASBESTOS	Compressed cement and asbestos sheets	123.00	86	2.70	0.37	(1)
	Corrugated asbestos board	20.40	110	0.48	2.08	(2)

For notes see Page 95

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
BUILDING BOARDS —Continued						
ASBESTOS	Pressed asbestos mill board	60.50	86	0.84	1.19	(1)
	Sheet asbestos	48.30	110	0.29	3.45	(2)
GYPSUM	Gypsum between layers of heavy paper	62.80	70	1.41	0.71	(3)
	Rigid, gypsum between layers of heavy paper (½-in thick)	53.50	90	2.60†	0.38	(1)
	Gypsum mixed with sawdust between layers of heavy paper (0.39-in thick)	60.70	90	3.60†	0.28	(1)
PLASTER BOARD	(⅜ in)			3.73†*	0.27	
	(½ in)			2.82†*	0.35	
ROOFING CONSTRUCTION						
ROOFING	Asphalt, composition or prepared	70.00	75	6.50†*	0.15	(3)
	Built up—⅜-in thick			3.53†*	0.28	
	Built up, bitumen and felt, gravel or slag surfaced[g]			1.33	0.75	(2)
	Plaster board, gypsum fiber concrete and 3-ply roof covering 2½ in thick	52.40	76	0.58†	1.72	(4)
SHINGLES	Asbestos	65.00	75	6.00†*	0.17	(3)
	Asphalt	70.00	75	6.50†*	0.15	(3)
	Slate	201.00		10.37*	0.10	(7)
	Wood			1.28†*	0.78	—
PLASTERING MATERIALS						
PLASTER	Cement			8.00	0.13	(2)
	Gypsum, typical			3.30*	0.30	—
	Gypsum and expanded aluminum-magnesium silicate, mix 4 to 1	39.9	75	0.85	1.18	(3)
	Thickness ⅜ in		73	8.80†	0.11	(4)
METAL LATH AND PLASTER	Total thickness ¾ in			4.40†*	0.23	
WOOD LATH AND PLASTER	⅜-in plaster, total thickness ¾ in		70	2.50†*	0.40	(4)
BUILDING CONSTRUCTIONS						
FRAME	1-in fir sheathing and building paper		30	0.86†*	1.16	(4)
	1-in fir sheathing, building paper, and yellow pine lap siding		20	0.50†*	2.00	(4)
	1-in fir sheathing, building paper and stucco		20	0.82	1.22	(4)
	Pine lap siding and building paper—siding 4 in wide		16	0.85†*	1.18	(4)
	Yellow pine lap siding			1.28†*	0.78	
FLOORING	Maple—across grain	40.00	75	1.20	0.83	(3)
	Battleship linoleum (¼-in)			1.36†*	0.74	
AIR SPACE AND SURFACE COEFFICIENTS						
AIR SPACES	Over ¾ in faced ordinary building materials		40	1.10†*	0.91	(4)
	3⅝ in faced ordinary building materials					
	Horizontal, heat flow upward, $e = 0.83^h$			1.32	0.76	(8)
	Vertical, $e = 0.83^h$			1.17	0.86	(8)
	Horizontal, heat flow downward, $e = 0.83^h$			0.94	1.06	(8)
	3⅝ in faced reflective building materials					
	Horizontal, heat flow upward, $e = 0.05^i$			0.56	1.79	(8)
	Vertical, $e = 0.05^i$			0.41	2.44	(8)
	Horizontal, heat flow downward, $e = 0.05^i$			0.17	5.88	(8)
SURFACES, ORDINARY	Still air (f_i), Avg			1.65†*	0.61	(4)
	Horizontal, heat flow upward, $e = 0.83(f_i)^h$			1.95†	0.51	(8)
	Vertical, $e = 0.83(f_i)^h$			1.52†	0.66	(8)
	Horizontal, heat flow downward, $e = 0.83(f_i)^h$			1.21†	0.83	(8)
	15 mph—(f_o)			6.00†*	0.17	(4)
SURFACES, ROUGH STUCCO	15 mph—(f_o)			9.00†	0.11	(4)
SURFACE, REFLECTIVE	Still air (f_i), Avg			0.80†*	1.25	(8)
	Horizontal, heat flow upward, $e = 0.05(f_i)^i$			1.16†	0.86	(8)
	Vertical, $e = 0.05(f_i)^i$			0.74†	1.35	(8)
	Horizontal, heat flow downward, $e = 0.05(f_i)^i$			0.44†	2.27	(8)

For notes see Page 95

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
AIR SPACES FACED WITH BRIGHT ALUMINUM FOIL	Air space, faced one side with bright aluminum foil over ¾-in wide		50	0.46†*	2.17	(4)
	Air space, faced one side with bright aluminum foil, ⅜-in wide		50	0.62†	1.61	(4)
	Air space, faced both sides with bright aluminum foil, over ¾-in wide		50	0.41†*	2.44	(4)
	Air space, faced both sides with bright aluminum foil ⅜-in wide		50	0.57†	1.75	(4)
	Air space divided in two with single curtain of bright aluminum foil (both sides bright) each space over ¾-in wide		50	0.23†*	4.35	(4)
	each space ⅜-in wide		50	0.31†	3.23	(4)
	Air space with multiple curtains of bright aluminum foil, bright on both sides, curtains more than ¾-in apart, air circulation between spaces prevented 2 curtains, forming 3 spaces		50	0.15†*	6.78	(4)
	3 curtains, forming 4 spaces		50	0.11†*	9.22	(4)
	4 curtains, forming 5 spaces		50	0.09†*	11.[illegible]6	(4)
SPACES FACED WITH NON-METALLIC REFLECTIVE SURFACE	Fabric with non-metallic reflective surface (1/32 in thick) placed in center of a 1½ in air space		70	0.33†	3.03	(3)
	Core of fiber board coated two sides with non-metallic reflective surface (⅜ in thick) placed in space having approximately ¾ in air space on each side	23.4	70	0.27†	3.70	(3)
	Fiber board coated one side with non-metallic reflective surface (3/16 in thick) placed in space having approximately ¾ in air space on each side		75	0.49†	2.04	(3)
	Air space divided in two with fabric faced both sides with non-metallic reflective surface, each space over ¾-in wide		40	0.33†	3.03	(4)
	Air space over ¾-in wide faced one side with non-metallic reflective surface		40	0.67†	1.49	(4)
WOODS (Across Grain)						
BALSA		20.0	90	0.58	1.72	(1)
		8.8	90	0.38	2.63	(1)
		7.3	90	0.33	3.03	(1)
CALIFORNIA REDWOOD	0% moisture	22.0	75	0.66	1.53	(4)
	0% "	28.0	75	0.70	1.43	(4)
	8% "	22.0	75	0.70	1.43	(4)
	8% "	28.0	75	0.75	1.33	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	28.0	75	0.80	1.25	(4)
CYPRESS		28.7	86	0.67	1.49	(1)
DOUGLAS FIR	0% moisture	26.0	75	0.61	1.64	(4)
	0% "	34.0	75	0.67	1.49	(4)
	8% "	26.0	75	0.66	1.52	(4)
	8% "	34.0	75	0.75	1.33	(4)
	16% "	26.0	75	0.76	1.32	(4)
	16% "	34.0	75	0.82	1.22	(4)
EASTERN HEMLOCK	0% moisture	22.0	75	0.60	1.67	(4)
	0% "	30.0	75	0.76	1.32	(4)
	8% "	22.0	75	0.63	1.59	(4)
	8% "	30.0	75	0.81	1.23	(4)
	16% "	22.0	75	0.67	1.49	(4)
	16% "	30.0	75	0.85	1.18	(4)
HARD MAPLE	0% moisture	40.0	75	1.01	0.99	(4)
	0% "	46.0	75	1.05	0.95	(4)
	8% "	40.0	75	1.08	0.93	(4)
	8% "	46.0	75	1.13	0.89	(4)
	16% "	40.0	75	1.15	0.87	(4)
	16% "	46.0	75	1.21	0.83	(4)

For notes see Page 95

TABLE 2. CONDUCTIVITIES (*k*) AND CONDUCTANCES (*C*) OF BUILDING MATERIALS AND INSULATORS[a]—Concluded

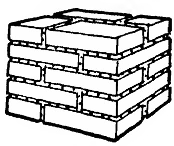
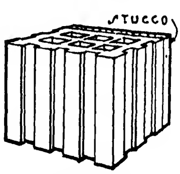
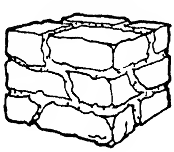
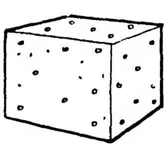
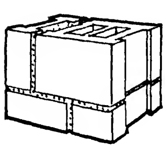
The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
WOODS—Continued						
LONGLEAF YELLOW PINE	0% moisture	30.0	75	0.76	1.32	(4)
	0% "	40.0	75	0.86	1.16	(4)
	8% "	30.0	75	0.83	1.21	(4)
	8% "	40.0	75	0.95	1.05	(4)
	16% "	30.0	75	0.89	1.12	(4)
	16% "	40.0	75	1.03	0.97	(4)
MAHOGANY		34.3	86	0.90	1.11	(1)
MAPLE		44.3	86	1.10	0.91	(1)
MAPLE OR OAK				1.15*	0.87	
NORWAY PINE	0% moisture	22.0	75	0.62	1.61	(4)
	0% "	32.0	75	0.74	1.35	(4)
	8% "	22.0	75	0.68	1.47	(4)
	8% "	32.0	75	0.83	1.21	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.91	1.10	(4)
RED CYPRESS	0% moisture	22.0	75	0.67	1.49	(4)
	0% "	32.0	75	0.79	1.27	(4)
	8% "	22.0	75	0.71	1.41	(4)
	8% "	32.0	75	0.84	1.19	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.90	1.11	(4)
RED OAK	0% moisture	38.0	75	0.98	1.02	(4)
	0% "	48.0	75	1.18	0.85	(4)
	8% "	38.0	75	1.03	0.97	(4)
	8% "	48.0	75	1.24	0.81	(4)
	16% "	38.0	75	1.07	0.94	(4)
	16% "	48.0	75	1.29	0.78	(4)
SHORTLEAF YELLOW PINE	0% moisture	26.0	75	0.74	1.35	(4)
	0% "	36.0	75	0.91	1.10	(4)
	8% "	26.0	75	0.79	1.27	(4)
	8% "	36.0	75	0.97	1.03	(4)
	16% "	26.0	75	0.84	1.19	(4)
	16% "	36.0	75	1.04	0.96	(4)
SOFT ELM	0% moisture	28.0	75	0.73	1.37	(4)
	0% "	34.0	75	0.88	1.14	(4)
	8% "	28.0	75	0.77	1.30	(4)
	8% "	34.0	75	0.93	1.08	(4)
	16% "	28.0	75	0.81	1.24	(4)
	16% "	34.0	75	0.97	1.03	(4)
SOFT MAPLE	0% moisture	36.0	75	0.89	1.12	(4)
	0% "	42.0	75	0.95	1.05	(4)
	8% "	36.0	75	0.96	1.04	(4)
	8% "	42.0	75	1.02	0.98	(4)
	16% "	36.0	75	1.01	0.99	(4)
	16% "	42.0	75	1.09	0.92	(4)
SUGAR PINE	0% moisture	22.0	75	0.54	1.85	(4)
	0% "	28.0	75	0.64	1.56	(4)
	8% "	22.0	75	0.59	1.70	(4)
	8% "	28.0	75	0.71	1.41	(4)
	16% "	22.0	75	0.65	1.54	(4)
	16% "	28.0	75	0.78	1.28	(4)
VIRGINIA PINE		34.3	86	0.96	1.04	(1)
WEST COAST HEMLOCK	0% moisture	22.0	75	0.68	1.47	(4)
	0% "	30.0	75	0.79	1.27	(4)
	8% "	22.0	75	0.73	1.37	(4)
	8% "	30.0	75	0.85	1.18	(4)
	16% "	22.0	75	0.78	1.28	(4)
	16% "	30.0	75	0.91	1.10	(4)
WHITE PINE		31.2	86	0.78	1.28	(1)
YELLOW PINE				1.00	1.00	(3)
YELLOW PINE OR FIR				0.80*	1.25	

For notes see Page 95

TABLE 3. COEFFICIENTS OF TRANSMISSION (*U*) OF MASONRY WALLS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF WALL	THICKNESS OF MASONRY (INCHES)	WALL No
	**Solid Brick** Based on 4-in hard brick and the remainder common brick	8 12 16	1 2 3
	**Hollow Tile** Stucco Exterior Finish The 8-in and 10-in tile figures are based on two cells in the direction of flow of heat The 12-in tile is based on three cells in the direction of flow of heat The 16-in tile consists of one 10-in tile and one 6-in tile each having two cells in the direction of heat flow.	8 10 12 16	4 5 6 7
	**Limestone or Sandstone**	8 12 16 24	8 9 10 11
	**Concrete (Monolithic)** These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.	6 10 16 20	12 13 14 15
	Cinder (Monolithic) Conductivity $k = 4.36$	6 10 16 20	16 17 18 19
	Haydite (Monolithic) Conductivity $k = 3.96$	6 10 16 20	20 21 22 23
	**Cinder Blocks** Cores filled with dry cinders, 69 7 lb per cu ft Cores filled with granulated cork, 5 12 lb per cu ft Cores filled with rock wool, 14 2 lb per cu ft Based on one air cell in direction of heat flow Cores filled with granulated cork, 5 24 lb per cu ft	8 8 8 8 12 12	24 25 26 27 28 29
	Concrete Blocks Cores filled with granulated cork, 5 14 lb per cu ft Based on one air cell in direction of heat flow	8 8 12	30 31 32
	Haydite Blocks Cores filled with granulated cork, 5 06 lb per cu ft.	8 8	33 34
	Haydite Blocks Cores filled with granulated cork, 5 6 lb per cu ft	12 12	35 36

[a]Computed from factors marked by * in Table 2

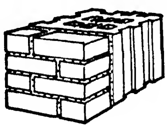
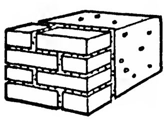
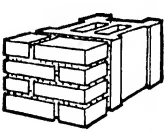
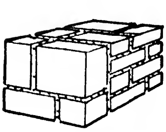
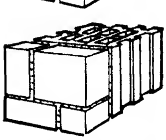
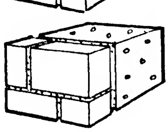
[b]Based on the actual thickness of 2-in furring strips.

INTERIOR FINISH											
UNINSULATED WALLS						INSULATED WALLS					
Plain walls—no interior finish	Plaster (½ in) on walls	Plaster on wood lath—furred	Plaster (¾ in) on metal lath—furred	Plaster (½ in) on plaster board (⅜ in)—furred	Decorated building board (½ in) without plaster—furred	Plaster (½ in) on rigid insulation (½ in.)—furred	Plaster (½ in) on rigid insulation (1 in.)—furred	Plaster (½ in) on corkboard (1½ in) set in cement mortar (½ in)	Plaster (¾ in) on metal lath attached to furring strips—furred space (over ¾-in wide) faced one side with bright aluminum foil	Plaster on metal lath attached to furring strips (2 in [b])—rock wool fill (1⅝ in.)[b][e]	Plaster (¾ in) on metal lath attached to furring strips (2 in [b]) — flexible insulation (½ in) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.50	0 46	0 30	0 32	0 30	0.23	0 22	0.16	0.14	0 23	0 12	0 20
0.36	0.34	0 24	0.25	0 24	0 19	0 19	0 14	0 12	0.19	0 11	0 17
0 28	0 27	0.20	0.21	0 20	0.17	0 16	0 13	0.11	0 17	0 10	0 15
0 40	0 38	0 26	0 28	0.26	0 20	0 20	0 15	0 13	0.20	0 11	0 18
0.39	0 37	0 26	9 27	0 26	0.20	0 19	0 15	0 13	0.20	0 11	0 18
0 30	0 29	0 22	0.22	0 22	0 17	0.17	0 14	0 12	0 18	0 10	0 16
0 25	0 24	0.19	0.19	0.19	0 16	0.15	0 12	0.11	0.16	0 097	0 14
) 71	0 64	0 37	0 39	0 37	0 26	0 25	0 18	0 15	0 26	0 13	0 23
) 58	0 53	0 33	0 34	0 33	0 24	0 23	0 17	0 14	0 24	0 13	0 21
) 49	0 45	0 30	0 31	0 30	0 22	0 22	0 16	0 14	0 22	0 12	0 20
) 37	0 35	0 25	0 26	0 25	0.20	0 19	0.15	0 13	0 20	0 11	0 18
) 79	0 70	0 39	0 42	0 39	0 27	0 26	0 19	0 16	0 27	0 13	0 23
) 62	0 57	0 34	0 37	0 34	0 25	0 24	0 18	0 15	0 25	0 13	0 22
) 48	0 44	0 29	0 31	0.29	0 22	0 21	0 16	0 14	0 22	0 12	0 20
) 41	0 39	0 27	0 28	0 27	0 21	0 20	0 15	0 13	0 21	0 12	0 18
46	0 43	0 29	0 30	0 29	0 22	0 21	0 16	0 14	0 22	0 12	0 19
33	0 31	0 23	0 24	0 23	0 18	0 18	0 14	0 12	0 18	0 11	0 16
22	0 22	0 17	0 18	0 17	0 15	0 14	0 12	0 10	0 15	0 09	0 13
19	0 18	0 15	0.15	0 15	0 13	0 13	0 11	0 09	0 13	0 09	0 12
44	0 41	0 28	0 29	0 28	0 21	0 21	0 16	0 13	0 21	0 12	0 19
30	0 29	0 22	0 23	0 22	0 17	0 17	0 14	0 12	0 18	0 10	0 16
21	0 20	0 16	0 17	0 16	0 14	0 14	0 11	0 10	0 14	0 09	0 13
17	0 17	0 14	0 14	0 14	0 12	0 12	0 10	0 09	0 12	0 08	0 11
42	0 39	0 27	0 28	0 27	0 21	0 20	0 16	0 13	0 21	0 12	0 19
31	0 29	0 23	0 23	0 22	0 18	0 17	0 14	0 12	0 18	0 11	0 16
.22	0 21	0 17	0 18	0 17	0 14	0 14	0 12	0 11	0 14	0 09	0 13
23	0 22	0 19	0 18	0 18	0 15	0 14	0 12	0 10	0 15	0 09	0 14
.37	0 35	0 25	0.26	0.25	0 19	0 19	0 15	0 13	0 19	0 11	0.17
.20	0 19	0 17	0 16	0 16	0 13	0 13	0 11	0 10	0 14	0 09	0 13
56	0.52	0 32	0 34	0 32	0 24	0 23	0 17	0 14	0 24	0 12	0 21
41	0 39	0 27	0 28	0 27	0 21	0 20	0 15	0 13	0 21	0 12	0 18
49	0 46	0 30	0 32	0 30	0 23	0 22	0 16	0 14	0 23	0 12	0 20
36	0 34	0 26	0 26	0 24	0 19	0 19	0 15	0 13	0 19	0 11	0 17
18	0 17	0 15	0 15	0 14	0 13	0 12	0 10	0 09	0 13	0 08	0 12
34	0 32	0 25	0.25	0 24	0 19	0 18	0 14	0 12	0 19	0 11	0 17
15	0 14	0 13	0 13	0 12	0 11	0 11	0 09	0 08	0 11	0 08	0 10

[e]A waterproof membrane should be provided between the outer material and the insulation fill to
vent possible wetting by absorption and a subsequent lowering of efficiency

TABLE 4. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS WITH VARIOUS TYPES OF VENEERS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph

TYPICAL CONSTRUCTION	TYPE OF WALL		WALL No
	FACING	BACKING	
	4 in **Brick Veneer**[d]	6 in. 8 in. **Hollow Tile**[c] 10 in 12 in	**37** **38** **39** **40**
	4 in **Brick Veneer**[d]	6 in. 10 in **Concrete** 16 in	**41** **42** **43**
	4 in **Brick Veneer**[d]	8 in **Cinder Blocks**	**44**
		8 in. **Cinder Blocks** — Cores filled with granulated cork, 5 12 lb per cu ft	**45**
		12 in. **Cinder Blocks**	**46**
		12 in **Cinder Blocks** — Cores filled with granulated cork, 5 24 lb per cu ft	**47**
		8 in **Concrete Blocks**	**48**
		8 in **Concrete Blocks**—Cores filled with granulated cork, 5 14 lb per cu ft	**49**
		12 in **Concrete Blocks**	**50**
		8 in **Haydite Block**	**51**
		8 in **Haydite Block**—Cores filled with granulated cork, 5 06 lb per cu ft	**52**
		12 in **Haydite Block**	**53**
		12 in **Haydite Block**—Cores filled with granulated cork, 5 6 lb per cu ft	**54**
	4 in **Cut-Stone Veneer**[d]	8 in 12 in **Common Brick** 16 in	**55** **56** **57**
	4 in **Cut-Stone Veneer**[d]	6 in. 8 in **Hollow Tile**[c] 10 in. 12 in	**58** **59** **60** **61**
	4 in **Cut-Stone Veneer**[d]	6 in 10 in **Concrete** 16 in	**62** **63** **64**

[a]Computed from factors marked by * in Table 2
[b]Based on the actual thickness of 2-in furring strips.
[c]The 6-in., 8-in and 10-in tile figures are based on two cells in the direction of heat flow The 12-in tile is based on three cells in the direction of heat flow.

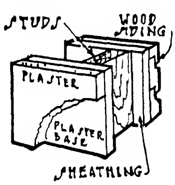
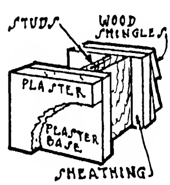
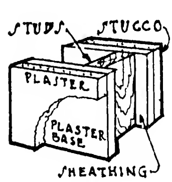
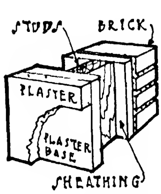
INTERIOR FINISH

Uninsulated Walls						Insulated Walls					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plaster board (⅜ in.)—furred	No plaster—decorated rigid or building board interior finish (½ in.)—furred	Plaster (½ in. on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on cork board (1½ in.) set in cement mortar (½ in.)	Plaster (¾ in.) on metal lath attached to furring strips—furred space (over ¾-in. wide) faced one side with bright aluminum foil	Plaster (¾ in.) on metal lath attached to furring strips (2 in.b)—rock wool fill (1⅝ in.b)c	Plaster (¾ in.) on metal lath attached to furring strips (2 in.b)—flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.19	0.11	0.17
.34	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
34	0.32	0.23	0.24	0.23	0.19	0.18	0.14	0.12	0.19	0.11	0.17
.27	0.26	0.20	0.21	0.20	0.16	0.16	0.13	0.11	0.16	0.10	0.15
.57	0.53	0.33	0.35	0.33	0.24	0.23	0.17	0.14	0.24	0.13	0.21
.48	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
20	0.19	0.16	0.16	0.16	0.13	0.13	0.11	0.10	0.13	0.09	0.12
31	0.30	0.22	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
18	0.18	0.15	0.15	0.15	0.13	0.12	0.10	0.09	0.13	0.08	0.12
44	0.42	0.28	0.30	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
34	0.32	0.24	0.25	0.23	0.19	0.18	0.14	0.12	0.19	0.11	0.17
40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
17	0.16	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
29	0.28	0.21	0.22	0.21	0.17	0.17	0.13	0.12	0.17	0.10	0.16
14	0.14	0.12	0.12	0.12	0.10	0.10	0.09	0.08	0.10	0.07	0.10
37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
28	0.27	0.21	0.21	0.21	0.17	0.16	0.13	0.12	0.17	0.10	0.15
23	0.22	0.18	0.18	0.18	0.15	0.14	0.12	0.11	0.15	0.095	0.14
7	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.20	0.11	0.18
6	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.19	0.11	0.17
5	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
8	0.26	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
1	0.56	0.34	0.36	0.34	0.25	0.24	0.18	0.15	0.25	0.13	0.22
1	0.47	0.31	0.32	0.31	0.23	0.22	0.17	0.14	0.23	0.12	0.20
1	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.21	0.11	0.18

Calculations include cement mortar (½ in.) between veneer or facing and backing
based on one air cell in direction of heat flow
 waterproof membrane should be provided between the outer material and the insulation fill to
nt possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 5. COEFFICIENTS OF TRANSMISSION (U) OF VARIOUS TYPES OF FRAME CONSTRUCTION[a]

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph

TYPICAL CONSTRUCTION	EXTERIOR FINISH	TYPE OF SHEATHING	WALL No
	**Wood Siding or Clapboard**	**1 in. Wood**[d]	65
		$^{25}\!/_{32}$ in. Rigid Insulation	66
		½ in Plaster Board	67
	**Wood Shingles**	**1 in Wood**[d]	68
		$^{25}\!/_{32}$ in. Rigid Insulation[e]	69
		½ in Plaster Board[e]	70
	**Stucco**	**1 in Wood**[d]	71
		$^{25}\!/_{32}$ in. Rigid Insulation	72
		½ in. Plaster Board	73
	**Brick**[f] **Veneer**	**1 in Wood**[d]	74
		$^{25}\!/_{32}$ in Rigid Insulation	75
		½ in Plaster Board	76

[a]Computed from factors marked by * in Table 2

[b]These coefficients may also be used with sufficient accuracy for plaster on wood lath or plaster on plaster board

[c]Based on the actual width of 2 by 4-in studding, namely, 3⅝ in

INTERIOR FINISH

No Insulation Between Studding								Insulation Between Studding				
Plaster on wood lath on studding	Plaster (3/4 in) on metal lath on studding	Plaster (1/2 in) on plaster board (3/8 in) on studding	Plaster (1/2 in) on rigid insulation (1/2 in) on studding	Plaster (1/2 in) on rigid insulation (1 in) on studding	Plaster (1/2 in) on corkboard (1 1/2 in) on studding	No plaster—decorated rigid or building board interior finish (1/2 in)	1 in wood sheathing,[d] furring strips, plaster (1/2 in) on wood lath	Plaster (3/4 in) on metal lath—stud space faced one side with bright aluminum foil	Plaster (3/4 in) on metal lath[b] on studding—flexible insulation (1/2 in) between studding and in contact with sheathing	Plaster (3/4 in) on metal lath[b] on studding—flexible insulation (1/2 in) between studding—2 air spaces	Plaster (3/4 in) on metal lath[b] on studding—flexible insulation (1 in) between studding—2 air spaces	Plaster (3/4 in) on metal lath[b] on studding—rock wool fill (3 5/8 in[e]) between studding[g,h]
A	B	C	D	E	F	G	H	I	J	K	L	M
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.19	0.17	0.15	0.12	0.072
0.19	0.20	0.19	0.16	0.13	0.10	0.16	0.14	0.16	0.15	0.13	0.10	0.068
31	0.33	0.31	0.22	0.17	0.13	0.23	0.19	0.24	0.20	0.17	0.13	0.076
25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.20	0.17	0.15	0.12	0.072
.17	0.17	0.17	0.14	0.11	0.092	0.14	0.14	0.14	0.13	0.11	0.094	0.064
24	0.25	0.24	0.19	0.15	0.11	0.19	0.19	0.19	0.17	0.15	0.12	0.071
30	0.32	0.30	0.22	0.16	0.12	0.22	0.19	0.23	0.20	0.17	0.13	0.076
22	0.23	0.22	0.17	0.14	0.11	0.19	0.15	0.18	0.16	0.14	0.11	0.071
40	0.43	0.40	0.26	0.19	0.14	0.28	0.22	0.29	0.24	0.20	0.14	0.081
27	0.28	0.27	0.20	0.15	0.12	0.21	0.17	0.21	0.18	0.16	0.12	0.074
21	0.21	0.21	0.16	0.11	0.10	0.17	0.15	0.17	0.15	0.13	0.11	0.068
35	0.37	0.35	0.24	0.18	0.13	0.25	0.21	0.26	0.22	0.18	0.14	0.079

[d]Yellow pine or fir—actual thickness about 25/32 in.

[e]Furring strips between wood shingles and sheathing

[f]Small air space and mortar between building paper and brick veneer neglected

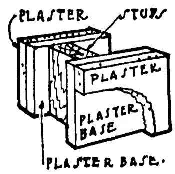
[g]A waterproof membrane should be provided between the outer material and the insulation fill to vent possible wetting by absorption and a subsequent lowering of efficiency.

[h]Stud and rock wool fill areas combined

TABLE 6. COEFFICIENTS OF TRANSMISSION (U) OF FRAME INTERIOR WALLS AND PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides

TYPICAL CONSTRUCTION



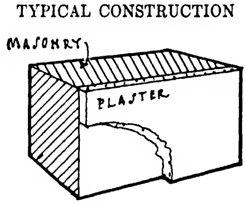
TYPICAL CONSTRUCTION	WALL No	SINGLE PARTITION (FINISH ON ONE SIDE OF STUDDING)	DOUBLE PARTITION (FINISHED ON BOTH SIDES OF STUDDING)				
			Air Space Between Studding	Flaked Gypsum Fill[b] Between Studding	Rock Wool Fill[b] Between Studding	½-in Flexible Insulation Between Studding (One Air Space)	Stud Space Faced One Side with Bright Aluminum Foil
TYPE OF WALL		A	B	C	D	E	F
Wood Lath and Plaster On Studding	77	0.62	0 34	0.11	0 076	0.21	0 24
Metal Lath and Plaster[c] On Studding	78	0.69	0 39	0 11	0.078	0 23	0 26
Plaster Board (⅜ in) and Plaster[d] On Studding	79	0 61	0 34	0.10	0 075	0.21	0 24
½ in Rigid Insulation and Plaster[d] On Studding	80	0.35	0 18	0.083	0.063	0 14	0 15
1 in. Rigid Insulation and Plaster[d] On Studding	81	0.23	0.12	0 066	0.054	0 097	0 10
1½ in Corkboard and Plaster[d] On Studding	82	0.16	0 081	0.052	0 044	0 070	0 073
2 in Corkboard and Plaster[d] On Studding	83	0 12	0 063	0 045	0 038	0 057	0 059

[a]Computed from factors marked by * in Table 2
[b]Thickness assumed 3⅝ in.
[c]Plaster on metal lath assumed ¾-in thick
[d]Plaster assumed ½-in. thick

TABLE 7. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides

TYPICAL CONSTRUCTION



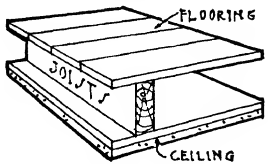
TYPICAL CONSTRUCTION	No	PLAIN WALLS (NO PLASTER)	WALLS PLASTERED ON ONE SIDE	WALLS PLASTERED ON BOTH SIDES
TYPE OF WALL		A	B	C
4-in Hollow Clay Tile	84	0 45	0 42	0.40
4-in. Common Brick	85	0.50	0.46	0.43
4-in Hollow Gypsum Tile	86	0 30	0.28	0.27
2-in Solid Plaster	87	.....	....	0.53

[a]Computed from factors marked by * in Table 2.

TABLE 8. COEFFICIENTS OF TRANSMISSION (U) OF FRAME CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides and are based on still air (no wind) conditions on both sides

TYPICAL CONSTRUCTION



TYPE OF CEILING	INSULATION BETWEEN JOISTS	No	TYPE OF FLOORING				
			No Flooring	Yellow Pine Flooring[b] on Joists	Yellow Pine Flooring on Rigid Insulation (½ in.) on Joists	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring[b] on Joists	¼-in. Battleship Linoleum on Yellow Pine Flooring[b]
			A	B	C	D	E
No Ceiling	None	**1**	. ..	0.46	0.27	0.34	0.34
Metal Lath and Plaster (¾ in.)	None	**2**	0.69	0.30	0.21	0.25	0.25
Wood Lath and Plaster	None	**3**	0.62	0.28	0.20	0.24	0.24
Plaster Board (⅜ in.) and Plaster (½ in.)	None	**4**	0.61	0.28	0.20	0.24	0.23
Rigid Insulation (½ in.) and Plaster (½ in.)	None	**5**	0.35	0.21	0.16	0.18	0.18
Rigid Insulation (1 in.) and Plaster (½ in.)	None	**6**	0.23	0.16	0.13	0.14	0.14
Metal Lath and Plaster	Bright Aluminum Foil[e]	**7**	0.59	0.22	0.17	0.19	0.19
Metal Lath and Plaster	Flexible[d] Insulation (1 in.)	**8**	0.17	0.13	0.11	0.12	0.12
Metal Lath and Plaster	Flexible[d] Insulation (2 in.)	**9**	0.10	0.086	0.076	0.081	0.081
Metal Lath and Plaster	Rock Wool Fill (3⅝ in.)	**10**	0.079	0.068	0.063	0.066	0.066
Corkboard (1½ in.) and Plaster (½ in.)	None	**11**	0.16	0.12	0.10	0.11	0.11
Corkboard (2 in.) and Plaster (½ in.)	None	**12**	0.12	0.10	0.087	0.094	0.094

[a]Computed from factors marked by * in Table 2

[b]Thickness assumed to be 25/32 in.

[c]Thickness assumed to be 13/16 in.

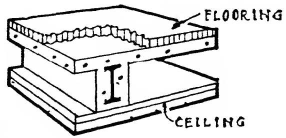
[d]Based on one air space with no flooring, and two air spaces with flooring. The value of U will be the same if insulation is applied to under side of joists and separated from lath and plaster ceiling by 1-in. furring strips.

[e]Air space faced on one side with bright aluminum foil.

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides

TYPICAL CONSTRUCTION



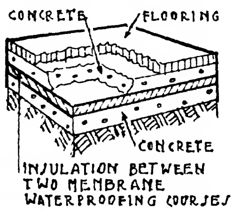
Type of Ceiling	Thickness of Concrete (Inches)	No	A: No Flooring (Concrete Bare)[b]	B: Yellow Pine Flooring[c] on Wood Sleepers Embedded in Concrete[d]	C: Maple or Oak Flooring[e] on Yellow Pine Sub-Flooring[c] on Wood Sleepers Embedded in Concrete	D: Tile or Terrazzo[f] Flooring on Concrete	E: ¼-in Battleship Linoleum Directly on Concrete
No Ceiling	4	1	0.65	0.40	0.31	0.61	0.44
	6	2	0.59	0.37	0.30	0.56	0.41
	8	3	0.53	0.35	0.28	0.51	0.38
	10	4	0.49	0.33	0.27	0.47	0.36
½ in Plaster Applied Directly to Under Side of Concrete	4	5	0.59	0.38	0.30	0.56	0.41
	6	6	0.54	0.35	0.28	0.52	0.38
	8	7	0.50	0.33	0.27	0.47	0.36
	10	8	0.45	0.32	0.26	0.44	0.34
Suspended or Furred Metal Lath and Plaster (¾ in.) Ceiling	4	9	0.37	0.28	0.23	0.36	0.29
	6	10	0.35	0.26	0.22	0.34	0.28
	8	11	0.33	0.25	0.21	0.32	0.27
	10	12	0.32	0.24	0.21	0.31	0.25
Suspended or Furred Ceiling of Plaster Board (⅜ in.) and Plaster (½ in.)	4	13	0.35	0.26	0.22	0.34	0.28
	6	14	0.33	0.25	0.21	0.32	0.26
	8	15	0.31	0.24	0.21	0.30	0.25
	10	16	0.30	0.23	0.20	0.29	0.24
Suspended or Furred Ceiling of Rigid Insulation (½ in) and Plaster (½ in)	4	17	0.24	0.20	0.17	0.24	0.21
	6	18	0.23	0.19	0.17	0.23	0.20
	8	19	0.22	0.18	0.16	0.22	0.19
	10	20	0.22	0.18	0.16	0.21	0.19
Plaster (½ in) on Corkboard (1½ in) Set in Cement Mortar (½ in.) on Concrete	4	21	0.15	0.13	0.12	0.14	0.14
	6	22	0.14	0.13	0.12	0.14	0.13
	8	23	0.14	0.12	0.11	0.14	0.13
	10	24	0.14	0.12	0.11	0.14	0.13

[a]Computed from factors marked by * in Table 2
[b]The figures in COLUMN A may be used with sufficient accuracy for concrete floors covered with carpet.
[c]Thickness of yellow pine flooring assumed to be 25/32 in.
[d]The figures in COLUMN B may be used with sufficient accuracy for maple or oak flooring[e] applied directly over the concrete on wood sleepers
[e]Thickness of maple or oak flooring assumed to be 13/16 in.
[f]Thickness of tile or terrazzo assumed 1 in.

TABLE 10 COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING[a, e]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the ground and the air over the floor, and are based on still air (no wind) conditions

TYPICAL CONSTRUCTION



Type and Thickness of Insulation	Thickness of Concrete (Inches)	No	A: No Flooring (Concrete Bare)	B: Yellow Pine Flooring[b] on Wood Sleepers Resting on Concrete	C: Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring on Wood Sleepers Resting on Concrete	D: Tile or Terrazzo[d] on Concrete	E: ¼-in Battleship Linoleum Directly on Concrete
None	4	1	1 07	0 35	0 28	0.98	0 60
	6	2	0 90	0 33	0 27	0 84	0 54
	8	3	0 79	0 32	0 26	0 74	0 50
	10	4	0 70	0 30	0 25	0.66	0.46
None[e]	4	5	0 66	0 29	0 24	0 63	0 44
	8	6	0 54	0 27	0 23	0 52	0.39
1 in. **Rigid Insulation**[e]	4	7	0 22	0 16	0 14	0 22	0 19
1 in **Rigid Insulation**[e]	8	8	0 21	0 15	0 13	0 20	0.18
2 in **Corkboard**[e]	4	9	0 12	0 099	0 093	0 12	0.11
2 in **Corkboard**[e]	8	10	0 12	0 096	0 090	0 12	0.11

[a]Computed from factors marked by * in Table 2

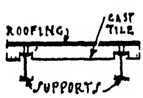
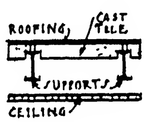
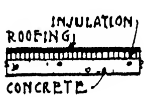
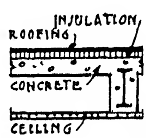
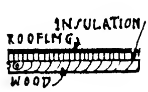
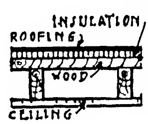
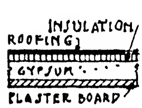
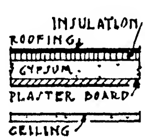
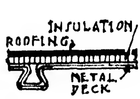
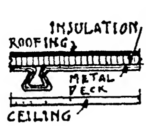
[b]Assumed 25/32 in thick.

[c]Assumed 13/16 in thick.

[d]Assumed 1 in thick

[e]The figures for Nos 5 to 10, inclusive, include 3-in cinder concrete placed directly on the ground The insulation is applied between the cinder concrete and the stone concrete Usually the insulation is protected on both sides by a waterproof membrane, but this is not considered in the calculations

TABLE 11. COEFFICIENTS OF TRANSMISSION (U) OF VARIOUS TYPES OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING[a]

TYPICAL CONSTRUCTION: Without Ceilings	TYPICAL CONSTRUCTION: With Metal Lath and Plaster Ceilings[d]	TYPE OF ROOF DECK	Thickness of Roof Deck (Inches)	No.
		**Precast Cement Tile**	1⅝	1
		**Concrete** **Concrete** **Concrete**	2 4 6	2 3 4
		**Wood** **Wood** **Wood** **Wood**	1[b] 1½[b] 2[b] 4[b]	5 6 7 8
		**Gypsum Fiber Concrete[c] (2 in.) on Plaster Board (⅜ in.)** **Gypsum Fiber Concrete[c] (3 in.) on Plaster Board (⅜ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (½ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (1 in.)**	2⅜ 3⅜ 2½ 3	9 10 11 12
		**Flat Metal Roofs** Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hour per square foot of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph	....	13

[a]Computed from factors marked by * in Table 2.
[b]Nominal thicknesses specified—actual thicknesses used in calculations.
[c]Gypsum fiber concrete—87½ per cent gypsum, 12½ per cent wood fiber.

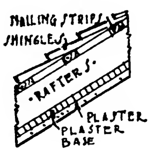
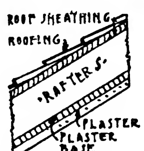
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

WITHOUT CEILING—UNDER SIDE OF ROOF EXPOSED								WITH METAL LATH AND PLASTER CEILINGS[d]							
No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)	No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
0.84	0.37	0.24	0.18	0.14	0.22	0.16	0.13	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	0.42	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11
0.64	0.33	0.22	0.16	0.13	0.21	0.15	0.12	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11
0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10
0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095
0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092
0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082
.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097
32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091
26	0.19	0.15	0.12	0.10	0.14	0.11	0.10	0.20	0.16	0.13	0.11	0.09	0.12	0.10	0.087
19	0.15	0.12	0.10	0.09	0.12	0.10	0.08	0.16	0.13	0.11	0.09	0.08	0.10	0.09	0.077
95	0.39	0.25	0.18	0.14	0.23	0.17	0.13	0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11

[d]These coefficients may be used with sufficient accuracy for wood lath and plaster, or plaster board and ster ceilings It is assumed that there is an air space between the under side of the roof deck and the per side of the ceiling.

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph

TYPICAL CONSTRUCTION	TYPE OF ROOFING AND ROOF SHEATHING	INSULATION BETWEEN ROOF RAFTERS	No	TYPE OF CEILING (APPLIED DIRECTLY TO ROOF RAFTERS)								
				No Ceiling (Rafters Exposed)	Metal Lath and Plaster (¾ in)	Plaster Board (⅜ in.) and Plaster (½ in)	Wood Lath and Plaster	Rigid Insulation (½ in)	Rigid Insulation (½ in) and Plaster (½ in)	Rigid Insulation (1 in) and Plaster (½ in)	Corkboard (1½ in) and Plaster (½ in)	Corkboard (2 in) and Plaster (½ in)
				A	B	C	D	E	F	G	H	I
	Wood Shingles on Wood Strips[b]	None	1	0 46	0 30	0 29	0 29	0 22	0 21	0 16	0 12	0 10
		Bright Aluminum Foil[g]	2		0 22	0 21	0 21	0 18	0 17	0 14	0 11	0 089
		1 in Flexible[c]	3		0 13	0 12	0 12	0 11	0 11	0 092	0 078	0 069
		2 in Flexible[c]	4	.	0 086	0 083	0 083	0 076	0 075	0 068	0 060	0.054
		3⅝ in Rock Wool[e]	5		0 063	0 062	0 062	0 058	0 058	0 053	0 048	0 044
	Asphalt Shingles, Rigid Asbestos Shingles, Composition Roofing, or Slate or Tile Roofing[d] on Wood Sheathing[f]	None	6	0 56	0 34	0 32	0 32	0 24	0 23	0 17	0 13	0 11
		Bright Aluminum Foil[g]	7		0 24	0 24	0 24	0 19	0 18	0 14	0 11	0 098
		1 in Flexible[c]	8		0 13	0 13	0 13	0 11	0 11	0 095	0 080	0 071
		2 in Flexible[c]	9		0 088	0 087	0 087	0 079	0 078	0 070	0 062	0 056
		3⅝ in Rock Wool[e]	10	.	0 065	0 064	0 064	0 060	0 059	0 054	0 049	0 045

[a]Computed from factors marked by * in Table 2 Nos 6 to 10, inclusive, based on ½-in thick slate
[b]Based on 1 in by 4 in strips spaced 2 in.
[c]Figures based on two air spaces Insulation may also be applied to under side of roof rafters with furring strips between
[d]Roofing felt between roof sheathing and slate or tile neglected in calculations
[e]Assumed 3⅝ in thick based on the actual width of 2 in by 4 in rafters
[f]Sheathing assumed 25/32 in thick
[g]Air space faced on one side with bright aluminum foil.

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS, SKYLIGHTS AND GLASS WALLS

Coefficients are based on a wind velocity of 15 mph, and are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air inside and outside of the door, window, skylight or wall

A Windows and Skylights

DESCRIPTION	U
Single	1.13[a, c]
Double	0.45[a]
Triple	0.281[a]

B Solid Wood Doors[b, c]

NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	U
1	25/32	0.69
1¼	1 1/16	0.59
1½	1 5/16	0.52
1¾	1⅜	0 51
2	1⅝	0 46
2½	2⅛	0.38
3	2⅝	0 33

C Glass Walls

DESCRIPTION	U
Hollow glass tile wall, 6 x 6 x 2 in. thick blocks, wind velocity 15 mph, outside surface, still air, inside surface	0 60
Still air, outside and inside surface	0 48

[a]See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932
[b]Computed using C = 1 15 for wood, f_i = 1 65 and f_o = 6 0
[c]It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1 13 Btu per hour per square foot per degree difference between inside and outside air temperatures

it is probable that the values of U for these two types of roofs will compare favorably.

The thicknesses upon which the coefficients in Tables 3 to 13 inclusive, are based are as follows:

Brick veneer	4 in.
Plaster and metal lath	¾ in
Plaster (on wood lath, plasterboard, rigid insulation, board form, or corkboard)	½ in.
Slate (roofing)	½ in.
Stucco on wire mesh reinforcing	1 in.
Tar and gravel or slag-surfaced built-up roofing	⅜ in.
1-in. lumber (S 2-S)	25/32 in.
1½-in lumber (S-2-S)	1 5/16 in.
2-in. lumber (S-2-S)	1⅝ in.
2½-in. lumber (S-2-S)	2⅛ in.
3-in. lumber (S-2-S)	2⅝ in.
4-in lumber (S-2-S)	3⅝ in.
Finish flooring (maple or oak)	13/16 in.

Solid brick walls are based on 4-in. hard brick (high density) and the remainder common brick (low density). Stucco is assumed to be 1-in. thick on masonry walls.. Where metal lath and plaster are specified, the metal lath is neglected.

The coefficients of transmission of the pitched roofs in Table 12 apply where the roof is over a heated attic or top floor so the heat passes directly through the roof structure including whatever finish is applied to the underside of the roof rafters.

Combined Coefficients of Transmission

If the attic is unheated, the roof structure and ceiling of the top floor must both be taken into consideration, and the combined coefficient of transmission determined. The formula for calculating the combined coefficient of transmission of a top floor ceiling, unheated attic space, and pitched roof, per square foot of ceiling area, is as follows:

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}} \tag{6}$$

where

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling

Stating the formula in terms of the total heat resistance of the ceiling and roof,

$$\frac{1}{U} = R = \frac{1}{U_{ce}} + \frac{1}{U_r \times n} \tag{7}$$

In selecting the values to be used for U_r and U_{ce} it should be noted that the under surface of the roof and the upper surface of the ceiling are more nearly equivalent to the boundary surfaces of an internal air space than they are to the external surfaces of a wall. It would be more nearly correct to use a value of 2.2 rather than the usual value of 1.65 as coefficients for these surfaces. In most cases this would make only a minor change in U. It should be noted that the over-all coefficient should be multiplied by the ceiling and not the roof area.

If the unheated attic space between the roof and ceiling has no dormers, windows or vertical wall spaces the combined coefficients may be used for determining the heat loss through the roof construction between the attic and top floor ceiling. If the unheated attic contains windows and vertical wall spaces these must be taken into consideration in calculating the roof area and also its coefficient U_r. In this case an approximate value of U_r may be obtained as the summation of the coefficient of each individual section such as the roof, vertical walls or windows times its percentage of total area. This coefficient may be used with reasonable accuracy in the above formulae. If, however, there are roof ventilators such that the attic air is substantially at outside temperature, then the roof should be neglected and only the coefficient for the top floor ceiling construction used.

Basements and Unheated Rooms

The heat loss through floors into basements and into unheated rooms kept closed may be computed by assuming a temperature for these rooms of 32 F. Additional information on the inside and outside temperatures to be used in heat loss calculations is given in Chapter 7.

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PROBLEMS IN PRACTICE

1 ● What is the coefficient *U* and how is it applied?

The coefficient *U* is the heat loss through walls, ceilings, and floors and the value depends upon the construction and material, expressed in Btu per hour per square foot per degree

difference in temperature between the inside and outside. To determine the total heat loss, multiply U for each material by the square feet of surface and the temperature difference.

2 ● Find the value of U for a 6-in. concrete wall with plaster on metal lath attached to 2-in. furring strips with flanged ½-in. blanket insulation.

0.23 (Table 3, Wall 12L).

3 ● A wall is built with two layers of ½-in. insulating material spaced 1 in. apart; the air space is lined on one side with bright aluminum foil; mean temperature is 40 F; still air on both sides of wall; *k* for insulating material is 0.34. Calculate the value of *U*.

$$f_i = 1.65; f_o = 1.65; a = 0.46$$

$$R = \frac{1}{1.65} + \frac{0.5}{0.34} + \frac{1}{0.46} + \frac{0.5}{0.34} + \frac{1}{1.65} = 6.327$$

$$U = \frac{1}{R} = 0.158$$

4 ● What is the inside surface temperature of a 6-in. solid concrete wall? Inside air, 70 F; outside air, −20 F with 15 mph wind.

The temperature drop from point to point through a wall is directly proportional to the heat resistance.

$$f_i = 1.65;\ k \text{ for concrete} = 12.0, f_o = 6.0$$

$$\text{Over-all resistance } R = \frac{1}{1.65} + \frac{6}{12.0} + \frac{1}{6.0} = 1.27$$

$$\frac{\text{Temperature drop, inside air to surface}}{\text{Temperature drop, air to air}} = \frac{\frac{1}{1.65}}{1.27}$$

$$\text{Temperature drop, inside air to surface} = \frac{90}{1.27 \times 1.65} = 43$$

$$70 - 43 = 27 \text{ F, inside surface temperature of wall}$$

5 ● How many inches of insulating material having a conductivity of 0.30 would be required, for the wall of Question 4, to raise the inside surface temperature to 60 F?

Temperature drop, air to inside surface = 10 F, temperature drop, inside surface to outside air = 80 F. Therefore, the heat resistance from inside wall surface to outside air must be eight times that from inside air to inside wall surface, or $8 \times \frac{1}{1.65} = 4.85$. The resistance for added material is, therefore,

$$4.85 - \left(\frac{6}{12.0} + \frac{1}{6}\right) = 4.19$$

$$4.19 \times 0.30 = 1.25 \text{ in. of insulation.}$$

6 ● An unheated attic space in a residence has an equivalent pitched roof area of 1560 sq ft and a ceiling area of 1200 sq ft. If 15 per cent of the roof area is composed of vertical wall spaces having a value of U = 0.52, determine the total heat loss per hour through the ceiling and roof for a temperature difference of 85 F, if U = 0.46 for the roof and U = 0.38 for the ceiling.

An approximate value of U for the roof is equivalent to the summation of coefficients for each individual section times its percentage of total area.

$U_r = (0.52 \times 0.15) + (0.46 \times 0.85) = 0.47.$

Ratio of roof area to ceiling = 1560 ÷ 1200 = 1.3

Substituting in Formula 6:

$$U = \frac{0.47 \times 0.38}{0.47 + \frac{0.38}{1.3}} = 0.235$$

$H = AU\,(t_i - t_o) = 1200 \times 0.235 \times 85 = 23{,}900$ Btu per hour.

Chapter 6

AIR LEAKAGE

Nature of Air Infiltration, Infiltration Through Walls, Window Leakage, Door Leakage, Selection of Wind Velocity, Crack Length used for Computations, Multi-Story Buildings, Heat Equivalent of Air Infiltration

AIR leakage losses are those resulting from the displacement of heated air in a building by unheated outside air, the interchange taking place through various apertures in the building, such as cracks around doors and windows, fireplaces and chimneys. This leakage of air must be considered in heating and cooling calculations. (See Chapters 7 and 8.)

NATURE OF AIR INFILTRATION

The natural movement of air through building construction is due to two causes. One is the pressure exerted by the wind; the other is the difference in density of outside and inside air because of differences in temperature.

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages owing to this building up of pressure will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Mechanical ventilating systems are frequently designed to produce positive or negative pressures in an enclosure which are greater or lower than prevalent wind pressures. In such designs, if the rate at which air is specified to be introduced to or removed from the enclosure by positive means exceeds the infiltration rate, it is common practice to use the greater value in determining the heating capacity to warm the outside air.

The air exchange owing to temperature difference, inside to outside, is not appreciable in low buildings. In tall, single story buildings with openings near the ground level and near the ceiling, this loss must be considered. Also in multi-story buildings it is a large item unless the sealing between various floors and rooms is quite perfect. This temperature effect is a *chimney action*, causing air to enter through openings at lower levels and to leave at higher levels.

A complete study of all of the factors involved in air movement through building constructions would be very complex. Some of the complicating factors are: the variations in wind velocity and direction; the exposure of the building with respect to air leakage openings and with respect to adjoining buildings; the variations in outside temperatures as influencing the chimney effect; the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors; the influence of a planned air supply and the related outlet vents; and the variation from the average of individual building units. A study of infiltration points to the need for care in the obtaining of good building construction, or unnecessarily large heat losses will result.

INFILTRATION THROUGH WALLS

Table 1 gives data on infiltration through brick and frame walls. The brick walls listed in this table are walls which show poor workmanship and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

The value of building paper when applied between sheathing and shingles is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it

TABLE 1. INFILTRATION THROUGH WALLS[a]

Expressed in cubic feet per square foot per hour

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in. Brick Wall	Plain	1.75	4.20	7.85	12 2	18.6	22.9
	Plastered	0.017	0.037	0.066	0.107	0.161	0.236
13 in. Brick Wall	Plain	1.44	3.92	7.48	11.6	16.3	21.2
	Plastered	0.005	0.013	0.025	0.043	0.067	0.097
Frame Wall, with lath and plaster[b]		0.03	0.07	0.13	0.18	0.23	0.26

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed at the end of this chapter.

[b]Wall construction Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and 3 coats gypsum plaster

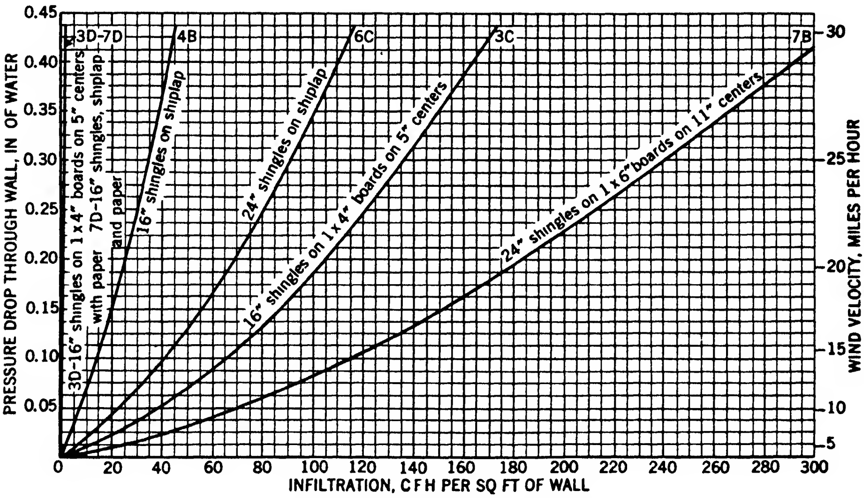

FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them.

The amount of infiltration that may be expected through single walls used in farm and other shelter buildings, is shown in Fig. 2. The infiltration indicated in Figs. 1 and 2 is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

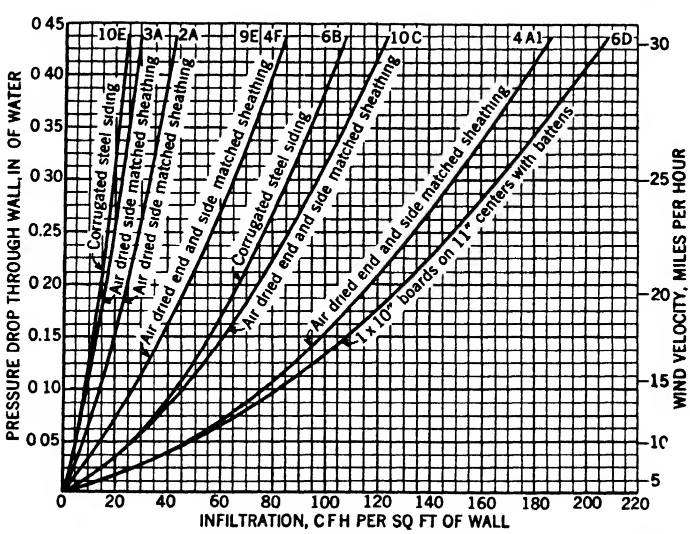

FIG. 2. INFILTRATION THROUGH SINGLE SURFACE WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

WINDOW LEAKAGE

The amount of infiltration for various types of windows is given in Table 2. The fit of double-hung wood windows is determined by crack and clearance as illustrated in Fig. 3. The length of the perimeter opening or crack for a double-hung window is equal to three times the width plus two times the height, or in other words, it is the outer sash perimeter length plus the meeting rail length. Values of leakage shown in Table 2 for the average double-hung wood window were determined by setting the average measured crack and clearance found in a field survey of a large number of windows on nine windows tested in the laboratory. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not

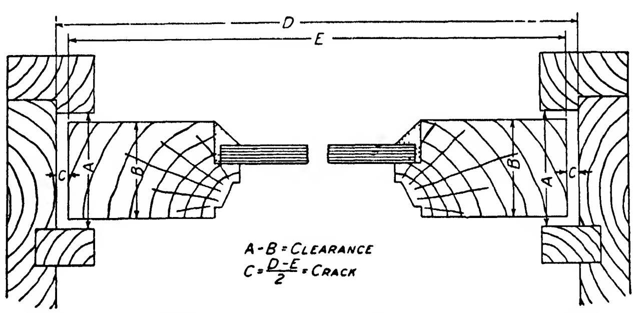

FIG. 3. DIAGRAM ILLUSTRATING CRACK AND CLEARANCE

fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of sash section in contact with steel work at mullions should be figured at 25 per cent of the values for industrial pivoted windows as given in Table 2.

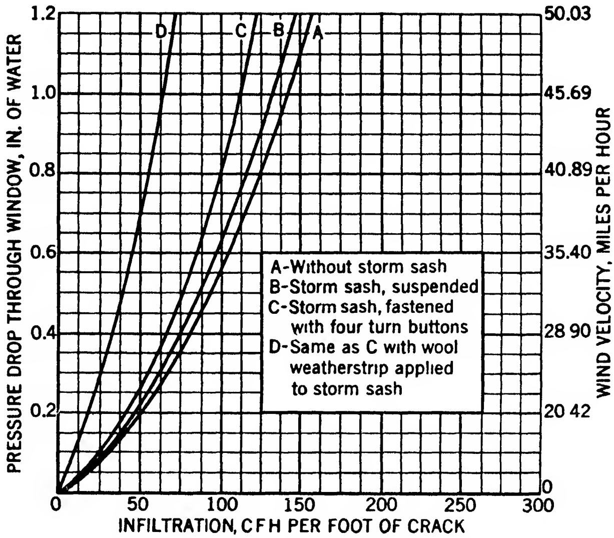

FIG. 4. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH—5/64-IN. CRACK AND 1/32-IN. CLEARANCE

Leakage values for storm sash are given in Figs. 4 and 5. When storm sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows. When storm sash are applied to poorly fitted windows, a reduction in leakage of 50 per cent may be secured.

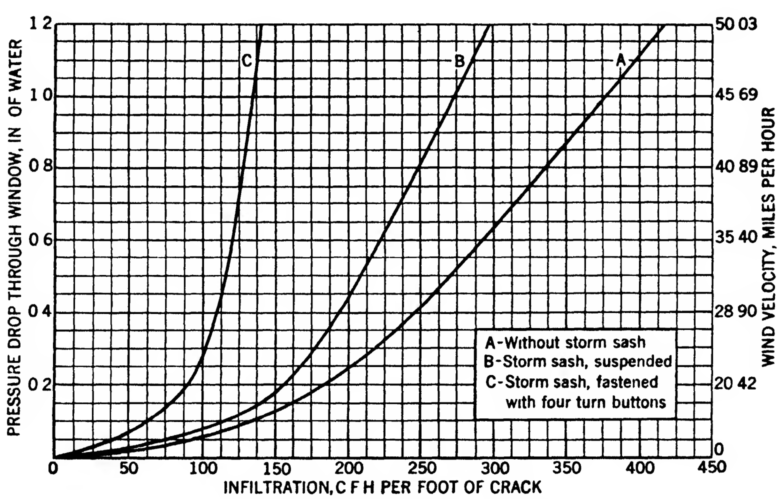

FIG. 5. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH—1/8-IN. CRACK AND 1/8-IN. CLEARANCE

TABLE 2. INFILTRATION THROUGH WINDOWS

Expressed in Cubic Feet per Foot of Crack per Hour[a]

TYPE OF WINDOW	REMARKS	WIND VELOCITY, MILES PER HOUR					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked[b]	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction[b]	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weatherstripped, 1/16-in crack and 3/64-in clearance[c] Includes wood frame leakage[d]	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped[d]	4.3	15.5	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weatherstripped, 3/32-in crack and 3/32-in clearance[e] Includes wood frame leakage[d]	26.9	69.0	110.5	153.9	199 2	249.4
	Ditto, weatherstripped[d]	5.9	18 9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted, 1/16-in crack[g]	52	108	176	244	304	372
	Architectural projected, 1/32-in crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in crack[j]	3	10	18	26	36	48
	Heavy casement section, projected 1/32-in crack[j]	8	24	38	54	72	92
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table, with the exception of those for double-hung and hollow metal windows, are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed at the end of this chapter.

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and not-calked tests

[c]The fit of the average double-hung wood window was determined as 1/16-in crack and 3/64-in clearance by measurements on approximately 600 windows under heating season conditions

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking

[e]A 3/32-in. crack and clearance represents a poorly fitted window, much poorer than average

[f]Windows tested in place in building

[g]Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out

[h]Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and it has refinements in weathering and hardware Used in semi-monumental buildings such as schools Ventilators swing in or out and are balanced on side arms. 1/32-in crack is obtainable in the best practice of manufacture and installation, 3/64-in crack considered to represent average practice

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight 1/64-in crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice

[j]Made of heavy sections Ventilators swing in or out and stay set at any degree of opening 1/64-in crack is obtainable in the best practice of manufacture and installation, 1/32-in crack considered to represent average practice.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible With 3/64-in crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table

DOOR LEAKAGE

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, such as might be found in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

The infiltration rate through swinging and revolving doors is generally a matter of judgment by the engineer making cooling load determinations and in the absence of adequate research data the values given in Table 3 represent current engineering practice. These values are based on the average number of persons in a room at a specified time, which may also be the same occupancy assumed for determining the outside ventilation requirements outlined in Chapters 3 and 8.

TABLE 3 INFILTRATION THROUGH OUTSIDE DOORS FOR COOLING LOADS[a]
Expressed in Cubic Feet per Minute per Person in Room

Application	Pair 36 in Swinging Doors, Single Entrance[b]
Bank	7.5
Barber Shop	4.5
Broker's Office	7.0
Candy and Soda	6.0
Cigar Store	25.0
Department Store	8.0
Dress Shop	2.5
Drug Store	7.0
Furrier	2.5
Hospital Room	3.5
Lunch Room	5.0
Men's Shop	3.5
Office	3.0
Office Building	2.0
Public Building	2.5
Restaurant	2.5
Shoe Store	3.5

[a]For doors located in only one wall or where doors in other walls are of revolving type

[b]Vestibules with double pair swinging doors, infiltration may be assumed 75 per cent of swinging door values

Infiltration for 72 in revolving doors may be assumed 60 per cent of swinging door values.

SELECTION OF WIND VELOCITY

Although all authorities do not agree upon the value of the wind velocity that should be chosen for any given locality, it is common engineering practice to use the average wind velocity during the three coldest months of the year. Until this point is definitely established the practice of using average values will be followed. Average wind velocities for the months of December, January and February for various cities in the United States and Canada are given in Table 2, Chapter 7.

In considering both the transmission and infiltration losses, the more exact procedure would be to select the outside temperature and the wind velocity corresponding thereto, based on Weather Bureau records, which would result in the maximum heat demand. Since the proportion of transmission and infiltration losses varies with the construction and is different for every building, the proper combination of temperature and wind velocity to be selected would be different for every type of building, even in the same locality. Furthermore, such a procedure would necessitate a laborious cut-and-try process in every case in order to determine the worst combination of conditions for the building under consideration. It would also be necessary to consider heat lag due to heat capacity in the case of heavy masonry walls, and other factors, to arrive at the most accurate solution of the problem. Although heat capacity should be considered wherever possible, it is seldom possible to accurately determine the worst combination of outside temperature and wind velocity for a given building and locality. The usual procedure, as explained in Chapter 7, is to select an outside temperature which is not more than 15 F above the lowest recorded, and the average wind velocity during the months of December, January and February.

The direction of prevailing winds may usually be included within an angle of about 90 deg. The windows that are to be figured for prevailing and non-prevailing winds will ordinarily each occupy about one-half the perimeter of the structure, the proportion varying to a considerable extent with the plan of the structure. (See discussion of wind movement in Chapter 36 and Table 2 in Chapter 7).

CRACK LENGTH USED FOR COMPUTATIONS

In no case should the amount of crack used for computation be less than half of the total crack in the outside walls of the room. Thus, in a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building.

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 4. This method may be used to advantage as a check on the calculations made in the more exact manner.

MULTI-STORY BUILDINGS

In tall buildings, infiltration may be considerably influenced by temperature difference or chimney effect which will operate to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors preventing free upward movement, so that wind and

TABLE 4. AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION

KIND OF ROOM OR BUILDING	NUMBER OF AIR CHANGES TAKING PLACE PER HOUR
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½ to ¾
Entrance Halls	2 to 3
Reception Halls	2
Living Rooms	1 to 2
Dining Rooms	1 to 2
Bath Rooms	2
Drug Stores	2 to 3
Clothing Stores	1
Churches, Factories, Lofts, etc.	½ to 3

temperature difference may seldom cooperate to the fullest extent. Making the rough assumption that the *neutral zone* is located at mid-height of a building, and that the temperature difference is 70 F, the following formulae may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$M_e = \sqrt{M^2 - 1.75\,a} \tag{1}$$

$$M_e = \sqrt{M^2 + 1.75\,b} \tag{2}$$

where

M_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
M = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height.
b = distance if *below* mid-height.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the greater heights makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring

buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels even to the point of omitting all heating surface on the top several floor levels. One rule is to calculate the heating surface of the entire stair-well in the usual way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

HEAT EQUIVALENT OF AIR INFILTRATION

Sensible Heat Loss

The heat required to warm cold outside air, which enters a room by infiltration, to the temperature of the room is given by the equation:

$$H_s = 0.24\,Q\,d\,(t_i - t_o) \tag{3}$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t_i Btu per hour.
0.24 = specific heat of air.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_o, pounds per cubic foot.
t_i = room air temperature, degrees Fahrenheit.
t_o = outside air temperature, degrees Fahrenheit.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room for the maintenance of proper winter comfort conditions, it is necessary to determine the heat equivalent to evaporate the required amount of water vapor, which may be calculated by the equation:

$$H_l = Q\,d\left(\frac{M_i - M_o}{7000}\right)L \tag{4}$$

where

H_l = heat required to increase moisture content of air leaking into building from M_o to M_i, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_i, pounds per cubic foot.
M_i = vapor density of inside air, grains per pound of dry air.
M_o = vapor density of outside air, grains per pound of dry air.
L = latent heat of vapor at M_i, Btu per pound.

It is sufficiently accurate to use d = 0.075 lb, in which case equation 3 reduces to 5 and if the latent heat of vapor is assumed for general conditions as 1060 Btu per pound Equation 4 reduces to 6.

$$H_s = 0.018\,Q\,(t_i - t_o) \tag{5}$$

$$H_l = 0.0114\,Q\,(M_i - M_o) \tag{6}$$

Changing the temperature and vapor subscripts in Equations 5 and 6 to $(t_o - t_i)$ and $(M_o - M_i)$ permits the use of these same formulas for determining the sensible and latent heat gains due to infiltration in cooling load computations.

If a building has more than one room which is divided by interior walls or partitions, it is sufficiently accurate to use half of the total infiltration losses for determining the total heat requirements. Where buildings have no interior walls, the infiltration losses are calculated by using one-half of the total crack, in which case the entire infiltration loss should be considered.

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PROBLEMS IN PRACTICE

1 ● What two natural forces cause infiltration?

a. Wind causes pressure to be exerted on one or two sides of a building with consequent movement of air through openings

b. Temperature difference inside to outside causes a difference in density with consequent entrance of air at lower openings and exit of air at higher openings.

2 ● What is the neutral zone as applied to infiltration in buildings?

Due to temperature difference inside air tends to flow out of openings near the top of the building and is replaced by outside air coming in at lower openings. At some height no flow occurs in or out. This is referred to as the neutral zone and often is taken at the mid-height of the building but may be higher or lower than this depending on several factors.

3 ● In what type of structure is infiltration due to temperature difference of importance?

In tall open buildings, in multi-story buildings inadequately sealed between floors and in stair-wells for multi-story buildings the temperature difference effect is important.

4 ● What procedure is sometimes used to compensate for this temperature or chimney effect in stair-wells?

One rule is to place 50 per cent of the calculated radiation in the bottom third, the normal amount in the middle third and the remainder in the top third.

5 ● Why is it customary to apply a correction factor to laboratory values before calculating infiltration in buildings?

Most, if not all, laboratory values have been determined by exerting a certain wind pressure across a single thickness of building construction. In actual building construction the pressure drop due to wind velocity takes place in two steps, one through the windward and the other through the leeward wall. Tests indicate that the leakage in actual construction will be about 80 per cent of laboratory values and therefore this factor has been applied in making up the tables in this chapter. The curves represent laboratory data as taken with no correction applied.

6 ● Is infiltration through walls of importance?

In the case of compound walls of good construction the heat loss due to infiltration is usually negligibly low. In the case of single thickness walls without building paper properly applied, the heat loss due to air leakage may be very high.

7 ● How are the wind velocities and outside temperatures selected for infiltration calculations in heating?

It is common practice to take the average wind velocity for the three coldest months and the temperature as 15 F above the coldest recorded by the Weather Bureau during the preceding 10 years.

8 ● Show the probable combined effects of infiltration due to temperature difference and wind velocity on the ground floor and on the 15th floor of a 20-story building 200 ft high with a 12 mph wind blowing.

At the ground floor the effective velocity is increased to.

$$M_e = \sqrt{12^2 + 1.75 \times 100} = 17.8 \text{ mph}$$

At the 15th floor level it would be reduced to:

$$M_e = \sqrt{12^2 - 1.75 \times 50} = 7.5 \text{ mph}$$

9 ● What is the value of storm sash in reducing infiltration?

The reduction depends on the relative fit of the window and the storm sash Fig 4 indicates for storm sash buttoned on a reduction at 20 mph from about 52 cfh per foot of crack to 42 cfh. Fig. 5 indicates a reduction for a storm sash buttoned over a loosely fitted window from 185 to 90 cfh.

Chapter 7

HEATING LOAD

Heat Demand Design Factors, Method of Procedure, Inside and Outside Temperatures, Wind Velocity Effects, Auxiliary Heat Sources, Wall Condensation, Heat Loss Computation

TO design any system of heating, the maximum probable heat demand must be accurately estimated in order that the apparatus installed shall be capable of maintaining the desired temperature at all times. The factors which govern this maximum heat demand—most of which are seldom, if ever, in equilibrium—include the following:

1. Outside temperature. 2. Rain or snow. 3. Sunshine or cloudiness. 4. Wind velocity.	*Outside Conditions (The Weather)*
5. Heat transmission of exposed parts of building. 6. Infiltration of air through cracks, crevices and open doors and windows. 7. Heat capacity of materials. 8. Rate of absorption of solar radiation by exposed materials.	*Building Construction*
9. Inside temperatures. 10. Stratification of air. 11. Type of heating system. 12. Ventilation requirements. 13. Period and nature of occupancy. 14. Temperature regulation.	*Inside Conditions*

The *inside conditions* vary from time to time, the physical properties of the *building construction* may change with age, and the *outside conditions* are changing constantly. Just what the worst combination of all of these variable factors is likely to be in any particular case is therefore conjectural. Because of the nature of the problem, extreme precision in estimating heat losses at any time, while desirable, is hard of attainment.

The procedure to be followed in determining the heat loss from any building can be divided into seven consecutive steps, as follows:

1. Determine on the inside air temperature, at the breathing line or the 30-in. line, which is to be maintained in the building during the coldest weather. (See Table 1.)

2. Determine on an outside air temperature for design purposes, based on the minimum temperatures recorded in the locality in question, which will provide for all but the most severe weather conditions. Such conditions as may exist for only a few consecutive hours are readily taken care of by the heat capacity of the building itself. (See Table 2.)

3. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 5.)

4. Measure up net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building.

5. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet and the temperature difference between the inside and outside air. (See Items 1 and 2.)

6. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 6.)

7. The sum of the heat losses by transmission (Item 5) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (Item 6) of the cold air entering by infiltration represents the total heat loss equivalent for any building.

Item 7 represents the heat losses after the building is heated and under stable operating conditions in coldest weather. Additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to the specified standard inside temperature.

The rate at which this additional heat is required depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated

This additional heat may be figured and allowed for as conditions re-

TABLE 1. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

Type of Building	Deg Fahr	Type of Building	Deg Fahr
SCHOOLS		THEATERS—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	HOTELS—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–68
		Toilets and service rooms	68
HOSPITALS—			
Private rooms	70–72	HOMES	70–72
Private rooms (surgical)	70–80	STORES	65–68
Operating rooms	70–95	PUBLIC BUILDINGS	68–72
Wards	68	WARM AIR BATHS	120
Kitchens and laundries	66	STEAM BATHS	110
Toilets	68	FACTORIES AND MACHINE SHOPS	60–65
Bathrooms	70–80	FOUNDRIES AND BOILER SHOPS	50–60
		PAINT SHOPS	80

[a]The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion These three factors considered together constitute what is termed the *effective temperature* (See Chapter 3)

quire, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 per cent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or have more time allowed for heating-up during the few minimum temperature days, no allowance is made except in the size of boilers or furnaces.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building and which should always be stated in the heating specifications is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 1 presents values which conform with good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 3. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H.V.E. Research Laboratory, is 66 deg.[1]

According to Fig. 6, Chapter 3, for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures listed in Table 1, consideration should be given to whether provision is to be made for humidification, and if so, the actual relative humidity to be maintained.

Temperature at Proper Level: In making the actual heat-loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 1 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level. In the case of heated spaces adjacent to unheated spaces, it will usually be sufficient to assume the temperature in such spaces as the mean between the temperature of the inside heated spaces and the outside air temperature, excepting where the combined heat transmission coefficient of the roof and ceiling can be used, in which case the usual inside and outside temperatures should be applied. (See discussion regarding the use of combined coefficients of pitched roofs, unheated attics and top-floor ceilings Chapter 5.)

[1]See Chapter 3, p 61.

High Ceilings: Research data concerning stratification of air in buildings are lacking, but in general it may be said that where the increase in temperature is due to the natural tendency of the warmer or less dense air to rise, as where a direct radiation system is installed, the temperature of the air at the ceiling increases with the ceiling height. The relation, however, is not a straight-line function, as the amount of increase per foot of height apparently decreases as the height of the ceiling increases, according to present available information[2].

Where ceiling heights are under 20 ft, it is common engineering practice to consider that the Fahrenheit temperature increases 2 per cent for each foot of height above the breathing line. This rule, sufficiently accurate for most cases, will give the probable air temperature at any given level for a room heated by direct radiation. Thus, the probable temperature in a room at a point 3 ft above the breathing line, if the breathing line temperature is 70 F, will be

$$(1.00 + 3 \times .02)\ 70 = 74.2 \text{ F.}$$

With certain types of heating and ventilating systems, which tend to oppose the natural tendency of warm air to rise, the temperature differential between floor and ceiling can be greatly reduced. These include unit heaters, fan-furnace heaters, and the various types of mechanical ventilating systems. The amount of reduction is problematical in certain instances, as it depends upon many factors such as location of heaters, air temperature, and direction and velocity of air discharge. In some cases it has been possible to reduce the temperature between the floor and ceiling by a few degrees, whereas, in other cases, the temperature at the ceiling has actually been increased because of improper design, installation or operation of equipment. So much depends upon the factors enumerated that it is not advisable to allow less than 1 per cent per foot (and usually more) above the breathing line in arriving at the air temperature at any given level for any of these types of heating and ventilating systems, unless the manufacturers are willing to guarantee that the particular type of equipment under consideration will maintain a smaller temperature differential for the specific conditions involved.

Temperature at Floor Level: In determining mean air temperatures just above floors which are next to ground or unheated spaces, a temperature 5 deg lower than the breathing-line temperature may be used, provided the breathing-line temperature is not less than 55 F.

OUTSIDE TEMPERATURES

The outside temperature used in computing the heat loss from a building is seldom taken as the lowest temperature ever recorded in a given locality. Such temperatures are usually of short duration and are rarely repeated in successive years. It is therefore evident that a temperature somewhat higher than the lowest on record may be properly assumed in making the heat-loss computations.

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[2]A.S H.V E REPORT No 958—Temperature Gradient Observations in a Large Heated Space, by G L. Larson, D W. Nelson and O. C Cromer (A S H V E TRANSACTIONS, Vol 39, 1933, p 243)
A S H V E REPORT No 1011—Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson and John James (A S H V E. TRANSACTIONS, Vol 41, 1935, p 185).

The outside temperature to be assumed in the design of any heating system is ordinarily not more than 15 deg above the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In the case of massive and well insulated buildings in localities where the minimum does not prevail for more than a few hours, or where the lowest recorded temperature is extremely unusual, more than 15 deg above the minimum may be allowed, due primarily to the *fly-wheel* effect of the heat capacity of the structure. The outside temperature assumed and used in the design should always be stated in the heating specifications. Table 2 lists the coldest dry-bulb temperatures ever recorded by the Weather Bureau at the places listed.

If Weather Bureau reports are not available for the locality in question, then the reports for the station nearest to this locality are to be used, unless some other temperature is specifically stated in the specifications. In computing the average heat transmission losses for the heating season in the United States the average outside temperature from October 1 to May 1 should be used.

WIND VELOCITY EFFECTS

The effect of wind on the heating requirements of any building should be given consideration under two heads:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.
2. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves, if such materials are at all porous.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to work out the heating load for a building for several different combinations of temperature and wind velocity which records show to have occurred and to select the worst combination; but designers generally do not feel that such a degree of refinement is justified. Therefore, pending further studies of actual buildings, it is recommended that the average wind movement in any locality during December, January and February be provided for in computing (1) the heat transmission of a building, and (2) the heat required to take care of the infiltration of outside air.

The first condition is readily taken care of, as explained in Chapter 5, by using a surface coefficient f_o for the outside wall surface which is based on the proper wind velocity. In case specific data are lacking for any given locality, it is sufficiently accurate to use an average wind velocity of approximately 15 mph which is the velocity upon which the heat transmission coefficient tables in Chapter 5 are based.

In a similar manner, the heat allowance for infiltration through cracks

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]

COL A	COL B	COL. C	COL D	COL E	COL F
State	City	Average Temp., Oct 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
Ala.	Birmingham	53.8	−10	8.5	N
	Mobile	58.9	−1	10.4	N
Ariz.	Flagstaff	35.8	−25	7.8	SW
	Phoenix	59.5	12	6.4	E
Ark.	Fort Smith	50.4	−15	8.1	E
	Little Rock	51.6	−12	8.7	NW
Calif.	Los Angeles	58.5	28	6.3	NE
	San Francisco	54.2	27	7.6	N
Colo.	Denver	38.9	−29	7.5	S
	Grand Junction	38.9	−21	5.3	NW
Conn.	New Haven	38.4	−15	9.7	N
D. C.	Washington	43.4	−15	7.1	NW
Fla.	Jacksonville	62.0	10	9.2	NE
Ga.	Atlanta	51.5	−8	12.1	NW
	Savannah	58.5	8	9.5	NW
Idaho	Lewiston	42.3	−23	5.3	E
	Pocatello	35.7	−28	9.6	SE
Ill.	Chicago	36.4	−23	12.5	W
	Springfield	39.8	−24	10.1	NW
Ind.	Evansville	45.1	−16	9.8	S
	Indianapolis	40.3	−25	11.5	SW
Iowa	Dubuque	33.9	−32	7.1	NW
	Sioux City	32.6	−35	11.6	NW
Kans.	Concordia	39.8	−25	8 1	S
	Dodge City	41.4	−26	9 8	NW
Ky.	Louisville	45.3	−20	9.9	SW
La.	New Orleans	61.6	7	8.8	N
	Shreveport	56.2	−5	8.9	SE
Me.	Eastport	31.5	−23	12.0	W
	Portland	33.8	−21	9.2	NW
Md.	Baltimore	43.8	−7	7.8	NW
Mass.	Boston	38.1	−18	11 2	W
Mich.	Alpena	29.6	−28	12.4	W
	Detroit	35.8	−24	12.7	SW
	Marquette	28.3	−27	11 1	NW
Minn.	Duluth	24.3	−41	12.6	SW
	Minneapolis	29.4	−33	11 3	NW
Miss.	Vicksburg	56.8	−1	8.3	SE
Mo.	St. Joseph	40.7	−24	9 3	NW
	St. Louis	43 6	−22	11 6	S
	Springfield	44.3	−29	10.8	SE
Mont.	Billings	34.0	−49	....	W
	Havre	27.6	−57	9.5	SW
Nebr.	Lincoln	37.0	−29	10.5	S
	North Platte	35.4	−35	8.5	W
Nev.	Tonopah	39.4	−10	10.0	SE
	Winnemucca	37.9	−28	8.7	NE
N. H.	Concord	33.3	−35	6.6	NW
N. J.	Atlantic City	41.6	−9	15.9	NW
N. Y.	Albany	35.2	−24	8.1	S
	Buffalo	34.8	−20	17.2	W
	New York	40.7	−14	17.1	NW

[a]United States data from U. S. Weather Bureau
Canadian data from Meteorological Service of Canada.

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]—(Concluded)

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F
State or Province	City	Average Temp., Oct 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
N. M.	Santa Fe	38.3	−13	7.8	NE
N. C.	Raleigh	50.0	−2	8.2	SW
	Wilmington	54.2	5	8.5	SW
N. Dak.	Bismarck	24.6	−45	9.1	NW
	Devils Lake	20.3	−44	10.6	W
Ohio	Cleveland	37.2	−17	13.0	SW
	Columbus	39.9	−20	12.0	SW
Okla.	Oklahoma City	47.9	−17	12.0	N
Oreg.	Baker	35.2	−24	6.9	SE
	Portland	46.1	−2	7.5	S
Pa.	Philadelphia	42.7	−6	11.0	NW
	Pittsburgh	41.0	−20	11.7	W
R. I.	Providence	37.2	−17	12.8	NW
S. C.	Charleston	57.4	7	10.6	SW
	Columbia	54.0	−2	8.1	NE
S. Dak.	Huron	28.2	−43	10.6	NW
	Rapid City	33.4	−34	8.2	W
Tenn.	Knoxville	47.9	−16	7.8	SW
	Memphis	51.1	−9	9.7	S
Texas	El Paso	53.5	−5	10.4	NW
	Fort Worth	55.2	−8	10.4	NW
	San Antonio	60.6	4	8.0	NE
Utah	Modena	36.3	−24	8.8	W
	Salt Lake City	40.0	−20	6.7	SE
Vt.	Burlington	31.5	−29	11.8	S
Va.	Lynchburg	46.8	−7	7.1	NW
	Norfolk	49.3	2	12.5	N
	Richmond	47.0	−3	7.9	SW
Wash.	Seattle	44.8	3	11.3	SE
	Spokane	37.7	−30	7.1	SW
W. Va.	Elkins	39.4	−28	6.6	W
	Parkersburg	42.6	−27	7.5	SW
Wis.	Green Bay	30.0	−36	10.4	SW
	La Crosse	31.7	−43	7.3	S
	Milwaukee	33.4	−25	11.5	W
Wyo.	Lander	30.0	−40	5.0	SW
	Sheridan	30.7	−41	6.0	NW
Alta.	Edmonton	23.0	−57	6.5	SW
B. C.	Vancouver	42.0	2	4.5	E
	Victoria	43.9	− 1.5	12.5	N
Man.	Winnipeg	17.5	−47	10.0	NW
N. B.	Fredericton	27.0	−35	9.6	NW
N. S.	Yarmouth	35.0	−12	14.2	NW
Ont.	London	32.6	−27	10.3	SW
	Ottawa	26.5	−34	8.4	NW
	Port Arthur	22.4	−37	7.8	NW
	Toronto	32.9	−26.5	13.0	SW
P. E. I.	Charlottetown	29.0	−27	9.4	SW
Que.	Montreal	27.8	−29	14.3	SW
	Quebec	24.2	−34	13.6	SW
Sask.	Prince Albert	15.8	−70	5.1	W
Yukon	Dawson	2.1	−68	3.7	S

[a]United States data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

and walls (Tables 1 and 2, Chapter 6) must be based on the proper wind velocity for a given locality. In the case of *tall buildings* special attention must be given to infiltration factors. (See Chapter 6).

In the past many designers have used empirical *exposure factors* which were arbitrarily chosen to increase the calculated heat loss on the side or sides of the building exposed to the prevailing winds. It is also possible to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities. Recent investigations show, however, that the wind direction indicated by Weather Bureau instruments does not always correspond with the direction of actual impact on the building walls, due to deflection by surrounding buildings.

The exposure factor, which is still in use by many engineers, is usually taken as 15 per cent, and is added to the calculated heat loss on the side or sides exposed to what is considered the prevailing winter wind. There is a need for actual test data on this point, and pending the time when it can be secured, the question must be left to the judgment of the designing engineer. It should be remembered that the values of U in the tables in Chapter 5 are based on a wind velocity of 15 mph and that the infiltration figures are supposed to be selected from the tables in Chapter 6 to correspond to the wind velocities given in Table 2 of the present chapter.

The *Heating, Piping and Air Conditioning Contractors National Association* has devised a method[3] for calculating the square feet of equivalent direct radiation required in a building. This method makes use of exposure factors which vary according to the geographical location and the angular situation of the construction in question in reference to prevailing winds and the velocity of them.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Electric Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is

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[3]See Standards of *Heating, Piping and Air Conditioning Contractors National Association*

retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In the first case the Btu supplied per hour $= \frac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2546$, and in the second case Btu per hour = bhp $\times$ 2546, in which 2546 is the Btu equivalent of 1 hp-hour. In high-powered mills this is the chief source of heating and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

The heat (in Btu per hour) from electric lamps is obtained by multiplying the watts per lamp by the number of lamps and by 3.415. One cubic foot of producer gas gives off about 150 Btu per hour; one cubic foot of illuminating gas gives off about 535 Btu per hour; and one cubic foot of natural gas gives off about 1000 Btu per hour. A Welsbach burner averages 3 cu ft of gas per hour and a fish-tail burner, 5 cu ft per hour. For information concerning the heat supplied by persons, see Chapter 3.

In intermittently heated buildings, besides the capacity necessary to care for the normal heat loss which may be calculated according to customary rules, additional capacity should be provided to supply the heat necessary to warm up the cold material of the interior walls, floors, and furnishings. Tests have shown that when a cold building has had its temperature raised to about 60 F from an initial condition of about 0 F, the heat absorbed from the air by the material in the structure may vary from 50 per cent to 150 per cent of the normal heat loss of the building. It is therefore necessary, in order to heat up a cold building within a reasonable length of time, to provide such additional capacity. If the interior material is cold when people enter a building, the radiation of heat from the occupants to the cold material will be greater than is normal and discomfort will result. (See Chapter 3.)

WALL CONDENSATION

Condensation in the interior surfaces[4] of buildings may cause irreparable damage to manufactured articles and machinery. It often results in short-circuiting of electric power, and causes disintegration of roof structures not properly protected.

The prevalence of moisture on a surface is caused by the contact of the warm humid air in a building with surfaces below the dew-point temperature. It can be eliminated by (1) raising the surface temperature with increased air velocities passing over the surface, or adding a sufficient thickness of insulation, and (2) by lowering the humidity which is often not possible due to manufacturing processes.

The condensation of moisture within walls[5] is an important problem with many types of construction under adverse conditions. The tempera-

[4]Preventing Condensation on Interior Building Surfaces, by Paul D Close (A S H V E. TRANSACTIONS, Vol 36, 1930, p 153)

[5]Condensation within Walls, by F B Rowley, A. B Algren and C E Lund (A.S.H V E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January, 1938)

tures of the various parts of a wall are controlled by the type and amount of insulation used and the vapor densities in the corresponding sections are controlled by the type of vapor barriers installed. The transmission of heat and vapor through a wall should be considered together, and in most cases the proper combination of insulation and vapor barriers will eliminate the possibilities of condensation within walls. A consideration often overlooked in problems of condensation within walls is that a vapor barrier should be placed on the warm side and not on the cold side of a wall.

HEAT LOSS COMPUTATION EXAMPLE

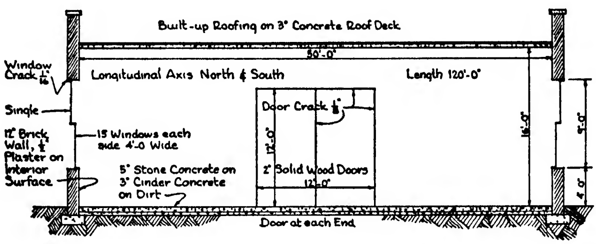

FIG. 1. ELEVATION OF FACTORY BUILDING

1. LOCATION .... Philadelphia, Pa

2. LOWEST OUTSIDE TEMPERATURE. (Table 2) .... − 6 F

3. BASE TEMPERATURE: *In this example* a design temperature 10 F above lowest on record instead of 15 F is used. Hence the base temperature =

$$(-6 + 10) = +4 \text{ F}.$$

4. DIRECTION OF PREVAILING WIND (during Dec., Jan., Feb.) .... Northwest

5. BREATHING-LINE TEMPERATURE (5 ft from floor) .... 60 F

6. INSIDE AIR TEMPERATURE AT ROOF:

   The air temperature just below *roof* is higher than at the breathing line. Height of roof is 16 ft, or it is 16 − 5 = 11 ft above breathing line. Allowing 2 per cent per foot above 5 ft, or 2 × 11 = 22 per cent, makes the temperature of the air under the roof = 1.22 × 60 = 73.2 F.

7. INSIDE TEMPERATURE AT WALLS:

   The air temperature at the mean height of the *walls* is greater than at the breathing line. The mean height of the walls is 8 ft and allowing 2 per cent per foot above 5 ft, the average mean temperature of the walls is 1.06 × 60 = 63.6 F. By similar assumptions and calculations, the mean temperature of the *glass* will be found to be 64.2 F and that of the *doors* 61.2 F.

8. AVERAGE WIND VELOCITY (Table 2) .... 11.0 mph

9. OVER-ALL DIMENSIONS (See Fig. 1) .... 120 x 50 x 16 ft

10. CONSTRUCTION:

    Walls—12-in. brick, with ½-in. plaster applied directly to inside surface.

    Roof—3-in. stone concrete and built-up roofing.

Floor—5-in. stone concrete on 3-in. cinder concrete on dirt.

Doors—One 12 ft x 12 ft wood door (2 in. thick) at each end.

Windows—Fifteen, 9 ft x 4 ft single glass double-hung windows on each side.

11. TRANSMISSION COEFFICIENTS:

Walls—(Table 3, Chapter 5, Wall 2B) .... $U = 0.34$
Roof—(Table 11, Chapter 5, Roofs 2A and 3A) .... $U = 0.77$
Floor—(Table 10, Chapter 5, Floors 5A and 6A) .... $U = 0.63$
Doors—(Table 13B, Chapter 5) .... $U = 0.46$
Windows—(Table 13A, Chapter 5) .... $U = 1.13$

12. INFILTRATION COEFFICIENTS:

Windows—Average windows, non-weatherstripped, $\frac{1}{16}$-in. crack and $\frac{3}{64}$-in. clearance. The leakage per foot of crack for an 11-mile wind velocity is 25.0 cfh. (Determined by interpolation of Table 2, Chapter 6.) The heat equivalent per hour per degree per foot of crack is taken from Chapter 6.

$25.0 \times 0.018 = 0.45$ Btu per deg Fahrenheit per foot of crack.

Doors—Assume infiltration loss through door crack twice that of windows or $2 \times 0.45 = 0.90$ Btu per deg Fahrenheit per foot of crack.

Walls—As shown by Table 1, Chapter 6, a plastered wall allows so little infiltration that in this problem it may be neglected.

13. CALCULATIONS: See calculation sheet, Table 3.

TABLE 3. CALCULATION SHEET SHOWING METHOD OF ESTIMATING HEAT LOSSES OF BUILDING SHOWN IN FIG. 1

PART OF BUILDING	WIDTH IN FEET	HEIGHT IN FEET	NET SURFACE AREA OR CRACK LENGTH	COEFFICIENT	TEMP DIFF	TOTAL BTU
North Wall						
Brick, ½-in plaster	50	16	656	0.34	59.6	13,293
Doors (2-in wood)	12	12	144	0.46	57.2	3,789
⅛ in Crack	1 pair doors		60	0.90	57.2	1,544[a]
West Wall:						
Brick ½-in plaster	120	16	1380	0.34	59.6	27,964
Glass (Single)	15 x 4	9	540	1.13	60.2	36,734
$\frac{1}{16}$ in Crack	Double Hung Windows (15)		450	0.45	60.2	6,095[a]
South Wall	Same as North Wall					18,626
East Wall	Same as West Wall					70,793
Roof, 3-in concrete and slag-surfaced built-up roofing	50	120	6000	0.77	69.2	319,704
Floor, 5-in stone concrete on 3-in. cinder concrete	50	120	6000	0.63	5[b]	18,900
GRAND TOTAL of heat required for building in Btu per hour						517,442

[a]This building has no partitions and whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, only one-half of the total crack will be used in computing infiltration for each side and each end of building.

[b]A 5 F temperature differential is commonly assumed to exist between the air on one side of a large floor laid on the ground and the ground

PROBLEMS IN PRACTICE

1 ● What is the relation between the sensible heat loss from a building and the heat required for humidification?

A house with a volume of 14,000 cu ft has a heat loss 120 Mbh for standard uninsulated frame construction and a 70 F temperature difference. Assuming a leakage rate of 1½ air changes per hour it would require about 10 Mbh to maintain a relative humidity of 45 per cent when the outside air is 0 F and 50 per cent relative humidity. By using an insulation such as rock wool, the sensible heat loss of this house may be reduced to approximately 77 Mbh. The insulation does not affect the humidification load, which now assumes greater importance.

2 ● What inside dry-bulb temperatures are usually assumed for: (a) homes, (b) schools, (c) public buildings?

Referring to Table 1:

a. 70 to 72 F.

b. Temperature varies from 55 to 75 F, depending on the room. Classrooms, for instance, are usually specified as 70 to 72 F.

c. 68 to 72 F.

3 ● How is the outside temperature selected for use in computing heat losses?

The outside temperature used in computing heat losses is generally taken from 10 to 15 F higher than the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In some cases where the lowest recorded temperature is extremely unusual, the design temperature is taken even higher than 15 F above the lowest recorded temperature.

4 ● What are the effects of wind movement on the heating load?

a. Wind movement increases the heat transmission of walls, glass, and roof, it affects poor walls to a much greater extent than good walls.

b. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves if such materials are at all porous

5 ● Calculate the heat given off by eighteen 200-watt lamps.

$200 \times 18 \times 3.415 = 12{,}294$ Btu per hour.

6 ● A two-story, six-room, frame house, 28-ft by 30-ft foundation, has the following proportions:

Area of outside walls, 1992 sq ft.
Area of glass, 333 sq ft.
Area of outside doors, 54 sq ft.
Cracks around windows, 440 ft.
Cracks around doors, 54 ft.
Area of second floor ceiling, 783 sq ft.
Volume, first and second floors, 13,010 cu ft.
Ceilings, 9 ft high.

The minimum temperature for the heating season is −34 F, and the required inside temperature at the 30-in. level is 70 F. The average number of degree days for a heating season is 7851, and the average wind velocity is 10 mph, northwest.

The walls are constructed of 2-in. by 4-in. studs with wood sheathing, building paper, and wood siding on the outside, and wood lath and plaster on the inside. Windows are single glass, double-hung, wood, without weatherstrips. The second floor ceiling is metal lath and plaster, without an attic floor. The roof is of wood shingles on wood strips with rafters exposed. The area of the roof is 20 per cent greater than the area of the ceiling. Select values for the following: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for roof;

(e) U for ceiling and roof combined; (f) air leakage, cubic feet per hour per foot of window crack; (g) air leakage, cubic feet per hour per foot of door crack.

a. 0.25 (Table 5, Chapter 5).
b. 1.13 (Table 13, Chapter 5).
c. 0.69 (Table 8, Chapter 5).
d. 0.46 (Table 12, Chapter 5).
e. 0.31 (Equation 6, Chapter 5).
f. 21.4 (Table 2, Chapter 6).
g. 42.8, which is double the window leakage.

7 ● Using the data of Question 6, calculate the maximum Btu loss per hour for the various constructions, and show the percentage of the total heat which is lost through each construction described.

Assume 2 per cent rise in temperature for each foot in height. The average temperature will be 72.8 F for walls, doors, and windows, and 79.1 F for the second floor ceiling

a. Outside walls	46,200 Btu loss	35 4 per cent of total
b. Glass	34,950 Btu loss	26 7 per cent of total
c. Doors	5,670 Btu loss	4.4 per cent of total
d. Second floor ceiling	24,050 Btu loss	18.4 per cent of total
e. Air leakage, windows	15,750 Btu loss	12.1 per cent of total
f. Air leakage, doors	3,865 Btu loss	3 0 per cent of total
Total	130,485 Btu loss	100.0 per cent of total

8 ● For the house in Question 6, place 1-in. insulation in the outside walls and second floor ceiling; k for insulation = 0.34. Use weatherstrip on doors and windows, and double glass on the windows; C = 0.55. Calculate or select the following values: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for combination of ceiling and roof; (e) air leakage, cubic feet per hour per foot of door crack; (f) air leakage, cubic feet per hour per foot of window crack.

a. 0.144
b. 0.55
c. 0.23
d. 0 16
e. 15.5
f. 31.0

9 ● Calculate the maximum Btu loss per hour and show the percentage loss by each channel for the house as insulated in Question 8.

a. Outside walls	26,650 Btu loss	35 1 per cent of total
b. Glass	17,000 Btu loss	22 4 per cent of total
c. Doors	5,670 Btu loss	7 4 per cent of total
d. Ceiling	12,420 Btu loss	16 4 per cent of total
e. Air leakage, windows	11,400 Btu loss	15 1 per cent of total
f. Air leakage, doors	2,795 Btu loss	3 6 per cent of total
Total	75,935 Btu loss	100.0 per cent of total

10 ● From the results of Questions 7 and 9, calculate the Btu saved and the percentage saved by each change in construction.

	UNINSULATED	INSULATED	BTU SAVED	PER CENT SAVED
a. Outside walls	46,200	26,650	19,550	42 3
b. Glass	34,950	17,000	17,950	51.4
c. Doors	5,670	5,670	0	0
d. Ceiling	24,050	12,420	11,630	48.3
e. Air leakage, windows	15,750	11,400	4,350	27.6
f. Air leakage, doors	3,865	2,795	1,070	27.7

11 ● From the results of Questions 7 and 9, calculate the heat loads per heating season in Btu and note the savings by better construction.

The 7851 degree days for the heating season multiplied by 24 hours, times the Btu loss per hour for 1 F drop in temperature gives the Btu load per heating season.

Saving = 262,000,000 − 152,500,000 = 109,500,000 Btu.

12 ● The dry-bulb temperature and the relative humidity at the ceiling of a mixing room in a bakery are 80 F and 60 per cent, respectively. The roof is a 4-in. concrete deck covered with built-up roofing. If the lowest outside temperature to be expected is — 10 F, what thickness of rigid fiber insulation will be required to prevent condensation?

From Table 11, Chapter 5, U for the uninsulated roof = 0.72. From Table 2, Chapter 5, k for rigid fiber insulation = 0.33. From the psychrometric chart the dew-point of air at 80 F and 60 per cent relative humidity is 65 F. The ceiling temperature, therefore, must not drop below 65 F if condensation is to be prevented.

When equilibrium is established, the amount of heat flowing through any component part of a construction is the same for each square foot of area.

Therefore,

$$U\,[80 - (-10)\,] = 1.65\,(80 - 65)$$

where

U is the transmittance of the insulated roof.

Solving the equation, $U = 0.275$.

The resistance of the insulated roof $= \dfrac{1}{0.275} = 3.64.$

The resistance of the uninsulated roof $= \dfrac{1}{0.72} = 1.39.$

The resistance of the insulation = 3.64 − 1.39 = 2.25.

Resistance per inch of insulation $= \dfrac{1}{0.33} = 3.0.$

Since a resistance of 2.25 is required, and 1 in. of insulation has a resistance of 3, one inch will be sufficient to prevent condensation.

The same result might have been obtained by selecting an insulated 4-in. concrete slab having a U of less than 0.275 from Table 11, Chapter 5. This 4-in. concrete slab with 1-in. rigid insulation has a U of 0.23 which is safe.

Chapter 8

COOLING LOAD

Conditions of Comfort, Design Outside Temperatures, Components of Heat Gain, Normal Heat Transmission, Solar Heat Transmission, Sun Effect Through Windows, Heat Emission of Occupants, Heat Introduced by Outside Air, Heat Emission of Appliances

LOAD calculations for summer air conditioning are more complicated than heating load calculations for the reason that there are more factors to be considered. Because of the variable nature of some of the contributing load components and the fact that they do not necessarily impose their maximum effect simultaneously, considerable care must be exercised in determining their phase relationship in order that equipment of proper capacity may be selected to maintain specified indoor conditions.

CONDITIONS OF COMFORT

The conditions to be maintained in an enclosure are variable and depend upon several factors, especially the outside design conditions, duration of occupancy and relationship between air motion, dry-bulb and wet-bulb temperatures. Information concerning the proper effective temperature to be maintained is given in Chapter 3, where are also tabulated the most desirable indoor conditions to be maintained in summer for exposures over 40 min (see Table 2, Chapter 3).

DESIGN OUTSIDE TEMPERATURES

Summer dry-bulb and wet-bulb temperatures of various cities are given in Table 1. It will be noted that the temperatures are not the maximums but the design temperatures which should be used in air conditioning calculations. The maximum outside wet-bulb temperatures as given in Weather Bureau reports usually occur only from 1 to 4 per cent of the time, and they are therefore of such short duration that it is not practical to design a cooling system covering this range. The temperatures shown in Table 1 are in part based on available design conditions known to be successfully applied and for those localities where this information is lacking they are based on a study of the hourly temperatures in New York City from which factors were derived and applied to the average maximum dry- and wet-bulb temperatures for other cities. This study covered a twenty-year record of Weather Bureau temperatures. The design

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER

State	City	Design Dry-Bulb	Design Wet-Bulb	Summer Wind Velocity MPH	Prevailing Summer Wind Direction
Ala	Birmingham	95	78	5.2	S
	Mobile	95	80	8.6	SW
Ariz	Phoenix	105	76	6.0	W
Ark	Little Rock	95	78	7.0	NE
Calif	Los Angeles	90	70	6.0	SW
	San Francisco	90	65	11.0	SW
Colo	Denver	95	64	6.8	S
Conn	New Haven	95	75	7.3	S
Dela	Wilmington	95	78	9.7	SW
D. C	Washington	95	78	6.2	S
Fla	Jacksonville	95	78	8.7	SW
	Tampa	94	79	7.0	E
Ga	Atlanta	95	76	7.3	NW
	Savannah	95	78	7.8	SW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
	Peoria	95	76	8.2	S
Ind	Indianapolis	95	76	9.0	SW
Iowa	Des Moines	95	77	6.6	SW
Kansas	Wichita	100	75	11.0	S
Ky	Louisville	95	76	8.0	SW
La	New Orleans	95	79	7.0	SW
	Shreveport	100	78	6.2	S
Maine	Portland	90	73	7.3	S
Md	Baltimore	95	78	6.9	SW
Mass	Boston	92	75	9.2	SW
Mich	Detroit	95	75	10.3	SW
Minn	Minneapolis	95	75	8.4	SE
Miss	Vicksburg	95	78	6.2	SW
Mo	Kansas City	100	76	9.5	S
	St. Louis	95	78	9.4	SW
Mont	Helena	95	67	7.3	SW
Nebr	Lincoln	95	75	9.3	S
Nev	Reno	95	65	7.4	W
N H	Manchester	90	73	5.6	NW
N. J	Trenton	95	78	10.0	SW
N. Y	Albany	92	75	7.1	S
	Buffalo	93	75	12.2	SW
	New York	95	75	12.9	SW
N. M	Santa Fe	90	65	6.5	SE
N. C	Asheville	90	75	5.6	SE
	Wilmington	95	79	7.8	SW
N. Dak	Bismarck	95	73	8.8	NW
Ohio	Cincinnati	95	78	6.6	SW
	Cleveland	95	75	9.9	S
Okla	Oklahoma City	101	76	10.1	S
Oreg	Portland	90	65	6.6	NW
Pa	Philadelphia	95	78	9.7	SW
	Pittsburgh	95	75	9.0	NW
R. I	Providence	93	75	10.0	NW
S. C	Charleston	95	80	9.9	SW
	Greenville	95	76	6.8	NE
S. Dak	Sioux Falls	95	75	7.6	S
Tenn	Chattanooga	95	77	6.5	SW
	Memphis	95	78	7.5	SW

TABLE 1. DESIGN DRY AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Concluded)

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Texas	Dallas	100	78	9 4	S
	El Paso	100	69	6 9	E
	Galveston	95	80	9 7	S
	Houston	95	78	7 7	S
	San Antonio	100	78	7 4	SE
Utah	Salt Lake City	92	63	8 2	SE
Vt	Burlington	90	73	8 9	S
Va	Norfolk	95	78	10 9	S
	Richmond	95	78	6 2	SW
Wash	Seattle	85	65	7.9	S
	Spokane	90	65	6 5	SW
W Va	Parkersburg	95	75	5 3	SE
Wisc	Madison	95	75	8 1	SW
	Milwaukee	95	75	10 4	S
Wyo	Cheyenne	95	65	9 2	S

temperatures given are not exceeded more than 5 to 8 per cent of the time during a cooling season of 1200 hours in June, July, August and September for an average year.

COMPONENTS OF HEAT GAIN

A cooling load determination is composed of five components which may be classified in the following manner:

1 Normal heat transfer through windows, walls, partitions, doors, floors, ceilings, etc.

2. Transfer of solar radiation through windows, walls, doors, skylights, or roof

3. Heat emission of occupants within enclosures.

4 Heat introduced by infiltration of outside air or controlled ventilation.

5 Heat emission of mechanical, chemical, gas, steam, hot water and electrical appliances located within enclosures

The components of heat gain, classified by source are further classified as sensible and latent heat gain.

The first two components fall into the classification of sensible heat gain, that is, they tend to raise the temperature of the air within the structure. The last three components not only produce sensible heat gain but they may also tend to increase the moisture content of the air within the structure.

Normal Heat Transmission

By normal heat transmission, as distinguished from solar heat transmission is meant the transmission of heat through windows, walls, partitions, etc. from without to interior of enclosure by virtue of difference between outside and inside air temperatures. This load is calculated in a

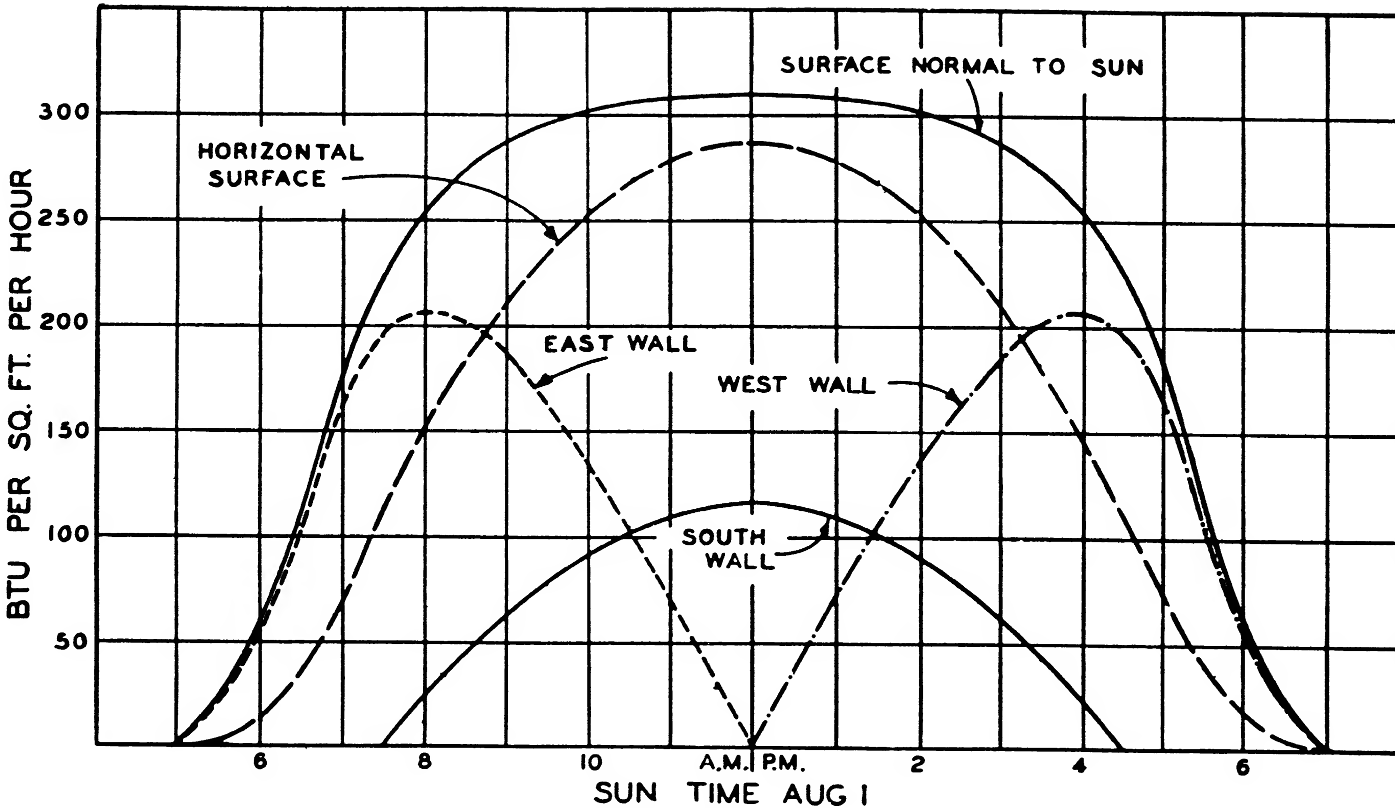

FIG. 1. CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1

manner similar to that described in Chapter 7 (except that flow of heat is reversed) by means of the formula:

$$H_t = A\,U\,(t_o - t) \tag{1}$$

where

H_t = heat transmitted through the material of wall, glass, floor, etc., Btu per hour.
A = net inside area of wall, glass, floor, etc., square feet.
t = inside temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
U = coefficient of transmission of wall, glass, floor, etc., Btu per hour per square foot per degree Fahrenheit difference in temperature (Tables 3 to 13, Chapter 5).

Solar Heat Transmission

Calculations of the solar heat transmitted through walls and roofs are difficult to determine because of periodical character of heat flow and time lag due to heat capacity of construction.

The variation in solar intensity normal to sun in Btu per square foot per hour on a horizontal surface, and on east, west, and south walls is given in Fig. 1. The curves are drawn from A.S.H.V.E. Laboratory data obtained by pyrheliometer, are based on sun time and apply for a perfectly clear day on August 1 at a north latitude of 40 deg. A study of these curves discloses the periodic relationship and wide variation in solar intensity on various surfaces. It will be observed that both the roof and south wall radiation curves are in exact phase relationship with each other and that whereas the east and west wall radiation curves overlap those for roof and south wall, they do not overlap each other. This phase relationship has an important bearing on the cooling load. Failure to consider the periodical character of heat flow resulting from diurnal movement of the sun and the lag due to heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall, may result in a large error in load calculations.

The values of solar intensity appearing in Fig. 1 must not be confused with the actual heat transmission through the wall for much of the solar radiation impinging against the outer surface fails to pass through the wall. Instead it is delivered to the outside air by reflection, radiation, convection and conduction. A mathematical solution for the determination of solar heat transmission has been developed but the equations involved are too complex for practical application.[1] From results of this investigation and earlier studies,[2] the Research Laboratory has prepared Tables 2, 3, 4 and 5 which give the solar intensity (I) for various hours of the day on walls of various orientations and horizontal surfaces. These values are shown for north latitudes from 30 to 45 deg.

Since the amount of solar intensity actually transmitted through a surface depends upon the nature of the exterior surface of wall or roof construction, it is necessary, in order to determine actual amount of solar

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[1]A S H V E RESEARCH REPORT No 923—Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F C Houghten, J L Blackshaw, E W. Pugh and Paul McDermott(A S H V E TRANSACTIONS, Vol 38, 1932, p 231)

[2]A S H V E RESEARCH REPORT No 853—Absorption of Solar Radiation in its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F C Houghten and Carl Gutberlet (A S H V E TRANSACTIONS, Vol 36, 1930, p 137)

TABLE 2. SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS, AND A HORIZONTAL SURFACE.

For 30 Deg Latitude on the twenty-first of July.

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4:59	0	0	0					0
5:00	1	1	0 3					0 01
6:00	47	51	24					9
7:00	136	160	90					68
8:00	151	205	136					147
9:00	127	189	140	8				214
10:00	79	141	122	31				265
11:00	21	78	85	45				296
12:00			36	50	36			305
1:00				45	85	78	21	296
2:00				31	122	141	79	265
3:00				8	140	189	127	214
4:00					136	205	151	147
5:00					90	160	136	68
6:00					24	51	47	9
7:00					0 3	1	1	0 01
7:01					0	0	0	0

TABLE 3. SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS, AND A HORIZONTAL SURFACE.

For 35 Deg Latitude on the twenty-first of July.

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4:46	0	0	0					0
5:00	9	9	3					0 01
6:00	67	72	35					15
7:00	142	174	103					77
8:00	150	209	145					151
9:00	118	191	154	26				214
10:00	60	143	139	55				264
11:00	2	75	103	72				291
12:00			55	78	55			300
1:00				72	103	75	2	291
2:00				55	139	143	60	264
3:00				26	154	191	118	214
4:00					145	209	150	151
5:00					103	174	142	77
6:00					35	72	67	15
7:00					3	9	9	0 01
7:14					0	0	0	0

TABLE 4. SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS, AND A HORIZONTAL SURFACE.

For 40 Deg Latitude on the twenty-first of July.

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4.31	0	0	0					0
5.00	14	14	5					1
6 00	72	80	40					19
7·00	143	180	112					82
8:00	143	211	155	8				152
9.00	104	192	168	46				213
10.00	46	143	156	77				258
11.00		75	121	95	15			284
12.00			73	103	73			293
1:00			15	95	121	75		284
2.00				77	156	143	46	258
3:00				46	168	192	104	213
4 00				8	155	211	143	152
5 00					112	180	143	82
6 00					40	80	72	19
7 00					5	14	14	1
7 29					0	0	0	0

TABLE 5. SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS, AND A HORIZONTAL SURFACE

For 45 Deg Latitude on the twenty-first of July.

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4 26	0	0	0					0
5 00	25	24	9					2
6 00	89	99	52					26
7 00	149	194	125					90
8 00	140	219	171	22				156
9·00	92	194	183	65				210
10·00	33	144	171	98				251
11:00		75	139	121	32			274
12:00			91	128	91			282
1·00			32	121	139	75		274
2.00				98	171	144	33	251
3:00				65	183	194	92	210
4.00				22	171	219	140	156
5:00					125	194	144	90
6:00					52	99	89	26
7:00					9	24	25	2

heat transmission, to apply correction factors to the values of (I). Solar radiation factors and solar absorption coefficients have been determined[3] as indicated in Fig. 2 and Table 6 respectively.

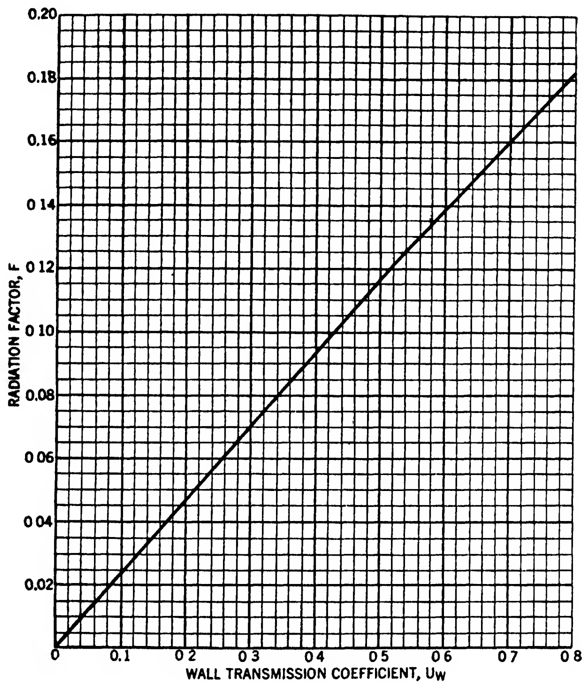

FIG. 2. SOLAR RADIATION FACTORS

The solar heat conduction through a wall or roof exposed to the sun may be expressed by the formula:

$$H_R = A\ F\ a\ I \tag{2}$$

where

H_R = Solar heat transmission, Btu per hour.
A = Area of wall or roof, square feet.
F = Percentage (expressed as a decimal) of the absorbed solar radiation which is transmitted to the inside (Fig 2)
a = Percentage (expressed as a decimal) of the incident solar radiation which is absorbed by the surface (Table 6)
I = Intensity of solar radiation striking surface, Btu per hour per square foot (Tables 2, 3, 4 and 5).

The total amount of heat conducted through a wall exposed to the sun is the sum of H_t and H_R from Formulas 1 and 2.

[3]A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, by F H. Faust, L Levine, and F. O. Urban (A S H V E TRANSACTIONS, Vol 41, 1935, p 327)

TABLE 6. SOLAR ABSORPTION COEFFICIENTS FOR DIFFERENT BUILDING MATERIALS

SURFACE MATERIAL		ABSORPTION COEFFICIENT (a)
Very Light Colored Surfaces	White stone Very light colored cement White or light cream-colored paint	0.4
Medium Dark Surfaces	Asbestos shingles Unpainted wood Brown stone Brick and red tile Dark-colored cement Stucco Red, green or gray paint	0 7
Very Dark Colored Surfaces	Slate roofing Tar roofing materials Very dark paints	0 9

The calculation of heat transmission through walls and roofs does not take into consideration the heat capacity of the structure nor the consequent time lag in the transmission of heat. In the case of massive walls the time lag may amount to several hours[4]. Thus in many cases the wall transmission cannot be added directly to the cooling load from other sources because the peak of the wall transmission load may not coincide with the peak of the total cooling load and may even occur after the cooling system has been shut down for the day. The data in Table 7 were taken from A.S.H.V.E. research papers and whereas they result from a study of experimental slabs, they give an approximate idea of the time lag to be expected in various structures.

Solar Radiation Transmitted Through Glass

Windows present a problem somewhat different from that of opaque walls, because they permit a large percentage of the solar energy to pass through, a small amount is reflected and the balance is absorbed by the glass. The amount absorbed depends upon the character and thickness of the glass and the angle between the sun's rays and the glass. The temperature of the glass is raised by the absorbed heat and this heat is then delivered to the air on the two sides of the glass in proportion to the the difference between the glass and air temperature.[5]

TABLE 7 TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

TYPE AND THICKNESS OF WALL OR ROOF	TIME LAG, HOURS
2-in pine	1½
6-in concrete	3
4-in. gypsum	2½
3-in. concrete and 1-in. cork	2
2-in. iron and cork (equivalent to ¾-in concrete and 2.15-in. cork)	2½
4-in. iron and cork (equivalent to 5½-in concrete and 1 94-in. cork)	7¼
8-in. iron and cork (equivalent to 16-in concrete and 1.53-in. cork)	19
22-in. brick and tile wall	10

[4]Loc Cit. Notes 1 and 2.

[5]Heat Absorbing Glass Windows, by W W Shaver (A S H V E TRANSACTIONS, Vol 41, 1935, p 287).

The A.S.H.V.E. tests[6] indicate that a single pane of double strength glass 0.127 in. thick absorbs approximately 11 per cent of the solar radiation passing through it when the impingement is normal. For smaller angles of impingement, the glass retards percentages of the total radiant energy approximately in proportion to the sine of the angle. Other experiments[7] indicate a glass absorption of 16.7 per cent for one pane of glass and 37.5 per cent for two ¼ in. panes separated by a 1¾ in. air space.

The amount of solar radiation delivered to an unshaded glass surface may be obtained from Tables 2, 3, 4 or 5. These values must be used only for the net glass area on which the sun shines and not the entire glass area. Tests at the A.S.H.V.E. Research Laboratory[8] have determined the percentage of heat from solar radiation actually delivered to a room with various types of outdoor and indoor shading. The data in Table 8 are taken from these tests.

The percentage values in this table were obtained by dividing the total amount of heat actually entering through the shaded window by the total amount of heat calculated to enter through a bare window (solar

TABLE 8. SOLAR RADIATION TRANSMITTED THROUGH SHADED WINDOWS

TYPE OF APPURTENANCE	FINISH FACING SUN	PER CENT DELIVERED TO ROOM
Canvas awning	Plain	28
Canvas awning	Aluminum	22
Inside shade, fully drawn	Aluminum	45
Inside shade, one-half drawn	Buff	68
Inside Venetian blind, fully covering window, slats at 45 deg	Aluminum	58
Outside Venetian blind, fully covering window, slats at 45 deg	Aluminum	22

radiation plus glass transmission, based on observed outside glass temperature). For bare windows on which the sun shines, the transmission of heat from outside air to glass may be small or negative as the glass temperature is raised by the solar radiation absorbed On the other hand, at times when the solar intensity is low, the heat gain as a result of solar radiation may be less than that due to normal transmission.

In calculating the total heat gain through windows on the sunny side of buildings, it is sufficiently accurate to figure the total heat gain to a window as follows:

Consider the total heat gain as that resulting from solar radiation and neglect the heat transmission through the glass caused by the difference between the temperatures of the inside and outside air. This method should be used except at times when the calculated heat gain per square foot due to normal transmission exceeds the solar intensity. At such times, solar radiation may be neglected and the total heat gain considered as resulting from normal transmission.

[6]A.S H V E RESEARCH REPORT No 974—Radiation of Energy Through Glass, by J L Blackshaw and F. C Houghten (A S H V E TRANSACTIONS, Vol 40, 1934, p 93) A S H V E RESEARCH REPORT No 975 —Studies of Solar Radiation Through Bare and Shaded Windows by F C Houghten, Carl Gutberlet, and J. L Blackshaw (A S H V E TRANSACTIONS, Vol. 40, 1934, p 101)

[7]A S H V E RESEARCH REPORT No 924—Field Studies of Office Building Cooling, by J H Walker, S S Sanford, and E P Wells, (A S H V E TRANSACTIONS, Vol 38, 1932, p 285)

[8]Loc Cit Note 6

The solar heat transmission through windows or skylights may be expressed by the formula:

$$H_G = A_G f I$$

where

H_G = Solar radiation transmitted through a window, Btu per hour.

A_G = Net area of glass exposed to sun's rays, square feet.

f = Percentage of solar radiation (expressed as a decimal) transmitted to the inside (Table 8). For bare windows, f = 1.

I = Intensity of solar radiation striking surface, Btu per hour per square foot (Tables 2, 3, 4 and 5).

Although the tests from which Table 8 was obtained showed that approximately all of the heat from solar radiation is delivered to a room through bare window glass, other tests[9] have indicated that in the case of a building having floors of high heat capacity such as concrete floors on which the solar radiation falls, approximately one half of the heat entering a bare window is absorbed by the floor and does not immediately become a part of the cooling load, but is delivered back to the air in the building at a slow rate over a period of 24 hr or longer.

The maximum solar intensity on any surface is of limited duration as shown in Fig. 1. In the case of windows the total energy impinging on the glass before and after the time of maximum intensity is further reduced by increased shading of the glass from the frame, or wall. The cooling load due to solar radiation therefore does not have to be calculated as a steady load. Another point which should be noted is that the maximum solar radiation load on the east wall occurs early in the morning when the outside temperature is low.

In a paper[10] by the A.S.H.V.E. Research Laboratory it was shown that ordinary double strength window glass transmits no measureable amount of energy radiated from a source at 500 F or lower; that it transmits only 6.0 and 12.3 per cent of the total radiation from surfaces at 700 F and 1000 F, respectively; and that it transmits 65.7 per cent of the radiation from an arc lamp, 76.3 per cent of the radiation from an incandescent tungsten lamp, and 89.9 per cent of the radiation from the sun. Thus, glass windows in a room constitute heat traps, which allow rather free transmission of radiant energy into the room from the sun to warm objects in it, but do not allow the transmission of re-radiated heat from these same objects.

Tests have been made which indicated that sunshine through window glass is the most important factor to contend with in the cooling of an office building. At times it was shown to account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sunshine load, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. If buildings are provided with awnings so that the window glass is shielded from sunshine, the amount of cooling required will be reduced and there will also be less difference in the cooling

[9] A S H V E RESEARCH REPORT No 1002—Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutberlet, and Albert J Wahl (A S H.V E TRANSACTIONS, Vol 41, 1935, p 53)

[10] Loc Cit Note 6

requirements of different sides of the building. The total cooling load for a building exposed to the sun on more than one side is of course less than the sum of the maximum cooling loads in the individual rooms since the maximum solar radiation load on the different sides occurs at different times. In determining the total cooling load for a building if the time when the maximum load occurs is not obvious, the load should be calculated for various times of day to determine the times at which the sum of the loads on the different sides of the building is a maximum.

Heat Emission of Occupants

The heat and moisture given off by human beings under various states of activity are shown in Figs. 8 to 11 and Table 4 of Chapter 3. It will be observed that the rate of sensible and latent heat emission by human beings varies greatly depending upon state of activity. In many applications this component becomes a large percentage of total load.

Heat Introduced by Outside Air

An allowance must be made for the heat and moisture in the outside air introduced for ventilation purposes or entering the building through cracks, crevices, doors, and other places where infiltration might occur.

The volume of air entering due to infiltration may be estimated from data given in Chapter 6. Information on the amount of outside air required for ventilation will be found in Chapter 3.

In the event the volume of air entering an enclosure due to infiltration exceeds that required for ventilation, the former should be used as a basis for determining the portion of the load contributed by outside air. Where volume of air required for ventilation exceeds that due to infiltration it is assumed that a slight positive pressure will exist within the enclosure with a resulting exfiltration instead of infiltration. In this case the air required for ventilation is used in determining outside air load.

The sensible heat gain resulting from the outside air introduced may be determined by the following formula:

$$H_s = 0.24 \times 60\, d_o\, Q\, (t_o - t) \tag{4}$$

where

H_s = sensible heat to be removed from outside air entering the building, Btu per hour
Q = volume of outside air entering building, cubic feet per minute.
d_o = density of air, pounds of dry air per cubic foot at temperature t_o
t_o = temperature of outside air, degrees Fahrenheit.
t = temperature of inside air, degrees Fahrenheit.

The total heat gain resulting from outside air introduced may be determined by the following formula:

$$H = 60\, d_o\, Q\, (h_o - h) \tag{5}$$

where

H = total heat to be removed from outside air entering the enclosure, Btu per hour.
Q = volume of outside air entering enclosure, cubic feet per minute.
d_o = density of air, pounds of dry air per cubic foot of air (at temperature t_o).

h_o = heat content of mixture of outside dry air and water vapor, Btu per pound of dry air (at temperature t_o).
h = heat content of mixture of inside dry air and water vapor, Btu per pound of dry air (at temperature t).

The latent heat gain resulting from outside air introduced may be determined by the following formula:

$$H_l = H - H_s \tag{6}$$

where

H_l = latent heat to be removed, Btu per hour.
H = total heat to be removed, Btu per hour.
H_s = sensible heat to be removed, Btu per hour.

Heat Emission of Appliances

Heat generating appliances which give off either sensible heat or both sensible and latent heat in an air conditioned enclosure may be divided into three general classes of equipment or devices:

1. Electrical appliances.
2. Gas appliances.
3. Steam heating appliances.

In the first group may be found such devices as lights, motors, toasters, waffle irons, etc. The capacities of most electrical devices may be determined from the watt capacity indicated on their name plates. The Btu equivalent of heat generated per hour is determined by multiplying the watt capacity by 3.4 (one watthour is equivalent to 3.413 Btu).

The capacities of electric motors are usually expressed in terms of horsepower instead of watts. If the motor efficiency is known, the watts input may be calculated from the formula:

$$P = \frac{746\ (hp)}{n} \tag{7}$$

where

P = motor input, watts.
hp = motor load, horsepower
n = motor efficiency (expressed as a decimal).

When the motor efficiency is not known the heat equivalent of electrical input can be approximately determined by applying data given in Table 9.

TABLE 9 HEAT GENERATED BY MOTORS

NAMEPLATE RATING HORSEPOWER	HEAT GAIN IN BTU PER HOUR PER HORSEPOWER	
	Connected Load in Same Room	Connected Load Outside of Room
1/8 to 1/2	4250	1700
1/2 to 3	3700	1150
3 to 20	2950	400

In the second group belong such appliances as coffee urns, gas ranges, steam tables, broilers, hot plates, etc. For heat generating capacities of such appliances refer to Table 10.

TABLE 10. HEAT GAIN FROM VARIOUS SOURCES

Source	Btu Per Hour Sensible	Btu Per Hour Latent	Btu Per Hour Total
Electric Heating Equipment			
Electrical Equipment—Dry Heat—No Evaporated Water	100%	0%	100%
Electric Oven—Baking	80%	20%	100%
Electric Equipment—Heating Water—Stewing, Boiling, etc.	50%	50%	100%
Electric Lights and Appliances per Watt (Dry Heat)	3 4	0	3 4
Electric Lights and Appliances per Kilowatt (Dry Heat)	3413	0	3413
Electric Motors per Horsepower	2546	0	2546
Electric Toasters or Electric Griddles	90%	10%	100%
Coffee Urn—Large, 18 in. Diameter—Single Drum	2000	2000	4000
Coffee Urn—Small, 12 in Diameter—Single Drum	1200	1200	2400
Coffee Urn—Approx Connected Load per Gallon of Capacity	600	600	1200
Electric Range—Small Burner	*	*	3400
Electric Range—Large Burner	*	*	7500
Electric Range—Oven	8000	2000	10000
Electric Range—Warming Compartment	1025	0	1025
Steam Table—Per Square Foot of Top Surface	300	800	1100
Plate Warmer—Per Cubic Foot of Volume	850	0	850
Baker's Oven—Per Cubic Foot of Volume	3200	1300	4500
Frying Griddles—Per Square Foot of Top Surface	*	*	4600
Hot Plates—Per Square Foot of Top Surface	*	*	9000
Hair Dryer in Beauty Parlor—600 w.	2050	0	2050
Permanent Wave Machine in Beauty Parlor—24-25 w Units	2050	0	2050
Gas Burning Equipment			
Gas Equipment—Dry Heat—No Water Evaporated	90%	10%	100%
Gas Heated Oven—Baking	67%	33%	100%
Gas Equipment—Heating Water—Stewing, Boiling, etc	50%	50%	100%
Stove, Domestic Type—No Water Evaporated—Per Medium Size Burner	9000	1000	10000
Gas Heated Oven—Domestic Type	12000	6000	18000
Stove, Domestic Type—Heating Water—Per Medium Size Burner	5000	5000	10000
Residence Gas Range—Giant Burner (About 5½ in Diameter)	*	*	12000
Residence Gas Range—Medium Burner (About 4 in. Diameter)	*	*	10000
Residence Gas Range—Double Oven (Total Size 18 in x 18 in x 22 in High)	*	*	18000
Residence Gas Range—Pilot	*	*	250
Restaurant Range—4 Burners and Oven	*	*	100000
Cast-Iron Burner—Low Flame—Per Hole	*	*	100
Cast-Iron Burner—High Flame—Per Hole	*	*	250
Simmering Burner	*	*	2500
Coffee Urn—Large, 18 in Diameter—Single Drum	5000	5000	10000
Coffee Urn—Small, 12 in Diameter—Single Drum	3000	3000	6000
Coffee Urn—Per Gallon of Rated Capacity	500	500	1000
Egg Boiler—Per Egg Compartment	2500	2500	5000
Steam Table or Serving Table—Per Square Foot of Top Surface	400	900	1300
Dish Warmer—Per Square Foot of Shelf	540	60	600
Cigar Lighter—Continuous Flame Type	2250	250	2500
Curling Iron Heater	2250	250	2500
Bunsen Type Burner—Large—Natural Gas	*	*	5000
Bunsen Type Burner—Large—Artificial Gas	*	*	3000
Bunsen Type Burner—Small—Natural Gas	*	*	3000
Bunsen Type Burner—Small—Artificial Gas	*	*	1800
Welsbach Burner—Natural Gas	*	*	3000
Welsbach Burner—Artificial Gas	*	*	1800
Fish-tail Burner—Natural Gas	*	*	5000
Fish-tail Burner—Artificial Gas	*	*	3000
Lighting Fixture Outlet—Large, 3 Mantle 480 C P	4500	500	5000
Lighting Fixture Outlet—Small, 1 Mantle 160 C P	2250	250	2500
One Cubic Foot of Natural Gas Generates	900	100	1000
One Cubic Foot of Artificial Gas Generates	540	60	600
One Cubic Foot of Producer Gas Generates	135	15	150
Steam Heated Equipment			
Steam Heated Surface Not Polished—Per Square Foot of Surface	330	0	330
Steam Heated Surface Polished—Per Square Foot of Surface	130	0	130
Insulated Surface, Per Square Foot	80	0	80
Bare Pipes, Not Polished Per Square Foot of Surface	400	0	400
Bare Pipes, Polished Per Square Foot of Surface	220	0	220
Insulated Pipes, Per Square Foot	110	0	110
Coffee Urn—Large, 18 in Diameter—Single Drum	2000	2000	4000
Coffee Urn—Small, 12 in Diameter—Single Drum	1200	1200	2400
Egg Boiler—Per Egg Compartment	2500	2500	5000
Steam Table—Per Square Foot of Top Surface	300	800	1100
Miscellaneous			
Heat Liberated By Food per person, as in a Restaurant	30	30	60
Heat Liberated from Hot Water used direct and on towels per hour—Barber Shops	100	200	300

*Per cent sensible and latent heat depends upon use of equipment; dry heat, baking or boiling

Considerable judgment must be exercised in the use of data given in Table 10. Consideration must be given to time of day when appliances are used and the heat they contribute to the space at time of peak load. Only those appliances in use at the time of the peak load need be considered. Consideration must also be given to the way appliances are installed, whether products of combustion are vented to a flue, whether products of combustion escape into the space to be conditioned or whether appliances are hooded allowing part of the heat to escape through a stack connected with the hood. There are no generally accepted data available on the effects of venting and shielding heating appliances but it is believed that when the appliances are properly hooded with a positive fan exhaust system through the hood that 50 per cent of the heat will be conveyed up into the hood and the balance of 50 per cent will be dissipated in the space to be conditioned. Where latent as well as sensible heat is given off, it is usually safe to assume that all latent heat will be removed by a properly designed and operated vent or hood.

GENERAL

From the foregoing discussion it is obvious that the determination of the maximum cooling load is rather complicated by reason of the variable nature of contributing load components. If the time when the maximum load occurs is not obvious the load should be calculated for various times of the day to determine the probable time at which the sum of the various component loads is a maximum.

Application of the foregoing data in determining cooling load requirements is illustrated in Example 1.

Example 1. Determine cooling load requirements for a clothing store illustrated in Fig. 3 and located in Cleveland, Ohio, Latitude 40 deg. This is a one-story building located on a corner and it faces south and west. Assume building on east and north sides conditioned.

Wall construction, 8 in. hollow tile, 4 in. brick veneer, plaster on walls, U = 0 33 (Table 4, Wall 38 B, Chapter 5).

Roof construction, 2 in. concrete, ½ in. rigid insulation, metal lath and plaster ceiling, U = 0 26 (Table 11, Wall 2 J, Chapter 5)

Floor, maple flooring on yellow pine, no ceiling below, U = 0 34 (Table 8, Wall 1 D, Chapter 5).

Partition, wood lath and plaster on both sides of studding, U = 0.34 (Table 6, Wall 77 B, Chapter 5).

Show windows, provided with awnings and thin panel partition at rear.

Front doors, 2 ft 6 in. x 7 ft (glass paneled), U = 1.13 (Table 13 A, Chapter 5).

Side door, 3 ft x 7 ft (solid, 1¾ in. thick), U = 0 51 (Table 13 B, Chapter 5).

Occupancy, 10 clerks, 40 patrons.

Lights, 4200 w.

Outside design conditions, dry-bulb 95 F; wet-bulb 75 F.

Inside design conditions, dry-bulb 80 F; wet-bulb 67 F.

Basement temperature, 85 F.

Store room temperature, 88 F.

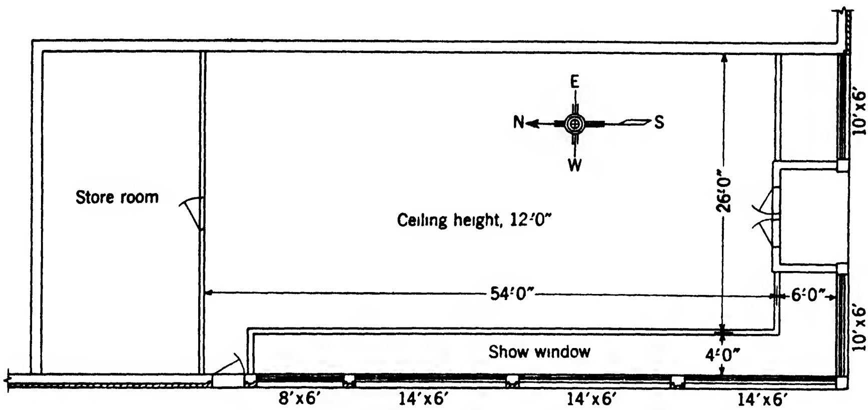

FIG. 3. PLAN DIAGRAM OF CLOTHING STORE

Solution: The normal heat transmission through various surfaces shown in load calculations are determined by application of Formula 1.

It is quite obvious from the shape and exposure of this store that the maximum sun load will exist on the west wall. Since the west wall has a large glass area with a negligible time lag, the peak load may be expected at 4:00 p.m. at which time, from Table 4, I = 211. I for the south side at 4:00 p m is 8. Because of the small amount of solar radiation transmitted through the south glass, it can be neglected and the total heat gain taken as that due to normal transmission. Assuming time lag in roof and walls to be 2 hours, the corresponding values for I for south and west walls and roof will be those shown in Table 4 for 2:00 p m. They are respectively 77, 143 and 258. A time lag of 1 hour was assumed for the west door amounting to I = 192. By substituting these values in equations 2 and 3 the solar heat load is determined

To determine the heat gain from the outside air it is necessary first to determine the volume of the outside air to be introduced. Since the show windows are sealed so as not to permit infiltration and since there are only three doors in this store through which infiltration can take place, it is obvious that infiltration of air will be a negligible quantity. The volume of the store is 21,600 cu ft Good practice indicates that in a store of this character there should be a minimum of from 1 to 1½ outside air changes per hour On a basis of 1½ air changes the volume of outside air to be introduced would be 32,400 cfh. By reference to Chapter 3 it will be noted that the minimum ventilation requirements are 10 cfm per person. On this basis the ventilation requirements would be 30,000 cfh. Since this will produce approximately 1½ outside air changes per hour, 30,000 cfh will be considered in this application.

To determine load imposed by occupants it will be found from Table 4, Chapter 3 that the average person standing at rest will dissipate 225 Btu sensible heat and 206 Btu latent heat per hour.

NORMAL TRANSMISSION LOAD:

SURFACE	DIMENSIONS	AREA SQ FT	U	TEMP DIFF DEG F	BTU PER HOUR
S Glass	2(2 ft 6 in x 7 ft)+ 2(10 ft x 6 ft)	155	1 13	15	2,627
S Wall	(30 ft x 12 ft) − 155	205	0 33	15	1,015
W Wall	(60 ft x 12 ft) − 321	399	0 33	15	1,975
W Door	3 ft x 7 ft	21	0 51	15	161
Roof	60 ft x 30 ft	1800	0 26	15	7,020
Floor	26 ft x 54 ft	1404	0 34	5	2,387
N Partition	30 ft x 12 ft	360	0 34	8	979
Total					16,164

SUN LOAD:

SURFACE	DIMENSIONS	AREA SQ FT	F	a	I	SHADE FACTOR	BTU PER HOUR
S Wall		205	0 078	0.7	77		862
W Glass	3(14 ft x 6 ft)+ (8 ft x 6 ft)	300			211	0 28	17,724
W Door		21	0 118	0 7	192		333
W Wall		399	0 078	0 7	143		3,113
Roof		1800	0 062	0 9	258		25,914
Total							47,946

OUTSIDE AIR HEAT GAIN:

Sensible heat, $H_s = 0.24 \times 60\, d_o\, Q\, (t_o - t)$ (Formula 4).

$Q = 50 \times 10 = 500$ cfm.

Density of air at 95 F dry-bulb and 75 F wet-bulb for a barometric pressure of 29.92 in. is 0 07105 lb per cubic foot (Table 4, Chapter 1).

Dew-point of outdoor air is 66 F (psychrometric chart).

Partial pressure of vapor is 0 64378 in. Hg. (Pressure of saturated vapor at 66 F, (Table 6, Chapter 1).

$W = 0.622 \left(\frac{e}{B - e}\right) = 0.622 \left(\frac{0\ 64378}{29\ 92 - 0.64378}\right) = 0.0137$ lb water vapor per pound dry air (Formula 5 a, Chapter 1).

$\frac{1}{1 \text{ plus } 0.0137} = 0.986$ lb dry air per pound outside air.

$d_o = 0.07105 \times 0.986 = 0.0699$ lb dry air per cubic foot outside air.

$H_s = 60 \times 500 \times 0.0699 \times 0.24\ (95\text{-}80) = 7549$ Btu per hour.

Total heat, $H = 60\, d_o\, Q\, (h_o - h)$ (Formula 5).

$h_o = 38.46$ Btu per pound dry air at 75 F wet-bulb (Table 6, Chapter 1).

$h = 31.51$ Btu per pound dry air at 67 F wet-bulb (Table 6, Chapter 1).

$H = 60 \times 0.0699 \times 500\ (38.46 - 31.51) = 14{,}574$ Btu per hour.

Latent heat gain from outside air $= 14{,}574 - 7549 = 7025$ Btu per hour.

PEOPLE HEAT GAIN:

$50 \times 225 = 11{,}250$ Btu per hour, sensible heat.

$50 \times 206 = 10{,}300$ Btu per hour, latent heat.

LIGHT HEAT GAIN:

$4200 \times 3.413 = 14{,}335$ Btu per hour.

SUMMARY:

COMPONENT OF LOAD	BTU PER HOUR	
	Sensible	Latent
Normal Transmission Load	16,164	
Sun Load	47,946	
Outside Air Heat Gain	7,549	7,025
People Heat Gain	11,250	10,300
Light Heat Gain	14,335	
Total	97,244	17,325

TOTAL LOAD:

$97{,}244 + 17{,}325 = 114{,}569$ Btu per hour.

PROBLEMS IN PRACTICE

1 ● a. What is the maximum heat transmission for a flat roof located in Pittsburgh (latitude 40 deg) exposed to the sun with the outdoor and indoor temperature 95 F and 80 F, respectively? The roof is of uninsulated 6 in. concrete with its underside exposed, and with a black upper surface. b. What time of day will maximum cooling load due to the roof exist?

a. $H_t + H_R = [A\,U\,(t_o - t)] + [A\,FaI]$ (Formulas 1 and 2).
U for roof $= 0.64$ (Table 11, Wall 4 A, Chapter 5).
$F = 0.147$ (Fig. 2).
$a = 0.9$ (Table 6).
$I = 293$ (Table 4)
$H_t + H_R = [1 \times 0.64\,(95\text{-}80)] + [1 \times 0.147 \times 0.9 \times 293] = 48.5$ Btu per square foot per hour.

b. Maximum sun intensity occurs at noon (Table 4). Maximum effect in cooling load will occur at 3 p.m. (Table 7).

2 ● a. What is the maximum rate of heat delivered to a room through a bare window in the west wall of a building located in New Orleans (30 deg latitude)? b. What time of day will it occur? c. What will maximum rate be if window is protected by awning?

a. $H_G = A_G fI$ (Formula 3).
$I = 205$ (Table 2).
$H_G = 1 \times 1 \times 205 = 205$ Btu per square foot per hour.

b. At 4 p.m (Table 2).

c. $205 \times 0.28 = 57.4$ Btu per square foot per hour (Table 8).

3 ● What is the heat gain per cubic foot of outside air introduced, under the following conditions if the barometric pressure is 29.5 in. Hg? Outdoor temperatures, 90 F dry-bulb and 75 F wet-bulb. Inside temperatures, 78 F dry-bulb and 65 F wet-bulb.

Density of air at 90 F dry-bulb, 75 F wet-bulb and 29.5 in. Hg is 0.0705 lb per cubic foot (Table 4, Chapter 1)

Dew-point of outdoor air is 68.2 F (psychrometric chart).

Pressure of saturated vapor at 68.2 F is 0.6946 in. Hg (Table 6, Chapter 1).

$W = 0.622\left(\frac{e}{B - e}\right) = 0.622\left(\frac{0.6946}{29.5 - 0.6946}\right) = 0.015$ lb water vapor per pound dry air (Formula 5a, Chapter 1).

$\frac{1}{1 \text{ plus } 0.015} = 0.985$ lb dry air per pound outside air.

$d_o = 0.0705 \times 0.985 = 0.06944$ lb dry air per cubic foot outside air

Heat content outside dry air at 75 F wet-bulb $= 38.46$ Btu per pound.

Heat content inside dry air at 65 F wet-bulb $= 29.96$ Btu per pound.

Total heat, $H = 0.06944\,(38.46 - 29.96) = 0.59$ Btu per cubic foot.

4 ● A 7 × 4 ft west window is equipped with an inside aluminum finished Venetian blind which is adjusted to fully cover the window when the sun shines. The net glass area is 75 per cent of the total area of the window. What is the cooling load due to the window at 10 a.m. and 4 p.m.? Temperatures are: 10 a.m., outside 85 F and inside 77 F; 4 p.m., outside 95 F and inside 80 F. Latitude 40 deg.

10 a.m.: $H_t = 28 \times 1.13\,(85\text{-}77) = 253$ Btu per hour (Equation 1, and Table 13A, Chapter 5).

4 p.m.: $28 \times 0.75 = 21$ sq ft net glass area.

$H_G = 21 \times 0.58 \times 211 = 2570$ Btu per hour (Tables 4 and 8).

Chapter 9

COMBUSTION AND FUELS

Principles of Combustion, Classification of Coals, Firing Methods for Coals, Firing Methods for Coke, Dustless Treatment of Coal, Classification of Oils, Combustion of Oil, Classification of Gas, Combustion of Gas

THE data given in the first part of this chapter are of general application to the various fuels used in domestic heating which are coal, coke, oil and gas. The choice of fuel is a question of dependability, cleanliness, fuel availability, economy, operating requirements and control.

FUNDAMENTAL PRINCIPLES OF COMBUSTION

Combustion may be defined as the chemical combination of a substance with oxygen with a resultant evolution of heat. The rate of combustion depends partly upon the specific rate of reaction of the combustible substance with oxygen and partly upon the rate at which oxygen is supplied and the surrounding conditions as they define the temperature.

Complete combustion is obtained when all of the combustible elements in the fuel are oxidized with all of the oxygen with which they can combine. All of the oxygen supplied may not be utilized.

Perfect combustion is defined as the result of supplying the required amount of oxygen for combination with all of the combustible elements of the fuel and utilizing all of the oxygen so supplied.

The oxygen required for the process of combustion is obtained from air which is a mechanical mixture of oxygen, nitrogen and small amounts of carbon dioxide, water vapor and inert gases. These inert gases are generally included with the nitrogen, and for engineering purposes the values given herewith may be used.

	By Volume Per Cent	By Weight Per Cent
Oxygen, O_2	20 9	23 15
Nitrogen, N_2	79 1	76 85

The combination of oxygen with the combustible elements and compounds of a fuel is in accordance with fixed laws. In the case of perfect combustion the reactions and resultant combinations are shown in Table 1.

The most important condition governing the process of combustion is temperature. It is necessary to bring a combustible substance to its

TABLE 1. GENERAL DATA OF COMBUSTIBLE ELEMENTS AND COMPOUNDS

Substance	Molecular Symbol	Chemical Reaction of Combustion	Ignition Temperature[a] Deg F	Calorific Value			Theoretical Oxygen and Air Requirements			
				Btu per pound		Btu per Cubic Foot	Lb per Lb		Cubic Ft per Cubic Ft	
				Higher	Lower	Higher	O_2	Air	O_2	Air
Carbon (to CO)	–	$2C + O_2 = 2CO$	–	4380	–	–	1.333	5.76	–	–
Carbon (to CO_2)	–	$2C + 2O_2 = 2CO_2$	–	14540	–	–	2.667	11.52	–	–
Sulphur	S_2	–	–	–	–	–	1 000	4 32	–	–
Sulphur (to SO_2)	–	$S + O_2 = SO_2$	–	4050	–	–	–	–	–	–
Sulphur (to SO_3)	–	$2S + 3O_2 = 2SO_3$	–	5940	–	–	–	–	–	–
Carbon monoxide	CO	$2CO + O_2 = 2CO_2$	1166-1319	4380	–	342	0 572	2 46	0 5	2 391
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	1202-1346	23850	21670	1073	4 000	17.28	2 0	9.564
Acetylene	C_2H_2	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	763-824	21460	21020	1590	3 077	13 29	2 5	11 955
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	986-1123	21450	20420	1675	3 429	14 81	3 0	14 346
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	986-1123	22230	20500	1883	3 733	16.13	3 5	16.737
Hydrogen	H_2	$2H_2 + O_2 = 2H_2O$	1063-1166	62000	52920	348	8.000	34 56	0 5	2.391
Hydrogen sulphide	H_2S	$2H_2S + 3O_2 = 2H_2O + 2SO_2$	599-608	–	–	–	1 412	6 10	1 5	7.173

[a]From International Critical Tables 1927

ignition temperature before it will unite in chemical combination with oxygen to produce combustion. The ignition temperatures for several of the combustible constituents of fuels are presented in Table 1.

HEAT OF COMBUSTION

As previously stated, the process of combustion results in the evolution of heat. The *heat of combustion*, or calorific value, of a fuel is the amount of heat generated by the complete combustion of a unit of the fuel and is constant for a given combination of combustible elements and compounds. The heat of combustion of the several fuel elements and compounds in their *pure* state is given in Table 1.

The reaction of the carbon in the fuel with oxygen may result in the formation of carbon monoxide or carbon dioxide. In burning to carbon monoxide, the carbon is not completely oxidized and, as shown by the data, the heat produced is considerably less than if it were completely oxidized. This fact is of greatest importance in considering the efficiency of combustion.

The calorific value of a fuel is determined by direct measurement of the heat evolved during combustion in a calorimeter. As the calorific value, on a *moisture and ash free* basis, of coal from a given district or mine remains substantially constant the calculation of the calorific value of a particular lot of coal can be made if the lot is analyzed for moisture and ash. From a known reliable calorific value for coal from the same mine or district the calorific value on a "moisture and ash free" basis, often called the H value, is calculated from formula (1).

$$\text{Calorific value, moisture and ash free} = \frac{100 \times \text{Calorific value (as received)}}{100 - (\text{Moisture} + \text{Ash})} \quad (1)$$

If a *dry* or *moisture free* analysis is used it is necessary to correct for *ash only* to reduce to moisture and ash free basis. From the value obtained by formula (1) the calorific value for the sample under consideration can be calculated as follows:

$$\text{Calorific value} = \frac{\text{Calorific value (moisture and ash free)} \times [100 - (\text{Moisture} + \text{Ash})]}{100} \quad (2)$$

In the above formulae moisture and ash are expressed in per cent.

The H values for Illinois coals are published[1] and it is to be expected that more data on H values for other coals will be available in the future.

As practically all fuels contain hydrogen they produce a certain amount of water vapor as one of the products of combustion. The amount of water vapor produced increases as the hydrogen content of the fuel increases. When the calorific value of a fuel is determined in a calorimeter the water vapor is condensed and the latent heat of vaporization that is given up during the condensation is reported as a portion of the heat value of the fuel. The heat value so determined is termed the *gross* or *higher* heat value and this is what is ordinarily meant when the heat value of a fuel is specified. In burning the fuel, however, the products of

[1]*State Geological Survey Bulletin*, No 62, Classification and Selection of Illinois Coals

combustion are not cooled to the dew-point and the higher calorific value cannot be obtained.

FLAME

The appearance of the flame or products of combustion may serve as an approximate measure of the temperatures developed in the combustion process. The luminosity of a flame is caused by the heating to incandescence of unconsumed particles of combustible matter in the gases and the higher the temperature of these particles the whiter the flame. Table 2 gives some approximate flame temperature data.

AIR AND COMBUSTION

The weight of air required for the perfect combustion of a pound of fuel may be determined by use of the ultimate analysis of the fuel as applied to formulae 3 to 5. The various elements are expressed in percentages by weight.

TABLE 2. FLAME TEMPERATURE DATA

Appearance of Flame	Temperature Deg F
Red, visible in daylight	975
Light red	1832
Orange-red	2012
Orange-yellow	2192
Yellow-white	2372
Bright white	2550

Solid and Liquid Fuels:

$$\text{Pounds air required per pound fuel} = 34.56 \frac{C}{3} + \left(H + \frac{O}{8}\right) + \frac{S}{8} \tag{3}$$

Gaseous Fuels.

$$\text{Pounds air required per pound fuel} = 2.46\,CO + 34.56\,H_2 + 17.28\,CH_4 + 13.29\,C_2H_2 + 14.81\,C_2H_4 + 16.13\,C_2H_6 + 6.10\,H_2S - 4.32\,O_2 \tag{4}$$

When the analysis is given on a volumetric basis the formula is expressed as follows:

$$\text{Cubic feet air required per cubic foot gas} = 2.39\,(CO + H_2) + 9.56\,CH_4 + 11.98\,C_2H_2 + 14.35\,C_2H_4 + 16.74\,C_2H_6 - 4.78\,O_2 \tag{5}$$

Formulae 6 and 7 may be used as approximate methods of determining the theoretical air requirement for any fuel.

$$\text{Pounds air required per pound fuel} = 0.755 \times \frac{\text{Calorific value (Btu per pound)}}{1000} \tag{6}$$

$$\text{Cubic feet air required per unit fuel} = \frac{\text{Calorific value (Btu per unit)}}{100} \tag{7}$$

Approximate values for the theoretical air required for different fuels are given in Table 3.

It is customary to make use of the analysis of the products of combustion to determine the amount of flue gas produced and the actual

amount of air supplied for combustion. The analysis of flue gases has been well described in various publications of the *Bureau of Mines* and in the literature and the details of Orsat manipulation need not be considered in this discussion. (See Chapter 45).

The weight of dry flue gas per pound of fuel burned is used in combustion loss calculations and may be determined by formula 8.

$$\text{Pounds dry flue gas per pound fuel} = \frac{11\,CO_2 + 8\,O_2 + 7\,(CO + N_2)}{3\,(CO_2 + CO)} \times C \qquad (8)$$

Values for CO_2, O_2, CO and N_2 are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

TABLE 3. THEORETICAL AIR REQUIREMENTS

SOLID FUEL	POUNDS AIR PER POUND FUEL
Anthracite	9 6
Semi-bituminous coal	11 2
Bituminous coal	10 3
Lignite	6 2
Coke	11 2

FUEL OIL	POUNDS AIR PER GALLON FUEL
Commercial Standard No 1	102 6
Commercial Standard No 2	104 5
Commercial Standard No 3	106 5
Commercial Standard No 5	112 0
Commercial Standard No 6	114 2

GASEOUS FUELS	CUBIC FEET AIR PER CUBIC FOOT GAS
Natural gas	10 0
Mixed, natural and water gas	4.4
Carbureted water gas	4 4
Water gas, coke	2 1
Coke oven gas	5 2

EXCESS AIR

Because the real measure of the efficiency of combustion is the relation existing between the amount of air theoretically required for *perfect* combustion and the amount of air actually supplied a method of determining the latter factor is of value. Formula 9 will give reasonably accurate results, for most solid and liquid fuels, for determining the amount of air supplied per pound of fuel.

$$\text{Pounds dry air supplied per pound of fuel} = \frac{3\,036\,N_2}{(CO_2 + CO)} \times C \qquad (9)$$

Values for CO_2, CO and N are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

The relationship of the air supplied, as determined from the previous formula, to the theoretical air required indicates the per cent of excess air supplied.

A formula that may be used to determine directly the per cent of *excess air* is expressed:

$$\text{Per cent excess air} = \frac{100\left(O_2 - \frac{CO}{2}\right)}{N_2 \times 0.264 - \left(O_2 - \frac{CO}{2}\right)} \qquad (10)$$

In this formula the symbols represent volumetric percentages of the flue gas constituents as determined by analysis.

The amount of excess air in its relation to the percentage of CO_2 is shown by the curves in Fig. 1 for several fuels. These are approximate values. It should be noted that in hand-fired furnaces with long periods between firings the combustion goes through a cycle in each period and the quantity of excess air present varies.

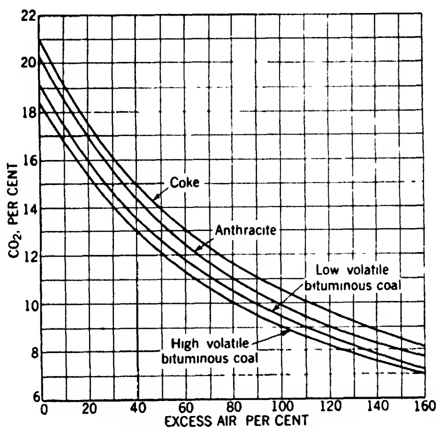

FIG. 1. RELATION BETWEEN CO_2 AND EXCESS AIR IN GASES OF COMBUSTION

Due to the different carbon-hydrogen ratios of the different fuels the maximum CO_2 attainable varies. Representative values for perfect combustion of several fuels are given in Table 4.

In considering the factor of excess air it should be noted that a deficiency of air supply will result in combustible products passing to the stack unburned. An excess of air absorbs heat from the products of combustion and results in a greater loss of sensible heat to the stack. An excess of air is usually required, however, to eliminate combustible losses occasioned by poor mixing of the fuel and air. It is considered good practice, under usual operating conditions, to supply from 25 to 50 per cent excess air, dependent upon the fuel utilized.

SECONDARY AIR

When a solid fuel is hand-fired in a furnace the volatile matter in the fuel distills off leaving coke on the grate. The product of combustion of

the coke is CO_2 and under certain conditions some CO may arise from the bed. The combustion of the volatile matter and the CO may amount to the liberation of from 40 to 60 per cent of the heat in the fuel in the combustion space over the fuel bed.

The air that passes through the fuel bed is called *primary air* and the air that is admitted over the fuel bed in order to burn the volatile matter and CO is called secondary air.

TABLE 4. MAXIMUM CO_2 VALUES

FUEL	PER CENT CO_2
Coke	21.0
Anthracite	20.2
Bituminous coal	18.2
Oil	15.5
Natural gas	11.5
Coke oven gas	9.25

This process of combustion is illustrated in Fig. 2[2]. The free oxygen of the air passes through the grate and the ash above it and burns the carbon in the lower three or four inches of the fuel bed forming carbon dioxide. This layer noted as the oxidizing zone is indicated by the symbols CO_2 and O_2. Some of the carbon dioxide of the oxidizing zone is reduced to carbon monoxide in the upper layer of the fuel bed noted as the reducing zone and indicated by the symbols CO_2 and CO. The gases leaving the fuel

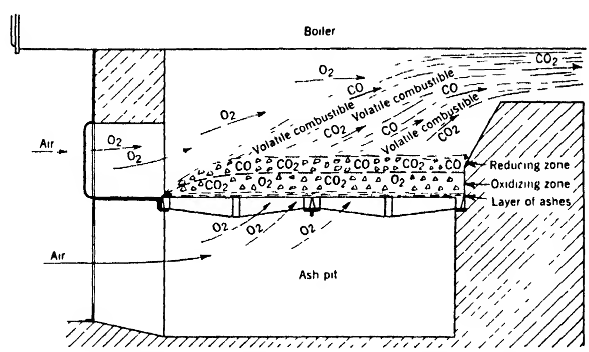

FIG 2 COMBUSTION OF FUEL IN A HAND-FIRED FURNACE

bed are mainly carbon monoxide, carbon dioxide, nitrogen and very little free oxygen. Free oxygen is admitted through the firing door to burn carbon monoxide and the volatile combustible distilled from the freshly fired fuel.

The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size of fuel, depth of fuel bed, and size of firepot. The ratio of the secondary to the primary air increases with

[2]From *Bureau of Mines Technical Paper* No 80.

decrease in the size of the fuel pieces, with increase in the depth of the fuel bed, and with increase in the area of the firepot; the ratio also increases with increase in rate of burning.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through the fuel bed and thus automatically causes a larger draft above the fuel bed, which draws in more secondary air through the same slot openings. In spite of this, a small size fuel requires a larger opening of the door slots; for a certain size for each fuel no slot opening is required, and for larger sizes too much excess air gets through the fuel bed

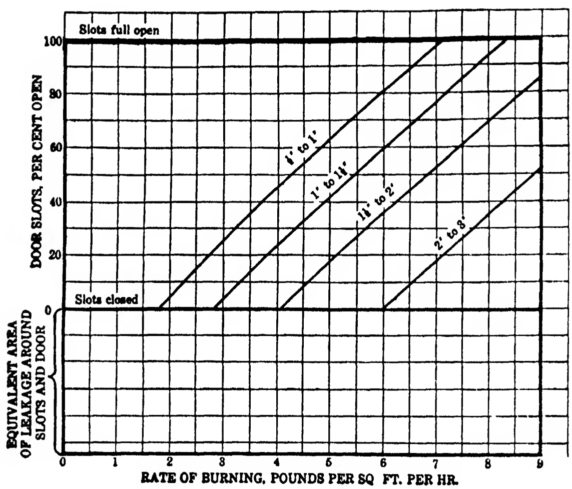

FIG. 3 RELATIVE AMOUNT OF FIRE DOOR SLOT OPENING REQUIRED IN A GIVEN FURNACE TO GIVE EQUALLY GOOD COMBUSTION FOR HIGH TEMPERATURE COKE OF VARIOUS SIZES WHEN BURNED AT VARIOUS RATES

It is impossible to establish a single rule for the correct slot opening for all types and sizes of fuels and for all rates of burning. Furthermore, the effect of slot opening is dependent on whether the ashpit damper is open or closed. It is better to have too much than too little secondary air; the opening is too small if there is a puff of flame when the firing door is opened

The relationship of the slot opening, for a domestic furnace, to the size of coke and the rate of burning is shown in Fig. 3[3]. These openings are with the ashpit damper wide open, and would be less if the available draft permits of its being partly closed. The same openings are satisfactory for anthracite.

Bituminous coals require a large amount of secondary air during the period subsequent to a firing in order to consume the gases and to reduce the smoke. The smoke produced is a good indicator, and that opening is

[3]From *Bureau of Mines Report of Investigations*, No 2980.

best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial. The following suggestions will be helpful:

1 In cold weather, with high combustion rates, the secondary air damper should be half open all the time.

2 In very mild weather, with a very low combustion rate, the secondary air damper should be closed all the time

3 For temperatures between very mild and very cold, the secondary air damper should be in an intermediate position.

4 For ordinary house operation, secondary air is needed after each firing for about one hour

In the field of domestic heating the use of secondary air in the combustion of oil is generally restricted to the larger semi-commercial types of oil burners used in large heating boilers. This factor is discussed in Chapter 11, Automatic Fuel Burning Equipment.

The air that is supplied around the flame in a domestic heating gas burner is considered as secondary air As it is drawn into the appliance by natural draft action, the need for proper draft control is evident.

Draft Requirements

The draft required to effect a given rate of burning the fuel as measured at the smokehood is dependent on the following factors:

1 Kind and size of fuel
2 Combustion rate per square foot of grate area per hour
3 Thickness of fuel bed
4 Type and amount of ash and clinker accumulation.
5 Amount of excess air present in the gases
6 Resistance offered by the boiler passes to the flow of the gases
7 Accumulation of soot in the passes

Insufficient draft will necessitate additional manipulation of the fuel bed and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control by adjustment of the dampers.

The quantity of excess air present has a marked affect on the draft required to produce a given rate of burning. If the excess is caused by holes in the fuel bed or an extremely thin fuel bed it is often possible to produce a higher rate of burning by increasing the thickness of the bed. The thickness of the fuel bed should not, however, be increased too much because the increased draft resistance will reduce the rate of primary air supply and the rate of burning.

DRAFT REGULATION

Because of the varying heating load demands present in most installations it is necessary to vary the rate of fuel burning. The maintenance of the proper air supply for the various rates of burning is accomplished by regulation of the drafts. Correct and incorrect methods of draft regulation are shown in Fig. 4. The air enters through the ashpit, firing door and by leaks in the setting, whereas the gases leave only through the up-take. By throttling the gases with the damper in the up-take all the

air entering by each of the three intakes is reduced in the same proportion. If the ashpit door is closed the air admitted through the ashpit is reduced and increased through the other two intake openings.

Methods of control of draft conditions when burning oil or gas are noted in Chapter 11, Automatic Fuel Burning Equipment.

CLASSIFICATION OF COALS

The complex composition of coal makes it difficult to classify it into clear-cut types. Its chemical composition is some indication but coals having the same chemical analysis may have distinctly different burning characteristics. Users are mainly interested in the available heat per

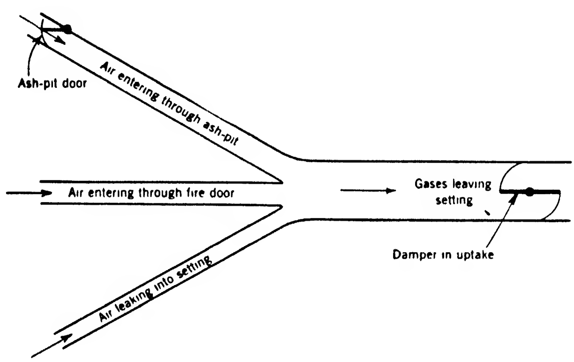

FIG. 4 CORRECT AND INCORRECT METHODS OF DRAFT REGULATION IN A HAND-FIRED FURNACE

pound of coal, in the handling and storing properties, and in the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space; a treatment applicable to heating boilers is given in *Bureau of Mines Bulletin* 276

The classification of coals by rank involves several of the items indicated in a proximate analysis of the coal. This analysis determines the content of volatile matter, fixed carbon, ash and moisture. The calorific value and sulphur content are often reported with the proximate analysis. Volatile matter is the loss of weight when the coal sample is heated to 1732 F for 7 min. Fixed carbon is the difference from 100 per cent of the sum of other losses, not including sulphur. Ash is the incombustible impurity in the coal and has no heating value. Moisture is the inherent and extraneous water in the fuel.

A classification of coals is given in Table 5, and a brief description of the kinds of fuels is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other.

Anthracite is a clean, dense, hard coal which creates very little dust in handling It is comparatively hard to ignite but it burns freely when well started It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires little attention to the fuel bed between firings. It is capable of giving a high efficiency in the common

types of hand-fired furnaces. A tabulation of the quality of the various anthracite sizes will be found in *Bureau of Mines Report of Investigations* No. 3283.

Semi-anthracite has a higher volatile content than anthracite, it is not as hard and ignites somewhat more easily; otherwise its properties are similar to those of anthracite.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak.

TABLE 5. CLASSIFICATION OF COALS BY RANK[a]

Legend · F C = Fixed Carbon. V.M. = Volatile Matter. Btu = British thermal units

CLASS	GROUP	LIMITS OF FIXED CARBON OR BTU MINERAL-MATTER-FREE BASIS	REQUISITE PHYSICAL PROPERTIES
I Anthracite	1 Meta-anthracite	Dry F C, 98 per cent or more (Dry V M, 2 per cent or less)	Non-agglomerating[b]
	2 Anthracite	Dry F C, 92 per cent or more and less than 98 per cent (Dry V M, 8 per cent or less and more than 2 per cent)	
	3 Semi-anthracite	Dry F C, 86 per cent or more and less than 92 per cent (Dry V M, 14 per cent or less and more than 8 per cent)	
II Bituminous[d]	1 Low volatile bituminous coal	Dry F C, 78 per cent or more and less than 86 per cent (Dry V M, 22 per cent or less and more than 14 per cent)	Either agglomerating[b] or non-weathering[f]
	2 Medium volatile bituminous coal	Dry F C, 69 per cent or more and less than 78 per cent (Dry V M, 31 per cent or less and more than 22 per cent)	
	3 High volatile *A* bituminous coal	Dry F C, less than 69 per cent (Dry V M, more than 31 per cent), and moist[c] Btu, 14,000[e] or more	
	4 High volatile *B* bituminous coal	Moist[c] Btu, 13,000 or more and less than 14,000[e]	
	5 High volatile *C* bituminous coal	Moist Btu, 11,000 or more and less than 13,000[e]	
III Sub-bituminous	1 Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[e]	Both weathering and non-agglomerating[b]
	2 Sub-bituminous *B* coal	Moist Btu 9500 or more and less than 11,000[e]	
	3 Sub-bituminous *C* coal	Moist Btu 8300 or more and less than 9500[e]	
IV Lignitic	1 Lignite	Moist Btu less than 8300	Consolidated
	2 Brown coal	Moist Btu less than 8300	Unconsolidated

[a]This classification does not include a few coals which have unusual physical and chemical properties and which come within the limits of fixed carbon or Btu of the high-volatile bituminous and sub-bituminous ranks All of these coals either contain less than 48 per cent dry, mineral-matter-free fixed carbon or have more than 15,500 moist, mineral-matter-free Btu

[b]If agglomerating classify in low-volatile group of the bituminous class

[c]Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal

[d]It is recognized that there may be non-caking varieties in each group of the bituminous class

[e]Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu

[f]There are three varieties of coal in the High-volatile *C* bituminous coal group, namely, Variety 1, agglomerating and non-weathering, Variety 2, agglomerating and weathering, Variety 3, non-agglomerating and non-weathering

Adapted from *A S T M* Standards, 1937, Supplement, p 145, *American Society for Testing Materials*, Philadelphia

Having only half the volatile matter content of the more abundant bituminous coals it can be burned with less production of smoke, and it is sometimes called *smokeless coal.*

The term *bituminous coal* covers a large range of coals and includes many types having distinctly different composition, properties, and burning characteristics The coals range from the high-grade bituminous coals of the East to the poorer coals of the West Their caking properties range from coals which completely melt, to those from which the volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to

permit of the screened sizes being delivered free from fines. In general, they ignite easily and burn freely; the length of flame varies with different coals, but it is long. Much smoke and soot are possible especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning, the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value, it is clean to handle It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition Freshly mined lignite, because of its high moisture, ignites slowly It is non-caking The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ashpit continue to burn Very little smoke or soot is formed

It is often desirable to learn about the properties of a coal, such as the various items noted in the discussion of proximate analyses As a guide for the consumer as to the expected characteristics of coals several commercial publications are available and numerous reports of the *Bureau of Mines* discuss the coals produced in individual state areas

CLASSIFICATION OF COKES

Coke is produced by the distillation of the volatile matter from coal The type of coke depends on the coal or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven, coke is also produced as a residue from the destructive distillation of oil.

High-temperature cokes Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke*, which covers the greater part of the coke sold, and *gas-house coke* The differences among these three cokes are relatively small, their denseness and hardness decrease and friability increases in the order named In general, the lighter and more friable cokes ignite and burn the more easily

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter In general, these cokes ignite and burn more readily than high-temperature cokes The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted

Petroleum cokes, which are obtained by coking the residue left from the distillation of petroleum, vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

FIRING METHODS FOR ANTHRACITE[4]

An anthracite fire should never be poked, as this serves to bring ash to the surface of the fuel bed where it melts into clinker.

Egg size is suitable for large firepots (grates 24 in. and over) if the fuel can be fired at least 16 in. deep. The air spaces between the pieces of coal are large, and for best results this coal should be fired deeply.

Stove size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 12 in. deep. The only instructions needed for burning this type of fuel are that the grate should be shaken daily, the fire should

[4]See reports published by *Anthracite Industries Laboratory*, Primos, Delaware County, Pennsylvania

never be poked or disturbed, and the fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for firepots up to 20 in in diameter, with a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. It is relatively low in price. When fired carefully, pea coal can be burned on standard grates. It is well to have a small amount of a larger fuel on hand when building new fires, or when filling holes in the fuel bed. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire-door. This keeps a bed of ignited coal in readiness against the time when a sudden demand for heat shall be made on the heater. A very satisfactory method of firing pea coal consists of drawing the red coals toward the front end and piling fresh fuel toward the back of the fire-box.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open, the cold-

TABLE 6 ANTHRACITE STANDARDS

Classification	Coal Size Inches	
Egg	Through 3-1/4	Over 2-7/16
Stove	Through 2-7/16	Over 1-5/8
Nut	Through 1-5/8	Over 13/16
Pea	Through 13/16	Over 9/16
Buckwheat	Through 9/16	Over 5/16

air check closed, and by controlling the fire with the air-inlet damper only. Pea size can also be fired in layers with stove or egg size anthracite and its use in this manner will reduce the fuel costs and attention required.

Buckwheat size coal for best results requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates Greater care must be taken in shaking the grates than with pea coal on account of the danger of the fuel falling through the grate. In house heating furnaces the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire it is advisable after coaling to expose a small spot of hot fire by putting a poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent an explosion of gas within the firepot, which in some cases depending upon the thickness of the bed of fresh coal is severe enough to blow open the doors and dampers of the furnace. A good draft is required and consequently the fire is best controlled by the air-inlet damper only. Where frequent attention can be given and care exercised in manipulation of the grates this fuel can be burned satisfactorily without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and

then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an easily broken up condition so that it readily can be shaken through the grate.

Forced draft and small mesh grates are frequently used for burning buckwheat anthracite. For best results and a higher degree of convenience, domestic stokers are used.

No. 2 buckwheat anthracite, or rice size, is used only in domestic stokers. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

The *Anthracite Institute* Standards of sizing are shown in Table 6 taken from *Anthracite Industries Manual,* Report No. 2403.

FIRING METHODS FOR BITUMINOUS COAL

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

Air should be admitted over the fire through a special secondary air device, or through a slide in the fire-door or by opening the fire-door slightly. If the quantity of air admitted is too great the gases will be cooled below the ignition temperature and will fail to burn. The fireman can judge the quantity of air to admit by noting when the air supplied is just sufficient to make the gases burn rapidly and smokelessly above the fuel bed.

The red fuel in the firebox, before firing, excepting only a shallow layer of coke on the grate, should be pushed to one side or forward or backward to form a hollow in which to throw the fresh fuel. Some manufacturers recommend that all red fuel be pushed to the rear of the firebox and that the fresh fuel be fired directly on the grate and allowed to ignite from the top. The object of this is to reduce the early rapid distillation of gases and to reduce the quantity of secondary air required for smokeless combustion.

It is well to have the bright fuel in the firebox so placed that the gases from the freshly fired fuel, mixed with the air over the fuel bed, pass over the bed of bright fuel on the way to the flues. The bed of bright fuel then supplies the heat to raise the mixture of air and gas to the ignition temperature, thereby causing the gaseous matter to burn and preventing the formation of smoke.

The importance of firing bituminous coal in small quantities at short intervals is discussed in the *U. S. Bureau of Mines Technical Paper,* No. 80. Better combustion is obtained by this method in that the fuel supply is maintained more nearly proportional to the air supply.

This is demonstrated in Fig. 5 where diagram A shows the air supply and the distillation of the volatile combustible when the firings are 5 min apart; and diagram B indicates the same relationships when the firings are 15 min apart. In both cases the amount of coal fired per hour and the weight of volatile combustible distilled from the coal are the same. This weight of volatile conbustible is represented by the shaded area under the saw-tooth curve. The horizontal dotted lines represent the constant air

supply sufficient to burn the volatile matter represented by the shaded areas under each line. The shaded areas above each horizontal line represent for each air supply the loss from incomplete combustion of the volatile matter. The clear area under each horizontal line represents the loss from excessive air. As the air supply increases the loss from incomplete combustion decreases but the loss from excessive air becomes larger. The sum of the two losses is the least when the air supply is introduced as noted by the average line. It is evident that the sum of the losses for the average air supply is much larger in diagram B than in A which would indicate that small and frequent firings are better than large firings at long intervals.

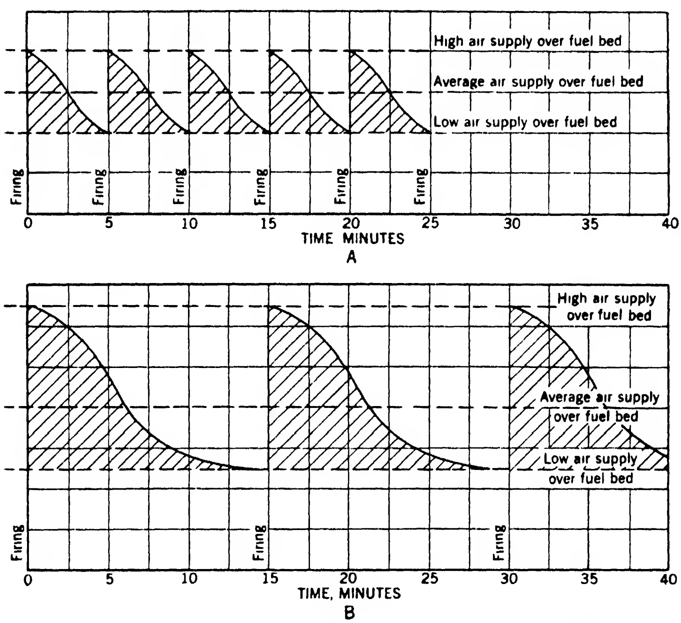

FIG. 5. RELATION OF RATE OF DISTILLATION OF VOLATILE MATTER AND NECESSARY AIR SUPPLY

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the firebox. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

It is acknowledged that it may be difficult to apply the outlined methods to domestic heating boilers of small size, especially when frequent attendance is impractical. The adherence to these methods insofar as practical, however, will result in better combustion.

The output obtained from any heater with bituminous coal will usually exceed that obtainable with anthracite, since bituminous coal burns more rapidly than anthracite and with less draft. Bituminous coal, however, will require frequent attention to the fuel bed, because it burns unevenly, even though the fuel bed may be level, forming holes in the fire which admit too much air, chilling the gases over the fuel bed and reducing the available draft.

FIRING METHODS FOR SEMI-BITUMINOUS COAL

The *Pocahontas Operators Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

FIRING METHODS FOR COKE

Coke ignites less readily than bituminous coal and more readily than anthracite and burns rapidly with little draft In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly A coke fire responds rapidly to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly In order to obtain the same interval of attention as with other fuels a deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should be shaken only until the first red particles drop from the grates in cold weather. The best size of coke for general use, for small firepots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in screen For large firepots where the fuel can be fired over 20 in deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory Large sizes of coke should be either mixed with fine sizes or broken up before using.

PULVERIZED COAL

Although several pulverized coal burning units for domestic heating plant firing have been developed, none has attained extended use Two general methods of adaptation have been employed, one where the coal is pulverized by the unit at the furnace and one where the coal is delivered to the home in pulverized form.

FURNACE VOLUME

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and correctly proportioned combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly. Anthracite requires practically no combustion space.

DUSTLESS TREATMENT OF COAL

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with petroleum products, particularly the lighter oils, a solution of calcium chloride or a mixture of calcium and magnesium chlorides. The latter salts are very hygroscopic and their moisture under normal atmospheric conditions keeps the surface of the coal damp, thus reducing the dust during delivery and in the cellar, and obviating the necessity of sprinkling the coal in the bin.

The coal is usually treated at the mine, but sometimes by the local distributor just before delivery. The salt solutions are sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size. Oil for the dustless treatment of coal is also applied under high pressure, in concentrations of 1 to 8 qt per ton of coal, depending upon the characteristics of the coal and oil.

CLASSIFICATION OF OILS

The Commercial Standard Specifications for Fuel Oils (CS 12-38) of the *U. S. Department of Commerce* are given in Table 7. These specifications conform with *American Society for Testing Materials* Tentative Specifications for Fuel Oils D396-38T.

The specific gravity of oil is of interest in its relationship to the calorific value and these data are given in Table 8.

COMBUSTION OF OIL

With oil, as with any kind of fuel, efficient heat production requires that all combustible matter in the fuel shall be completely consumed and that it shall be done with a minimum of excess air. The combustion of oil is a rather rapid chemical reaction. Excess air provides an over supply of oxygen so that all of the oil, composed of carbon and hydrogen, will be completely oxidized and thus produce all the heat possible. The use of unreasonable quantities of air in excess of theoretical combustion requirements results in lowered efficiencies due to increased stack losses. Such losses, if not accompanied by unburned products of combustion (saturated and unsaturated hydrocarbons, hydrogen, etc.) may be offset somewhat by increasing the secondary heating surfaces of the heat absorbing medium boiler or furnace.

Oil is a highly concentrated fuel composed mainly of hydrogen and carbon. In its liquid form oil cannot burn. It must be converted into a

TABLE 7 DETAILED REQUIREMENTS FOR FUEL OILS[a]

Grade[b]	Flash Point Deg F		Pour Point Deg F	Water and Sediment Per Cent	Carbon Residue Per Cent	Ash Per Cent	Distillation Temperatures Deg F: 10 Per Cent Point	Distillation Temperatures Deg F: 90 Per Cent Point		Distillation Temperatures Deg F: End Point	Viscosity Seconds: Saybolt Universal at 100 F		Viscosity Seconds: Saybolt Furol at 122 F	
	Min	Max	Max	Max	Max	Max	Max	Max	Min	Max	Max.	Min	Max	Min
No. 1 Fuel oil—a distillate oil for use in burners requiring a volatile fuel.	100 or Legal	165	15[c]	Trace	0 05 on 10% Residuum[d]		410			560[e]				
No. 2 Fuel oil—a distillate oil for use in burners requiring a moderately volatile fuel	110 or Legal	190	15[c]	0 05	0.25 on 10% Residuum[f]		440	600						
No. 3 Fuel oil—a distillate oil for use in burners requiring a low viscosity fuel.	110 or Legal	230	20[c]	0 10[h]	0 15 Straight			675	600[g]		55			
No. 5 Fuel oil—an oil for use in burners requiring a medium viscosity fuel.	130 or Legal			1 00		0 10						60	40	
No. 6 Fuel oil—an oil for use in burners equipped with preheaters permitting a high viscosity fuel	150			2 00[h]									300	45

[a]Recognizing the necessity for low sulphur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulphur requirement may be specified in accordance with the following table

Grade of Fuel Oil	Sulphur Max. Percent
No 1	0 5
No 2	0 5
No 3	0 75
No 5	No Limit
No 6	No Limit

Other sulphur limits may be specified only by mutual agreement between the buyer and seller

[b]It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade

[c]Lower or higher pour points may be specified whenever required by conditions of storage or use However, these specifications shall not require a pour point lower than 0 F under any conditions

[d]For use in other than sleeve type blue flame burners carbon residue on 10 per cent residuum may be increased to a maximum of 0 12 per cent This limit may be specified by mutual agreement between the buyer and seller

[e]The maximum end point may be increased to 590 F when used in burners other than sleeve type blue flame burners

[f]To meet certain burner requirements the carbon residue limit may be reduced to 0 15 per cent on 10 per cent residuum

[g]The minimum distillation temperature of 600 F for 90 per cent may be waived if A P I gravity is 26 or lower

[h]Water by distillation plus sediment by extraction Sum, maximum 2 0 per cent The maximum sediment by extraction shall not exceed 0 50 per cent A deduction in quantity shall be made for all water and sediment in excess of 1 0 per cent

TABLE 8. APPROXIMATE GRAVITY AND CALORIFIC VALUE OF STANDARD GRADES OF FUEL OIL

COMMERCIAL STANDARD NO	APPROXIMATE GRAVITY, RANGE BAUME	CALORIFIC VALUE BTU PER GALLON
1	38-40	136,000
2	34-36	138,500
3	28-32	141,000
5	18-22	148,500
6	14-16	152,000

gas or vapor by some means. If the excess air is to be kept within efficient limits it means that air must be supplied in carefully regulated quantities. The air and oil vapor must be vigorously mixed to get a rapid and complete chemical reaction. The better the mixing, the less excess air that will be needed. The combustion must take place in a space that maintains the temperatures high so the reaction will not be stopped before completion. When equipped with a means of igniting the oil and safety devices to guard against mishaps, the oil burner possesses all of the elements to be efficient and automatic.

CLASSIFICATION OF GAS

Gas is broadly classified as being either *natural* or *manufactured.* Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually a combination of certain proportions of gases produced by two or more processes, and is often designated as *city gas.* Representative properties of gaseous fuels commonly used in domestic heating are presented in Table 9.

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat values varies from 700 to 1500 Btu per cubic foot, the majority of natural gases averaging about 1000 Btu per cubic foot. Table 9 shows typical values for the four main oil fields, although values from any one field vary materially.

Table 9 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district.

COMBUSTION OF GAS

The majority of gas burners utilized in central domestic heating plants are of the Bunsen type and operate with a non-luminous flame. In this type of burner part of the air required for combustion is mixed with the gas as primary air, the air and gas mixture being fed to the burner ports.

Additional secondary air is introduced around the flame by draft inspiration. In the luminous flame burner, which is sometimes used, all of the air for combustion is brought in contact with the flame as secondary air. The importance of bringing the secondary air into intimate contact with the gas is noted.

Some makes of burners use radiants or refractories to convert some of the energy in the gas to radiant heat by utilizing the principle of surface combustion. The radiants also serve as baffles in directing the flow of the products of combustion.

Table 9. Representative Properties of Gaseous Fuels Based on Gas at 60 F and 30 in Hg.

Gas	Btu per Cu Ft		Specific Gravity Air = 1 00	Air Required for Combustion, (Cu Ft)	Products of Combustion				Theoretical Flame Temperature (deg Fahr)
					Cubic Feet			Ultimate CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—California	1200	1087	0.67	11.26	1.24	2 24	12 4	12 2	3610
Natural gas—Mid-Continental	967	873	0 57	9 17	0 97	1 92	10 2	11 7	3580
Natural gas—Ohio	1130	1025	0 65	10 70	1.17	2 16	11 8	12 1	3600
Natural gas—Pennsylvania	1232	1120	0 71	11.70	1.30	2 29	12 9	12 3	3620
Retort coal gas	575	510	0.42	5.00	0.50	1 21	5 7	11 2	3665
Coke oven gas	588	521	0 42	5.19	0.51	1 25	5 9	11.0	3660
Carbureted water gas	536	496	0.65	4.37	0.74	0 75	5 0	17 2	3815
Blue water gas	308	281	0.53	2 26	0 46	0 51	2 8	22 3	3800
Anthracite producer gas	134	124	0 85	1.05	0 33	0 19	1 9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0 19	2 0	19.0	3160
Oil gas	575	510	0 35	4.91	0 47	1.21	5 6	10 7	3725

Since one of the main functions of a gas burner is to properly proportion the air and gas, any marked change in the gas composition which affects the specific gravity necessitates a readjustment of the burner. It is necessary to supply a greater amount of air when the specific gravity of a gas is increased.

The quantity of air given in Table 9 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. The division of the air into primary and secondary is a matter of burner design and the pressure of gas available, and also of the type of flame desired.

The air gas ratio has a decided effect upon flame propagation. It is necessary that the gas will flow out of the burner ports fast enough so that the flame cannot travel back into the burner head, i.e. *flash back*, but the velocity must not be so high that it blows the flame away from the port.

The maximum and minimum flow speeds from burner ports which may be permitted are known to be very close together when air-gas mixtures in theoretical proportions are being supplied to the burner. As the air-gas ratio is lowered, and the mixture becomes more *gas rich*, the limiting speeds become further apart, until with 100 per cent gas, in an all-yellow flame, flash back cannot occur and a much higher velocity is needed to blow off the flames.

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PROBLEMS IN PRACTICE

1 ● What effect does moisture in fuels have on their efficiency?

With any solid fuel, latent and sensible heat are lost at the stack when moisture is dried out of the fuel in burning, and when its hydrogen is burned. Therefore, such fuels as sub-bituminous coal and lignite, which are high in moisture content, have a low efficiency. However, these efficiencies may be improved if the stack gases are cooled to room temperature, by heating the feed water, for example.

2 ● Does the size of a fuel affect the quantity of air required to burn it at a given rate?

The total air required to give the same gas analysis at the stack is independent of the size of the fuel burned, but for non-caking fuels the ratio of the air passing through the fuel bed to the total air entering the burner base decreases, for the same thickness of bed, as the size of the fuel becomes smaller; this decrease is very rapid for sizes less than one inch. For coals that cake, this ratio will depend on the way the caked bed is broken up and on the size of the resulting pieces.

3 ● Name several important properties of coal from a utilization standpoint.

a. Caking tendency, whether none, weak, or strong, *b.* quantity of volatile matter, *c.* friability, and *d.* fusibility of the ash.

4 ● What are the main data commonly available that fix the qualities of coal, and do these tell the whole story?

a. Calorific value, Btu per pound, *b.* proximate analysis giving percentages of moisture, volatile matter, fixed carbon, ash, and sulphur, *c.* temperature at which the ash softens, and *d.* screen sizes.

Other important qualities not usually given are the friability of the coal, its caking tendency, and the qualities of the volatile matter. The percentage of ash and its fusion temperature do not tell how the ash is distributed or how much of it is less fusible lumps of slate or shale.

5 ● Differentiate between the general characteristics of hard and soft coals.

Hard coals contain fixed carbon in large proportions and in addition more ash is present especially in the smaller sizes. Soft coals have an increasing percentage of carbon in combination with hydrogen which is volatile and will distill off under high temperature, producing smoke.

6 ● Is the volatile matter which is given off when coals are burned of the same nature in all coals?

No. The products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 per cent of ash-free and moisture-free coal, increasing amounts of oils and tars are given up. For coals of higher volatile content, the relative quantity of oils and tars decreases, so it is low in the sub-bituminous coals and in lignite.

7 ● Is smoke a primary product in the burning of fuels?

Visible smoke may include very small particles of carbon, oil, tar, water (condensed steam), and ash. Of these, the oils, tars, and ash are mainly primary products, and the water is partly primary. The carbon, which usually comprises the greater part of the smoke, results from the breaking up by heat of oils, tars, and such gases as methane, so it may be considered a secondary product.

8 ● Is the sulphur in coals detrimental to combustion?

Not so far as is known, but its complete combustion gives only 25 per cent as much heat as is given by the same weight of carbon. Sulphur is undesirable because it causes corrosion of flues and stacks, and also because its gases pollute the atmosphere, and damage buildings and vegetation.

9 ● How do deposits of soot on the surfaces of a boiler or heater affect the quantity of fuel burned?

There are two effects. The soot acts as an insulating layer over the surface and reduces the heat transmission to the water or air, the *Bureau of Mines Report of Investigations* No. 3272 shows that the loss of seasonal efficiency is not as great as has been believed and should not be over 6 per cent because the greater part of the heat is transmitted through the firepot. The soot clogs the passages and reduces the draft, the loss of efficiency from this action may be much more, and also the lack of draft results in unsatisfactory heating.

Chapter 10

CHIMNEYS AND DRAFT CALCULATIONS

Natural Draft, Mechanical Draft, Characteristics of Natural Draft Chimneys, Determining Chimney Sizes, General Equation, Chimney Construction, Chimneys for Gas Heating

THE design and construction of a chimney is so important a part of the heating engineer's work that a general knowledge of draft characterictics and calculations is essential.

Draft, in general, may be defined as the pressure difference between the atmospheric pressure and that at any part of an installation through which the gases flow. Since a pressure difference implies a head, draft is a static force. While no element of motion is inferred, yet motion in the form of circulation of gases throughout an entire boiler plant installation is the direct result of draft. This motion is due to the pressure difference, or unbalanced pressure, which compels the gases to flow. Draft is often classified into two kinds according to whether it is created thermally or artificially, *viz*, (1) natural or thermal draft, and (2) artificial or mechanical draft.

Natural Draft

Natural draft is the difference in pressure produced by the difference in weight between the relatively hot gases inside a natural draft chimney and an equivalent column of the cooler outside air, or atmosphere. Natural draft, in other words, is an unbalanced pressure produced thermally by a natural draft chimney as the pressure transformer and a temperature difference. The intensity of natural draft depends, for the most part, upon the height of the chimney above the grate bar level and also the temperature difference between the chimney gases and the atmosphere.

A typical natural draft system consists essentially of a relatively tall chimney built of steel, brick, or reinforced concrete, operating with the relatively hot gases which have passed through the boilers and accessories and from which all the heat has not been extracted. Hot gases are an essential element in the operation of a natural draft system, although inherently a heat balance loss.

A natural draft chimney performs the two-fold service of assisting in the creation of draft by aspiration and also of discharging the gases at an elevation sufficient to prevent them from becoming a nuisance.

Natural draft is quite advantageous in installations where the total loss of draft due to resistances is relatively low and also in plants which have practically a constant load and whose boilers are seldom operated above

their normal rating. Natural draft systems have been, and are still being, employed in the operation of large plants during the periods when the boilers are operated only up to their normal rating. When the rate of operation is increased above the normal rating, some form of mechanical draft is employed as an auxiliary to overcome the increased resistances or draft losses. Natural draft systems are used almost exclusively in the smaller size plants where the amount of gases generated is relatively small and it would be expensive to install and operate a mechanical draft system.

The principal advantages of natural draft systems may be summarized as follows: (1) simplicity, (2) reliability, (3) freedom from mechanical

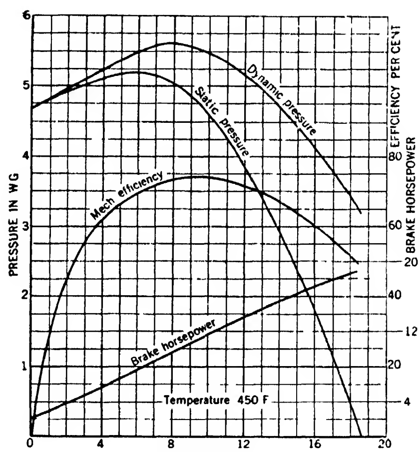

FIG. 1. GENERAL OPERATING CHARACTERISTICS OF TYPICAL INDUCED DRAFT FAN

parts, (4) low cost of maintenance, (5) relatively long life, (6) relatively low depreciation, and (7) no power required to operate. The principal disadvantages are: (1) lack of flexibility, (2) irregularity, (3) affected by surroundings, and (4) affected by temperature changes.

Mechanical Draft

Artificial draft, or mechanical draft, as it is more commonly called, is a difference in pressure produced either directly or indirectly by a forced draft fan, an induced draft fan, or a Venturi chimney as the pressure transformer. The intensity of mechanical draft is dependent for the most part upon the size of the fan and the speed at which it is operated. The element of temperature does not enter into the creation of mechanical draft and therefore its intensity, unlike natural draft, is independent of the temperature of the gases and the atmosphere. Mechanical draft includes the induced and Venturi types of draft systems in which the pressure difference is the result of a suction, and also the forced draft system in which the pressure difference is the result of a blowing. Mechanical draft systems tend to produce a vacuum or a plenum, as the system used in its production creates a pressure difference below, or above, atmospheric

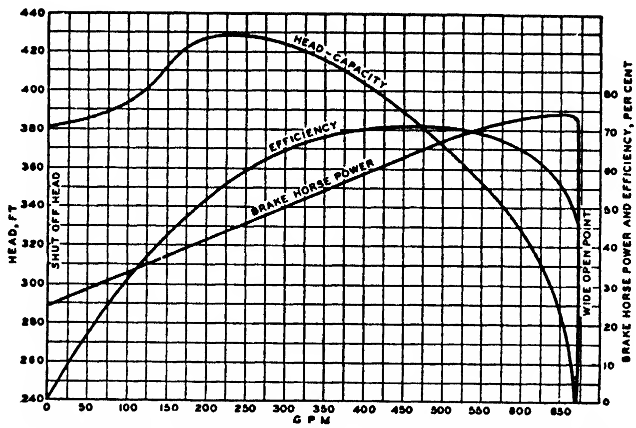

FIG. 2. OPERATING CHARACTERISTICS OF TYPICAL CENTRIFUGAL PUMP

pressure, respectively. A mechanical draft system may be used either in conjunction with, or as an adjunct to, a natural draft system.

Draft Control

To obtain the maximum efficiency of combustion, a definite minimum supply of air to the combustion chamber must be maintained. To provide this condition, it is necessary to have some mechanical means of draft control or adjustment, because of variable wind velocities, fluctuations in atmospheric temperatures and barometric pressures, and their effect upon draft.

For this purpose there are various mechanical devices which automatically control the volume of air admitted to the combustion chamber. Mechanical draft regulators designed to control or adjust draft, should

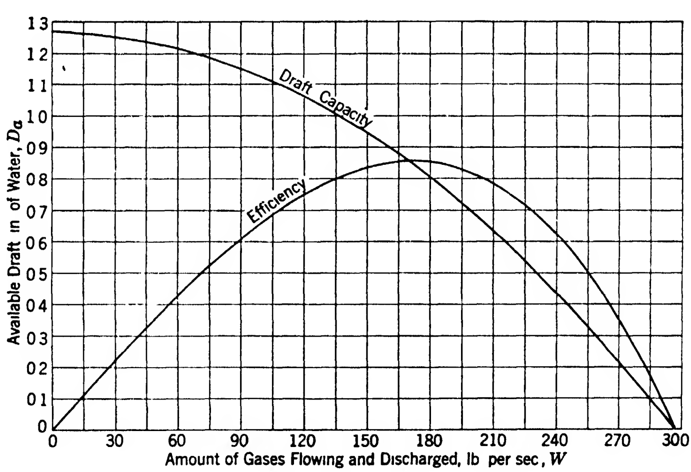

FIG. 3. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

not be confused with mechanical draft systems that *create* draft mechanically, but which must also be automatically controlled.

The use of such a device to provide a more uniform and dependable control of draft than could be maintained by manually operated dampers, will produce better combustion of fuel. This higher efficiency of combustion together with the reduced heat losses up the chimney by reason of decreased gas velocity, results in fuel economy, with consequent lower costs of plant operation.

CHARACTERISTICS OF CHIMNEYS

In order to analyze the performance of a natural draft chimney, it may be advantageous to compare its general operating characteristics with those of a centrifugal pump and also of a centrifugally-induced draft fan, there being a similarity among the three. Figs. 1, 2 and 3 show the general operating characteristics of a typical centrifugally-induced draft fan, a typical centrifugal pump, and a typical natural draft chimney, respectively. The draft-capacity curve of the chimney corresponds to the head-capacity curve of the pump and also to the dynamic-head-capacity curve of the fan.

When the gases in the chimney are stationary, the draft created is termed the *theoretical draft.* When the gases are flowing, the theoretical intensity is diminished by the draft loss due to friction, the difference between the two being termed the total *available draft.* The general equation for this net total available draft intensity of a natural draft chimney with a circular section is as follows:

$$D_a = 2.96HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126W^2T_cfL}{D^5B_oW_c} \tag{1}$$

where

D_a = available draft, inches of water.
H = height of chimney above grate bars, feet.
B_o = barometric pressure corresponding to altitude, inches of mercury
W_o = unit weight of a cubic foot of air at 0 F and sea level atmospheric pressure, pounds per cubic foot.
W_c = unit weight of a cubic foot of chimney gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
T_o = absolute temperature of atmosphere, degrees Fahrenheit.
T_c = absolute temperature of chimney gases, degrees Fahrenheit.
W = amount of gases generated in the combustion chamber of the boiler and passing through the chimney, pounds per second.
f = coefficient of friction.
L = length of friction duct of the chimney, feet.
D = minimum diameter of chimney, feet.

The first term of the right hand expression of Equation 1 represents the theoretical draft intensity, and the second term, the loss due to friction.

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, B_o = 29.92 in. of mercury; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Substituting these values in Equation 1 and reducing:

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^3 \times 960 \times 0.016 \times 200}{10^6 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 3 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1. When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available intensity decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss due to friction is equal to the theoretical intensity. The draft-capacity curve corresponds to the head-capacity curve of centrifugal

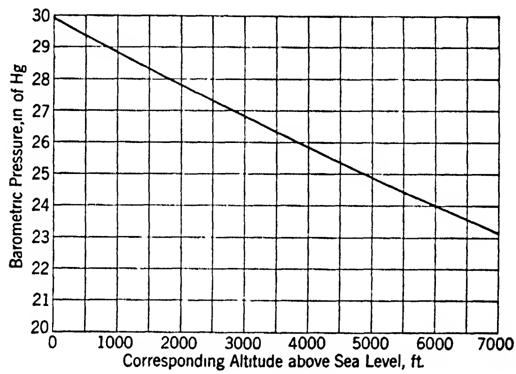

FIG 4. RELATION BETWEEN BAROMETRIC PRESSURE AND ALTITUDE

pump characteristics and the dynamic-head-capacity curve of a fan. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 1 and then plotting the results in the manner shown in Fig. 3.

In substituting the values for the various factors in Equation 1, care should be exercised that the selections be as near the actual conditions as is practically possible. The following notes will serve as a guide for these selections:

1. The *barometric pressure* varies inversely as the altitude of the plant above sea level. Fig. 4 gives the barometric pressure corresponding to various elevations as computed from the equation:

$$E_1 = 62{,}737 \log_{10} \frac{29.92}{B_o} \qquad (2)$$

where

E_1 = altitude of plant above sea level, feet.

In general, the barometric pressure decreases approximately 0.1 in. of mercury per 100 ft increase in elevation.

2. The ***unit weight of a cubic foot*** of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_c = 0.131 CO_2 + 0.095\ O_2 + 0.083\ N_2 \tag{3}$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_c may be assumed at 0.09.

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltrations in the chimney proper are here disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases

3. The ***atmospheric temperature*** is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The ***chimney gas temperature*** does not vary appreciably from the gas temperature as it leaves the breeching and enters the chimney. For average operating conditions, the chimney gas temperature will vary between 500 F and 650 F except in the case when economizers and recuperators are used, when the temperature will vary between 300 F and 450 F. If a chimney has been properly constructed, properly lined and has no air infiltration due to open joints, the temperature of the gases throughout the chimney will not differ appreciably from the foregoing figures. In most up-to-date heating plants, the temperature may be read from instruments or ascertained from a pyrometer. The analysis of this section is predicated on the assumption of constant gas temperature and no air infiltration throughout the height of the chimney.

5. The ***coefficient of friction*** between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the materials of construction becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and in general constant for all conditions of operation For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

However, much to be recommended as an alternate method is the practise of separately determining duct friction factors as a function of the flow conditions, specifically as a function of the Reynolds number and the relative duct roughness. The Reynolds criterion is based on the physical properties of the gas, the duct dimensions, and the gas velocity. The gas velocity for a chimney is usually well above the critical velocity. It is likely that this procedure of using a separately determined variable friction factor for chimney flow will give results that are to be preferred over those based on a set constant.

The Reynolds number, a dimensionless ratio, may be stated as follows:

$$C_r = \frac{DV\rho}{\mu} \tag{4}$$

where

D = chimney diameter, feet.
V = velocity of hot gas, feet per second
ρ = mass density of the chimney gas per cubic foot.
μ = viscosity of the gas in pounds-second per square foot taken at the gas temperature

In another form:

$$C_r = \frac{1.27\ W}{D\mu g} = \frac{0.0396\ W}{D\mu} \tag{5}$$

where

W = weight of gas passed per second
g = acceleration of gravity.

The value of μ for chimney gases is usually taken as that of air or nitrogen, and for the variation of μ with temperature, the Sutherland equation may be employed as follows, giving μ in pounds-second per square foot.

$$\mu = \mu_0 \left[\frac{273 + C}{T_c + C}\right] \left[\frac{T_c}{273}\right]^{1\,5}$$

where

T_c = chimney gas temperature, degrees Centigrade
μ_0 = gas viscosity at *O C*
C = constant for specific gas.

Using International Critical Table values, for air $\mu_0 = 35\,6 \times 10^{-8}$, $C = 124$, for nitrogen $\mu_0 = 34\,5 \times 10^{-8}$, and $C = 110$.

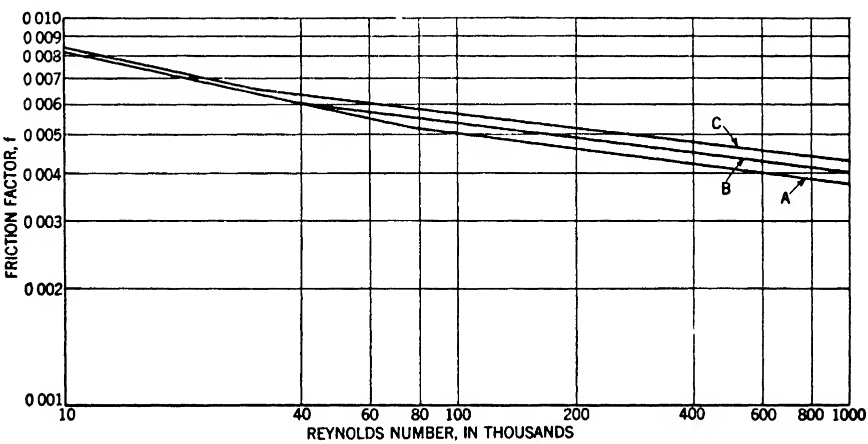

FIG 5. VARIATION OF FRICTION FACTOR f WITH REYNOLDS NUMBER

Values for the viscosity of air and of nitrogen (the principal component of chimney gases) for the different temperatures follow, in which the values given in pounds-second per square foot are to be multiplied by 10^{-8}

Temp. F	300	400	500	600	700	800
Air	49.7	54.5	58.5	62 5	66 7	70 5
Nitrogen	47.7	52.2	56 0	59.8	63.5	67.0

Example 2. To determine the Reynolds number C_r for a flow of 118 lb gas per second up a 12 ft diameter chimney at a temperature of 500 F The gas may be assumed to have the same viscosity as nitrogen at 500 F Using Equation 5.

$$C_r = 0\,0396 \frac{W}{D\mu} = \frac{0.0396 \times 118}{12 \times 56\,0 \times 10^{-8}} = 698{,}000$$

The variation of the friction factor f with the Reynolds number is shown in Fig. 5[1]. Three curves are shown: A, B, and C, where the choice of the friction factor curve depends on the relative surface roughness, and this for usual chimney construction may

[1]See also Flow of Fluids in Closed Circuits, by R J S. Pigott (*Mechanical Engineering*, August, 1933)

be selected by size since surface conditions in service are always undeterminant. For sizes up to 3 ft in diameter, Curve C may be used; from 3 to 6 ft, Curve B; and from 6 ft upwards, Curve A. Thus for the previous example with $C_r = 698{,}000$ and 12 ft diameter, f would be taken from Curve A as 0.0039.

6. The *length of the friction duct* is the vertical distance between the bottom of the breeching opening and the top of the chimney. Ordinarily this distance is approximately equal to the height of the chimney above the grate level.

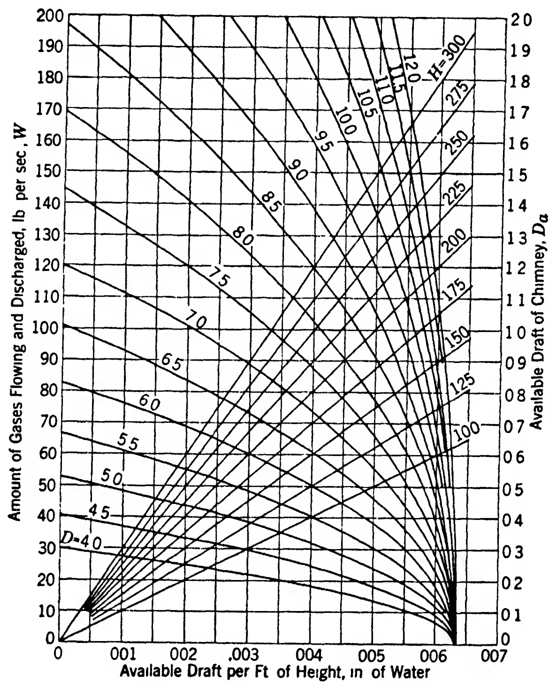

FIG. 6 CHIMNEY PERFORMANCE CHART[a]

[a]*To solve a typical example:* Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line; from this intersection follow horizontally to the right to Available Draft scale Starting from a point of Available Draft, take steps in reverse order

7. Assuming no air infiltration the *amount of gases flowing and being discharged* is, of course, equal to the amount of gases generated in the combustion chamber of the boiler. The total products of combustion in pounds per second for a grate-fired boiler may be computed from the equation:

$$W = \frac{C_g G W_{tp}}{3600} \tag{6}$$

where

C_g = pounds of fuel burned per square foot of grate surface per hour.

G = total grate surface of boilers, square feet.

$C_g \times G$ = total weight of fuel burned per hour.

W_{tp} = total weight of products of combustion per pound of fuel.

A similar computation may be made in the case of gas, oil, or stoker-fired fuel

Fig. 6 is a typical chimney performance chart giving the available draft intensities for various amounts of gases flowing and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific operating conditions, a new chart should be constructed from Equation 1.

It has been the usual custom, and still is to a lamentably great extent, to select the required size of a natural draft chimney from a table of chimney sizes based only on boiler horsepowers. After the ultimate horsepower of the projected plant had been determined, the chimney size in the table corresponding to this figure was then selected as the proper size required. Generally, no further attempt was made to determine if the height thus selected was sufficient to help create the required draft demanded by the entire installation, or the diameter sufficiently large to enable the chimney quickly, efficiently, and economically to dispose of the gases. Since the operating characteristics of a natural draft chimney are similar in all respects to those of a centrifugal pump, or a centrifugal fan, it is no more possible to select a proper size chimney from such a table, even with correction factors appended, than it is to select the proper size pump from tables based only on the amount of water to be delivered.

DETERMINING CHIMNEY SIZES

The required diameter and height of a natural draft chimney are given by the following equations:

$$H = \frac{D_r}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \tag{7}$$

$$D = 0.288\sqrt{\frac{WT_c}{B_oW_cV}} \tag{8}$$

where

H = required height of chimney above grate bar level, feet.

D = required minimum diameter of chimney, feet (constant for entire height).

V = chimney gas velocity, feet per second.

D_r = total required draft demanded by the entire installation outside of the chimney, inches of water

Equations 7 and 8 give the required size of a natural draft chimney with all of the operating factors taken into consideration. Values for all of the factors with the exception of the chimney gas velocity may be either observed or computed. It is, of course, necessary to assume an arbitrary value for the velocity in order to arrive at some definite size. For any one set of operating conditions there will be as many sizes of chimneys as there are values of reasonable velocities to assume. Of the number of sizes corresponding to the various assumed velocities, there is one size which will be least expensive. Since the cost of a chimney structure, regardless

of the kind of material used in the construction, varies as the volume of material in the structure, the cost criterion then may be represented by the approximate equation:

$$Q = \pi tHD \tag{9}$$

where

Q = volume of material, cubic feet
t = average wall thickness feet

For all practical purposes, the value of πt may be taken as a constant regardless of the size of the structure. Hence, in general, the volume, and consequently the cost, of a chimney structure may be based on the factor HD as a criterion. Therefore, the value of the chimney gas velocity which will result in the least value of HD for any one set of operating conditions will produce a structure which will be the most economical to use, because its cost will be least.

The problem at hand is to deduce an equation for the chimney gas velocity which will result in a combination of a height and a diameter whose product HD will be least. The solution is obtained by equating the product of Equations 7 and 8 to HD, differentiating this product with respect to V and equating the resulting expression to zero. This procedure results in the following expression:

$$V_e = \left(\frac{0.772 T_c \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \sqrt{\frac{W T_c}{B_o W_c}}}{f W_c} \right)^{2/5} \tag{10}$$

where V_e = economical chimney gas velocity, feet per second.

Equation 10 gives the economical velocity of the chimney gases for any set of operating conditions, and represents the velocity which will result in a chimney the size of which will cost less than that of any other size as determined by any other velocity for the same operating conditions. After the value of the economical velocity has been determined, the corresponding height and diameter can then be determined from Equations 7 and 8, respectively, and the economical size will then be attained. Equations 7, 8 and 10 may be simplified considerably for average operating conditions in an average size steam plant by assuming typical conditions.

Average chimney gas temperature, 500 F	T_c = 960
Mean atmospheric temperature, 62 F	T_o = 522
Average coefficient of friction, 0 016	f = 0 016
Average chimney gas density, 0 09	W_c = 0 09
Sea level elevation, with barometer of 29 92	B_o = 29 92

Substituting these values in Equations 10, 8 and 7, respectively, and reducing, the results are substantially:

$$V_e = 13\ 7W^{1/5} \tag{11}$$

$$D = 1\ 5W^{\cdot 2/5} \tag{12}$$

$$H = 190D_r \tag{13}$$

Fig. 7 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities as computed from Equations 11, 12 and 13. They are based on the operating factors used in reducing Equations 7, 8 and 10 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or for installations where the required data necessary for an exact deter-

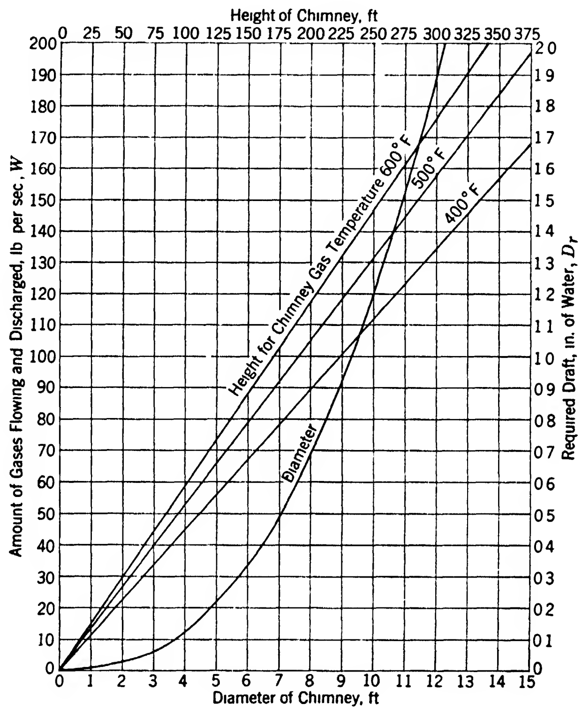

FIG. 7. ECONOMICAL CHIMNEY SIZES[a]

[a]Diameter values also for gas temperatures of 400, 500 and 600 F

mination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 7, 8 and 10. The recommended minimum inside dimensions and heights of chimneys for small and medium size installations are given in Table 1.

GENERAL EQUATION

The general draft equation for a steam producing plant may be stated as follows:

$$D_t - h_f = h_F + h_B + h_{Bd} + h_C + h_{Br} + h_V + h_O + h_E + h_R \qquad (14)$$

where

D_t = theoretical draft intensity created by pressure transformer, inches of water
h_f = draft loss due to friction in pressure transformer, inches of water.
h_F = draft loss through the fuel bed, inches of water.
h_B = draft loss through the boiler and setting, inches of water.
h_{Br} = draft loss through the breeching, inches of water.
h_V = draft loss due to velocity, inches of water.
h_{Bd} = draft loss due to bends, inches of water.
h_C = draft loss due to contraction of opening, inches of water.
h_O = draft loss due to enlargement of opening, inches of water.
h_E = draft loss through the economizer, inches of water.
h_R = draft loss through recuperators, regenerators, or air heaters, inches of water

The left hand member of Equation 14 represents the total amount of available draft created by the pressure transformer, that is, the natural draft chimney, Venturi chimney, or fan, and is equal to the theoretical intensity less the internal losses incidental to operation. The right hand member represents the sum of all of the various losses of draft throughout the entire boiler plant installation outside of the pressure transformer itself. The left hand member expresses the available intensity and is analogous to the head developed by a centrifugal pump in a water works system, while the right hand member expresses the required draft intensity and is analogous to the total dynamic head in a water works system. For a general circulation of gases

$$D_a = D_r \tag{15}$$

where

D_a = available draft intensity, inches of water
D_r = required draft, inches of water.

The *draft loss through the fuel bed* (h_F), or the amount of draft required to effect a given or required rate of combustion, varies between wide limits and represents the greater portion of the required draft. In coal-fired installations, the draft loss through the fuel bed is dependent upon the following factors: (1) character and condition of the fuel, clean or dirty; (2) percentage of ash in the fuel; (3) volume of interstices in the fuel bed, coarseness of fuel; (4) thickness of the fuel bed, rate of combustion; (5) type of grate or stoker used; (6) efficiency of combustion.

There is a certain intensity of draft with which the best results will be obtained for every kind of coal and rate of combustion. Fig. 8 gives the intensity of draft, or the vacuum in the combustion chamber required to burn various kinds of coal at various rates of combustion. Expressed in other words, these curves represent the amount of draft required to force the necessary amount of air through the fuel bed in order to effect various rates of combustion. It will be noted that the amount of draft increases as the percentage of volatile matter diminishes, being comparatively low for the lower grades of bituminous coals and highest for the high grades and small sizes of anthracites. Also, when the interstices of the coal are large and the particles are not well broken up, as with bituminous coals, much less draft is required than when the particles are small and are well

broken up, as with bituminous slack and the small sizes of anthracites. In general, the draft loss through the fuel bed increases as: (1) the percentage of volatile matter diminishes; (2) the percentage of fixed carbon increases; (3) the thickness of the bed increases; (4) the percentage of ash increases; (5) the volume of the interstices diminishes.

In making the preliminary assumptions for the draft loss through the fuel bed, due allowances should be made for a possible future change in the grade of fuel to be burned and also in the rate of combustion. A value

TABLE 1 RECOMMENDED MINIMUM CHIMNEY SIZES FOR HEATING BOILERS AND FURNACES[a]

Warm Air Furnace Capacity in Sq In of Leader Pipe	Steam Boiler Capacity Sq Ft of Radiation	Hot Water Heater Capacity Sq Ft of Radiation	Nominal Dimensions of Fire Clay Lining in Inches	Rectangular Flue		Round Flue		Height in Ft Above Grate
				Actual Inside Dimensions of Fire Clay Lining in Inches	Actual Area Sq In	Inside Diameter of Lining in Inches	Actual Area Sq In	
790	590	973	8½ x 13	7 x 11½	81			35
1000	690	1,140				10	79	
	900	1,490	13 x 13	11¼ x 11¼	127			
	900	1,490	8½ x 18	6¾ x 16¼	110			
	1,100	1,820				12	113	40
	1,700	2,800	13 x 18	11¼ x 16¼	183			
	1,940	3,200				15	177	
	2,130	3,520	18 x 18	15¾ x 15¾	248			
	2,480	4,090	20 x 20	17¼ x 17¼	298			45
	3,150	5,200				18	254	50
	4,300	7,100				20	314	
	4,600	7,590	20 x 24	17 x 21	357			
	5,000	8,250	24 x 24	21 x 21	441			55
	5,570	9,190		24 x 24[b]	576			60
	5,580	9,200				22	380	
	6,980	11,500				24	452	65
	7,270	12,000		24 x 28[b]	672			
	8,700	14,400		28 x 28[b]	784			
	9,380	15,500				27	573	
	10,150	16,750		30 x 30[b]	900			
	10,470	17,250		28 x 32[b]	896			

[a]This table is taken from the A S.H.V E Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929)

[b]Dimensions are for unlined rectangular flues

should be selected for this loss which will represent not only the highest rate of combustion which will be encountered, but also the grade of coal which has the greatest resistance through the fuel bed and which may be burned at a later date.

In powdered-fuel and oil-fired installations, there will be no draft loss through the fuel bed since there is none and, consequently, this factor becomes zero in the general draft equation. All other factors being constant, the height of the chimney in installations of this character will be less than the height in coal-fired installations, and in the case of mechanical draft installations the driving units need not be as large since the head against which the fan is to operate is not as great in the former as in the latter.

The *draft loss through the boiler and setting* (h_B) also varies between wide limits and, in general, depends upon the following factors:

1. Type of boiler.
2. Size of boiler.
3. Rate of operation.
4. Arrangement of tubes.
5. Arrangement of baffles.
6. Type of grate.
7. Design of brickwork setting.
8. Excess air admitted.
9. Location of entrance into breeching.

Curves showing the draft loss through the boiler are usually based on the load or quantity of gases passing through the boiler, expressed in terms of percentage of normal rate of operation. Owing to the great variety of boilers of different designs and the various schemes of baffling, it is impossible to group together a set of curves for the draft loss through

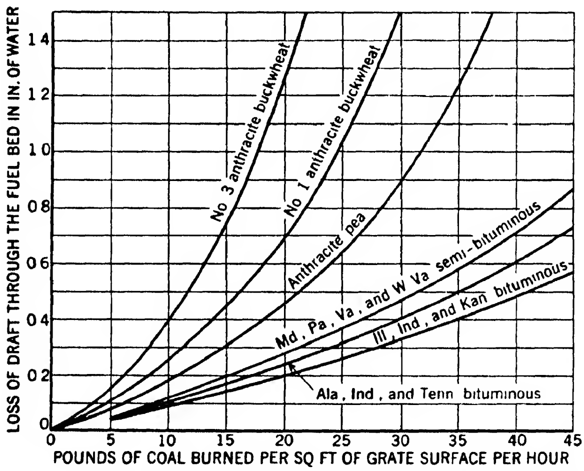

FIG. 8 DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

the boiler which may even be used generally. It is therefore necessary to secure this information from the manufacturer of the particular type of boiler and baffle arrangement under consideration.

When a boiler is installed and in operation, the draft loss depends upon the amount of gases flowing through it This, in turn, depends upon the proportion of excess air admitted for combustion. Primarily, the amount of excess air is measured by the CO_2 content; the less the amount of CO_2, the greater the amount of excess air and hence the greater the draft loss.

The loss of draft through the boiler will vary directly as the size of the boiler and the length of the gas passages within. The loss also varies as the number of tubes high, but not in a direct ratio inasmuch as the loss due to the reversal of flow at the ends of the baffles remains constant regardless of the height of the boiler. The arrangement of the tubes, whether the gases flow parallel to or at right angles to the tubes, has an appreciable effect on the loss. The arrangement of the baffles influences the draft loss greatly, the loss through a boiler with five passes being

greater than the loss through one of three or four passes. A poor design and a rough condition of the brickwork will increase the loss greatly, whereas a proper design and a smooth condition will keep the loss at a minimum. The loss through the boiler will be less when the breeching entrance is located at or near the top of the boiler than when it is located at or near the bottom since the gases have a shorter distance to travel in the former instance.

The *draft loss through the breeching* (h_{Br}) is given by the general equation:

$$h_{Br} = \frac{0.000194\,W^2 T_c f L}{A^2 B_o W_c C_{br}} \tag{16}$$

where

W = the amount of gases flowing, pounds per second
T_c = absolute temperature of breeching gases, degrees Fahrenheit
f = coefficient of friction.
L = length of breeching, feet.
A = area of breeching, square feet.
B_o = atmospheric pressure corresponding to altitude, inches of mercury.
W_c = weight of a cubic foot of breeching gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
C_{br} = hydraulic radius of breeching section.

It has been the general custom to *lump off* the intensity of the breeching loss at 0.10 in. of water per 100 ft of breeching length regardless of its size or shape or the amount and temperature of the gases flowing through it. This practice is hazardous and has no more foundation in fact than that of determining the friction head in a water works system without taking into consideration the size of the pipe or the amount of water flowing through it. When the length of the breeching is relatively short, any variation in any one of the factors in the equation will have no appreciable effect on the draft loss. However, when the breeching is relatively long, the draft loss is affected greatly by the various factors, particularly by the size and shape as well as by the weight of gases flowing.

The *draft loss due to velocity* (h_V) is given by the equation

$$h_V = \frac{0.000194\,W^2 T_c}{A^2 B_o W_c} \tag{17}$$

and represents the amount of draft required to accelerate the gases from zero velocity to the velocity at which the gases are flowing, or in other words, from a static gas condition of zero flow to the amount of gases flowing throughout the installation. This loss corresponds to the velocity head in water works systems.

The *draft loss due to bends* (h_{Bd}) is equivalent to the loss due to the velocity head for a 90-deg bend. In changing direction of flow, the gas velocity decreases to zero with a loss of velocity head and then increases to its proper value at the expense of a loss in pressure head, the net result being a loss in pressure head equal to the velocity head at the bend. This loss is given by the equation:

$$h_{Bd} = \frac{0.000194\,W^2 T_c}{A^2 B_o W_c} \tag{18}$$

The friction at a right-angle bend is sometimes expressed as the equivalent of a straight length of flue of a certain length for a certain diameter, similar to the procedure used in estimating the loss due to bends in piping systems conducting water. Most flues, however, particularly breechings, are built square or rectangular in section and no general equation based on the shape of the flue can be conveniently expressed.

The *draft loss due to sudden contraction of an area* (h_C) is given by the equation:

$$h_C = \frac{0.000194 K_C W^2 T_C}{A_s^2 B_O W_C} \tag{19}$$

where

K_C = coefficient of sudden contraction based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= 0.5\left(1 - \frac{A_s}{A_1}\right)$

A_s = area of the smaller section

When the flue or passage through which the gases flow is suddenly contracted, a considerable portion of the static head in the larger section is converted into velocity head and a draft loss of some consequence, particularly in a short breeching, takes place. A sudden contraction should always be avoided where possible. At times, however, due to obstructions or limited head-room, it is necessary to alter the size of the breeching, but a sudden contraction may be avoided by gradually decreasing the area over a length of several feet.

The *draft loss due to a sudden enlargement of an area* (h_O) is given by the equation:

$$h_O = \frac{0.000194 K_O W^2 T_C}{A_s^2 B_O W_C} \tag{20}$$

where

K_O = coefficient of sudden enlargement based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= \left(1 - \frac{A_s}{A_1}\right)^2$

When the flue or passage through which the gases flow is suddenly enlarged, a portion of the velocity head is converted into static head in the larger section and, like the loss due to sudden contraction, a loss of some consequence, particularly in short breechings, takes place. A sudden enlargement in a breeching may be avoided by gradually increasing the area over a length of several feet. In large masonry chimneys, the area of the flue at the region of the breeching entrance is considerably larger than the area of the breeching at the chimney, and a sudden enlargement exists.

The *draft loss through the economizer* (h_E) should be obtained from the manufacturer but for general purposes it may be computed from the following general equation:

$$h_E = \frac{6.6 W_n^2 N T_C}{10^{12}} \tag{21}$$

where

W_n = pounds of gases flowing per hour per linear foot of pipe in each economizer section.

N = number of economizer sections.

An economizer in a steam plant affects the draft in two ways, (1) it offers a resistance to the flow of gases, and (2) it lowers the average chimney gas temperature, thereby decreasing the available intensity. In the case of a natural draft installation, both of these factors result in a relative increase in the height of the chimney and, in the case of a large plant, they may add as much as 20 or 30 ft to the height. The decrease in the temperature of the gases after they have passed through the economizer has an extremely important effect on the performance of a natural draft chimney and also upon the performance of a fan.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Standard Ordinance for Chimney Construction of the *National Board of Fire Underwriters.* Briefly summarized, these provisions are as follows for heating boilers and furnaces:

The construction, location, height and area of the chimney to which a heating boiler or warm-air furnace is connected affect the operation of the entire heating system. Most residence chimneys are built of brick and may be either lined or unlined, but in either case the *walls must be air-tight* and there should be *only one smoke opening* into the chimney. Cleanout, if provided, must be *absolutely air-tight* when closed.

The walls of brick chimneys shall be not less than 3¾ in. thick (width of a standard size brick) and shall be lined with fire-clay flue lining. Fire-clay flue linings shall be manufactured from suitable refractory clay, either natural or compounded, and shall be adapted to withstand high temperatures and the action of flue gases. They shall be of standard commercial thickness, but not less than ¾ in. All fire-clay flue linings shall meet the standard specification of the *Eastern Clay Products Association.* The flue sections shall be set in special mortar, and shall have the joints struck smooth on the inside. The masonry shall be built around each section of lining as it is placed, and all spaces between masonry and linings shall be completely filled with mortar. No broken flue lining shall be used. Flue lining shall start at least 4 in. below the bottom of smoke-pipe intakes of flues, and shall be continued the entire heights of the flues and project at least 4 in. above the chimney top to allow for a 2 in. projection of lining. The wash or splay shall be formed of a rich cement mortar. To improve the draft the wash surface should be concave wherever practical.

Flue lining may be omitted in brick chimneys, provided the walls of the chimneys are not less than 8 in. thick, and that the inner course shall be a refractory clay brick. All brickwork shall be laid in spread mortar, with all joints push-filled. Exposed joints both inside and outside shall be struck smooth. No plaster lining shall be permitted.

Chimneys shall extend at least 3 ft above flat roofs and 2 ft above the ridges of peak roofs when such flat roofs or peaks are within 30 ft of the chimney. The chimney shall be high enough so that the wind from any direction shall not strike the top of the chimney from an angle above the horizontal. The chimney shall be properly capped with stone, terra cotta, concrete, cast-iron, or other approved material; but no such cap or coping shall decrease the flue area.

There shall be but one connection to the flue to which the boiler or furnace smoke-pipe is attached. The boiler or furnace smoke-pipe shall be thoroughly grouted into the chimney and shall not project beyond the inner surface of the flue lining.

The size or area of flue lining or of brick flue for warm-air furnaces depends on height of chimney and capacity of heating system. For chimneys not less than 35 ft in height above grate line, the net internal dimensions of lining should be at least 7 x 11½ in.

for a total leader pipe area up to 790 sq in. Above 790 and up to 1,000 sq in. of leader pipe area the lining should be at least 11¼ x 11¼ in. inside. In case of brick flues not less than 35 ft in height with no linings, the internal dimensions should be at least 8 x 12 in. up to 790 sq in. of leader area, and at least 12 x 12 in. for leader capacities up to 1,000 sq in. Chimneys under 35 ft in height are unsatisfactory in operation and hence should be avoided.

CHIMNEYS FOR GAS HEATING

The burning of gas differs from the burning of coal in that the force which supplies the air for combustion of the gas comes largely from the pressure of the gas in the supply pipe, whereas air is supplied to a bed of burning coal by the force of the chimney draft. If, with a coal-burning boiler, the draft is poor, or if the chimney is stopped, the fire is smothered and the combustion rate reduced. In a gas boiler or furnace such a condition would interfere with the combustion of the gas, but the gas would continue to pass to the burners and the resulting incomplete combustion would produce a dangerous condition. In order to prevent incomplete combustion from insufficient draft, all gas-fired boilers and furnaces should have a back-draft diverter in the flue connection to the chimney.

TABLE 2. SUGGESTED GENERAL DIMENSIONS FOR VERTICAL BACK-DRAFT DIVERTER

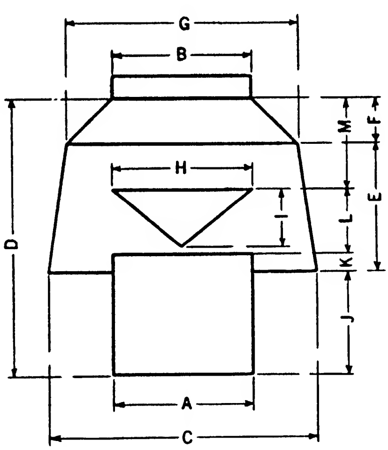

Table of Dimensions (In)

PIPE SIZE	A	B	C	D	E	F	G	H	I	J	K	L	M
3	3	3	5.5	7.0	3.8	0.7	4.4	3 0	1.5	2.5	0.7	1 5	2.3
4	4	4	7.2	9.5	5.0	1.0	6 0	4.0	2 0	3.5	1 0	2.0	3.0
5	5	5	9.4	10.8	5.3	1.5	8.0	5 0	2.3	4.0	0.9	2.4	3.5
6	6	6	11 5	12 0	5.6	1.9	9.8	6 0	2 5	4 5	0.8	2.7	4 0
7	7	7	13.5	13 9	6.4	2.3	11 6	7 0	2.9	5.3	0.9	3.1	4.6
8	8	8	15.5	15.8	7.1	2.7	13.4	8 0	3 2	6.0	1.0	3.5	5.3
9	9	9	17.5	17.5	7.7	3.1	15.2	9 0	3.5	6.7	1.0	4.0	5.8
10	10	10	19.7	18 8	7 9	3.6	17 2	10 0	3.8	7.3	1.0	4.3	6.2
11	11	11	22.2	20.7	8.4	4.3	19.6	11 0	4.1	8.0	1.5	4.6	6.6
12	12	12	24.7	22 2	8.7	5.0	22.0	12.0	4.4	8.5	1.7	5.0	7.0

A study of a typical *back-draft diverter* shows that partial or complete chimney stoppage will merely cause some of the products of combustion to be vented out into the boiler room, but will not interfere with combustion. In fact, gas-designed appliances must perform safely under such a condition to be approved by the *American Gas Association* Laboratory. Other functions of the back-draft diverter are to protect the burner and pilot from the effects of down-drafts, and to neutralize the effects of variable chimney drafts, thus maintaining the appliance efficiency at a substantially constant value. Converted boilers or furnaces, as well as gas-designed appliances, should be provided with back-draft diverters.

Since back-draft diverters have a special function to perform in protecting gas burning appliances, it is necessary that they should be built to the proper size as shown in Table 2 for a vertical type and in Table 3 for a horizontal arrangement. Equipment of this kind listed by the *American Gas Association* Testing Laboratory must bear the listing symbol *A.G.A.*

TABLE 3. SUGGESTED GENERAL DIMENSIONS FOR HORIZONTAL BACK-DRAFT DIVERTER

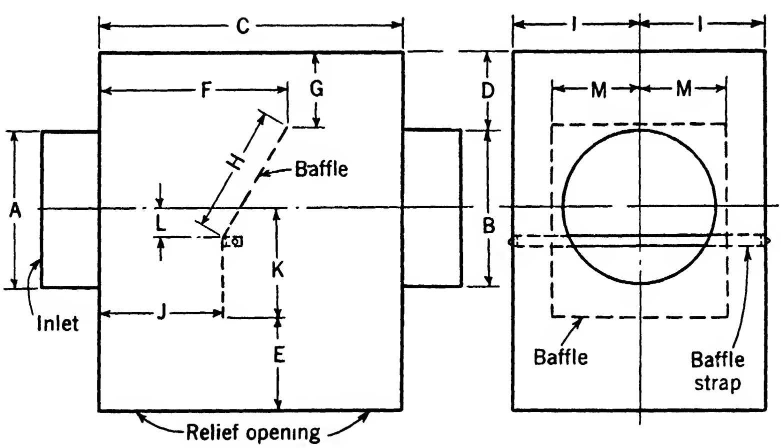

Table of Dimensions (In)

PIPE SIZE	A	B	C	D	E	F	G	H	I	J	K	L	M
3	3	3	6	1 5	4 8	3 8	1 4	2 5	2 5	2 5	2 1	0 6	1 8
4	4	4	8	2 0	4 8	5 0	1 9	3 4	3 4	3 4	2 9	0 8	2 3
5	5	5	10	2 5	4 8	6 3	2 4	4 2	4 2	4 2	3 5	0 9	2 9
6	6	6	12	3 0	4 8	7 5	2 9	5 0	5 0	5 0	4 3	1 1	3 5
7	7	7	14	3 5	4 8	8 8	3 4	5 9	5 9	5 9	5 0	1 3	4 1
8	8	8	16	4 0	4 8	10 0	3 9	6 7	6 7	6 7	5 6	1 5	4 7
9	9	9	18	4 5	4 8	11 3	4 4	7 5	7 5	7 5	6 4	1 7	5 3
10	10	10	20	5 0	4 8	12 5	4 9	8 4	8 4	8 4	7 0	1 9	5 8
11	11	11	22	5 5	4 8	13 8	5 4	9 2	9 2	9 2	7 8	2 1	6 4
12	12	12	24	6 0	4 8	15 0	5 9	10 0	10 0	10 0	8 5	2 3	7 0

As is the case with the complete combustion of almost all fuels, the products of combustion for gas are carbon dioxide (CO_2) and water vapor with just a trace of sulphur trioxide (SO_3). Sulphur usually burns to the trioxide in the presence of an iron oxide catalyst. The volume of water vapor in the flue products is about twice the volume of the carbon dioxide when coke oven or natural gas is burned. Because of the large quantity of water vapor which is formed by the burning of gas, it is quite important that all gas-fired central heating plants be connected to a chimney having a good draft. Lack of chimney draft causes stagnation of the products of combustion in the chimney and results in the condensation of a large amount of the water vapor. A good chimney draft draws air into the chimney through the openings in the back-draft diverter, lowers the dew-point of the mixture, and reduces the tendency of the water vapor to condense.

The flue connections from a gas-fired boiler or furnace to the chimney should be of a non-corrosive material. In localities where the price of gas requires the use of highly efficient appliances, the material used for the flue connection not only should be resistant to the corrosion of water, but should resist the corrosion of dilute solutions of sulphur trioxide in water. Local practice should be followed in the selection of the most appropriate flue materials.

TABLE 4. MINIMUM ROUND CHIMNEY DIAMETERS FOR GAS APPLIANCES (INCHES)

Height of Chimney Feet	Gas Consumption in Thousands of Btu per Hour								
	100	200	300	400	500	750	1000	1500	2000
20	4.50	5.70	6.60	7.30	8 00	9 40	10.50	12 35	13 85
40	4.25	5.50	6 40	7 10	7.80	9.15	10 25	12.10	13.55
60	4 10	5.35	6 20	6 90	7.60	8 90	10.00	11.85	13.25
80	4 00	5.20	6 00	6 70	7.35	8 65	9 75	11.50	12 85
100	3.90	5.00	5.90	6.50	7.20	8.40	9.40	11.00	12.40

When condensation in a chimney proves troublesome, it may be necessary to provide a drain to a dry well or sewer. The cause of the excessive condensation should be investigated and remedied if possible. This may be done by raising the flue temperature slightly or increasing the size of the back-draft diverter. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, the spraying with a water emulsion of asphalt-chromate provides an excellent protection.

A chimney for a gas-fired boiler or furnace should be constructed in accordance with the principles applicable to other boilers. Where the wall forming a smoke flue is made up of less than an 8-in. thickness of brick, concrete, or stone, a burnt fire-clay flue tile lining should be used. Care should be used that the lengths of flue tile meet properly with no openings at the joints. Cement mortar should be used for the entire chimney.

Table 4 gives the minimum cross-sectional diameters of round chim-

neys (in inches) for various amounts of heat supplied to the appliance, and for various chimney heights. This is in accordance with *American Gas Association* recommendations.

PROBLEMS IN PRACTICE

1 ● What are the principle factors influencing the intensity of natural draft?

The intensity of natural draft depends largely upon the height of chimney above the grate bar level and the temperature difference between the chimney gases and the atmosphere.

2 ● What two kinds of draft need be considered?

Natural draft caused by temperature differences, and artificial draft caused by mechanical forcing.

3 ● What is the effective height of a chimney?

The height from the grate level to the top of the chimney is the effective height in producing natural draft.

4 ● What dual purpose does a tall chimney fulfill?

A tall chimney primarily creates the necessary draft to move the air required for the combustion process and to move the products of combustion, and secondarily it discharges the gases at a high elevation to prevent them from becoming a nuisance.

5 ● What is the direct influence of the height on the design of a chimney?

The immediate purpose of height is to provide that draft intensity under the conditions of chimney gas temperature such that it will be adequate to overcome all the frictional resistances of the installation, as well as to provide for the actual gas movement

6 ● Of what importance is chimney cross-sectional area in stack design?

The area should be as large as is economically feasible in order that the frictional loss for the chimney height should not destroy the effectiveness of the height-created draft in overcoming the necessary frictional resistances of the boiler and its flue connections.

7 ● Of what importance is the Reynolds number in chimney design?

It permits the selection of a more specific value of the chimney friction factor, rather than a general one, to correspond with conditions of size, nature of the gas, rate of gas flow, and condition of the surface.

8 ● a. Name the principle advantages of natural draft.
b. Name the principle disadvantages of natural draft.

a. Simplicity, reliability, freedom from mechanical parts, low cost of maintenance, relatively long life, relatively low depreciation, operation with no power requirement.
b. Lack of flexibility, irregularity, dependence on surroundings, susceptibility to temperature changes.

9 ● How is mechanical draft created?

By forced draft, by induced-draft fans, or by a Venturi chimney.

10 ● Distinguish between theoretical and available draft.

Theoretical draft is the difference in pressure inside and outside the base of a chimney when it is under operating temperatures but when there are no gases flowing. Available draft is less than theoretical draft by the friction loss due to the flow of gases through the chimney.

11 ● Explain the term efficiency of a natural draft chimney.

The efficiency of a chimney is the ratio of the work it does in moving gases to the theoretical amount of power it generates.

12 ● How is the available draft used in a heating plant?

The available draft at the base of the chimney is used to overcome the loss in pressure through the grate, the fuel bed, the boiler passes, the breeching.

13 ● What are some of the factors that influence the draft loss through the fuel bed?

Uniformity and size of coal, the amount of ash mixed with the fuel on the grate, thickness of fuel bed, rate of combustion, amount of air supply as related to the coal burning rate.

14 ● How does the volatile matter content affect the draft loss through the fuel bed?

The higher the volatile content and the lower the fixed carbon content, the lower the draft loss

15 ● In what cases will there be no fuel bed draft loss?

In oil, gas, and powdered fuel firing the fuel is mixed and burned in suspension, consequently, no measurable resistance is encountered in the combustion zone.

16 ● Is it possible to state an average value for the draft loss through a boiler and its setting?

No The draft loss varies widely and depends on many factors such as the size and type of gas passageways. The manufacturer is usually able to supply such information

17 ● Of what significance is the CO_2 content of stack gases in establishing draft loss?

The CO_2 content of the exit gases is a measure of the completeness of the combustion and the amount of excess air supplied Low CO_2 indicates a high excess of air and hence a high draft loss.

18 ● What two effects does an economizer have on the draft loss?

An economizer offers resistance to the flow of gases over the added surfaces; it lowers the temperature of the gases going to the chimney and therefore decreases the available draft. This decrease often necessitates the addition of forced draft

19 ● What main provisions should be considered in good chimney construction?

Chimneys should be air-tight and connected to only one smoke opening. The chimney top should be high enough above surroundings so the wind will not strike it at any angle above the horizontal. Chimney walls should be not less than one brick in width, and they should be lined with fire-clay tile of the size required for the attached heating unit. Tile lining sizes are stated as outside dimensions, therefore, their effective dimensions are less by the thickness of the wall.

20 ● What is the purpose of a back-draft diverter as used on gas burning units?

Since the fuel is supplied under pressure independent of draft it is necessary to free the unit from the variable chimney draft and to supply air for combustion in direct proportion to the supply of fuel gas. The back-draft diverter protects the pilot and burners from down-drafts.

Chapter 11

AUTOMATIC FUEL BURNING EQUIPMENT

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Classification of Oil Burners, Combustion Chamber Design, Classification of Gas-Fired Appliances

AUTOMATIC mechanical equipment for the combustion of solid, liquid and gaseous fuels is considered in this chapter.

MECHANICAL STOKERS

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber, provides a supply of air for burning the fuel under automatic control and, in some cases, incorporates a means of removing the ash and refuse of combustion automatically. Coal can be burned more efficiently by a mechanical stoker than by hand firing because the stoker provides a uniform rate of fuel feed, better distribution in the fuel bed and positive control of the air supplied for combustion.

Stokers may be divided into four types according to their construction, namely, (1) overfeed flat grate, (2) overfeed inclined grate, (3) underfeed side cleaning type, and (4) underfeed rear cleaning type.

Overfeed Flat Grate Stokers

This type is represented by the various chain- or traveling-grate stokers. These stokers receive fuel at the front of the grate in a layer of uniform thickness and move it back horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ashpit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the stoker to maintain ignition of the incoming fuel. Frequently, a rear combustion arch is required to maintain ignition until the fuel is fully consumed. A typical traveling-grate stoker is illustrated in Fig. 1.

Another and distinct type of overfeed flat-grate stoker is the spreader (Fig. 2) or sprinkler type in which coal is distributed either mechanically or by air over the entire grate surface. This type of stoker has a wide application on small sized fuels and on certain special fuels such as lignites, high-ash coals, and coke breeze.

Overfeed Inclined Grate Stokers

In general the combustion principle is similar to the flat grate stoker, but this stoker (Fig. 3) is provided with rocking grates set on an incline to advance the fuel during combustion. Also this type is provided with an ash plate where ash is accumulated and from which it is dumped periodically. This type of stoker is suitable for all types of coking fuels but preferably for those of low volatile content. Its grate action has the tendency to keep the fuel bed well broken up thereby allowing for free

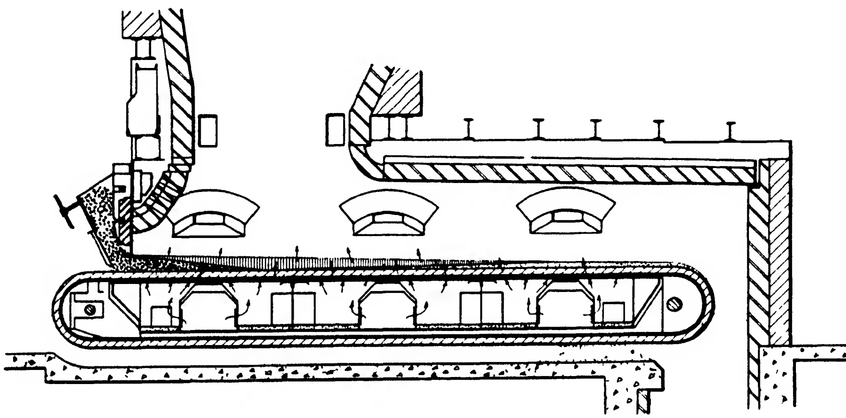

FIG. 1. OVERFEED TRAVELING-GRATE STOKER

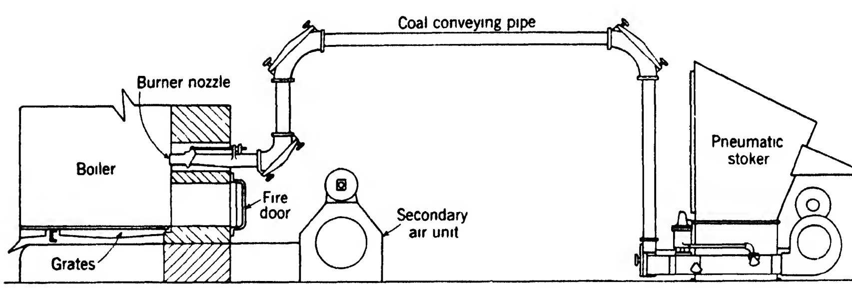

FIG. 2. SPREADER STOKER-PNEUMATIC TYPE

passage of air. Because of its agitating effect on the fuel it is not so desirable for badly clinkering coals. Furthermore, it should usually be provided with a front arch to care for the volatile gases.

Underfeed Side Cleaning Stokers

In this type (Fig. 4), the fuel is introduced at the front of the furnace to one or more retorts, is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of stoker is suitable for all coking coals while in the smaller sizes it is suitable for small sizes of anthracites. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process

the volatile gases are released, are mixed with air, and pass through the fire where they are burned. The ash may be continuously discharged as in the small stoker or may be accumulated on a dump plate and periodically discharged. This stoker requires no arch as it automatically provides for the combustion of the volatile gases.

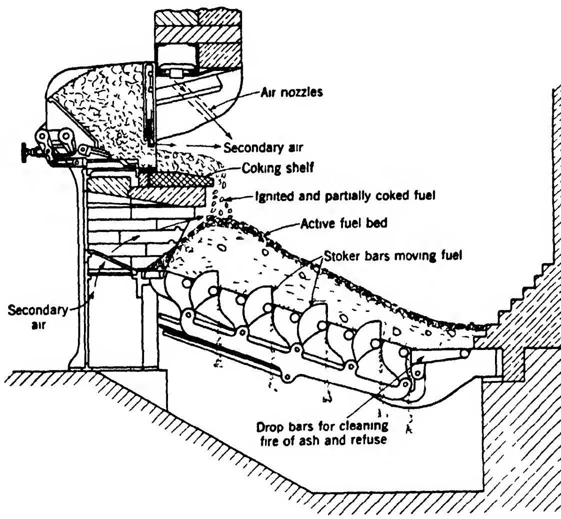

FIG 3 OVERFEED INCLINED GRATE STOKER

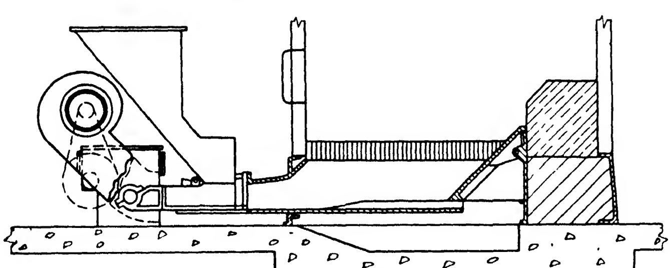

FIG. 4. UNDERFEED SIDE CLEANING STOKER

Underfeed Rear Cleaning Stokers

This type of stoker accomplishes combustion in much the same manner as the side cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed.

Stokers also may be classified according to their size based upon coal feed rates. The following classification has been made by the *United States Department of Commerce* in cooperation with the *Stoker Manufacturers' Association.*

Class 1. Up to and including 60 lb of coal per hour.

Class 2. 60 to 100 lb of coal per hour.

Class 3. 100 to 300 lb of coal per hour.
Class 4. 300 to 1200 lb of coal per hour.
(A fifth class is included covering stokers having a feeding capacity above 1200 lb of coal per hour).

Class 1 and Class 2 Stokers, Household

Since these stokers are used primarily for home heating, it is desirable that their design be simple and attractive in appearance, and that they be quiet and automatic in operation.

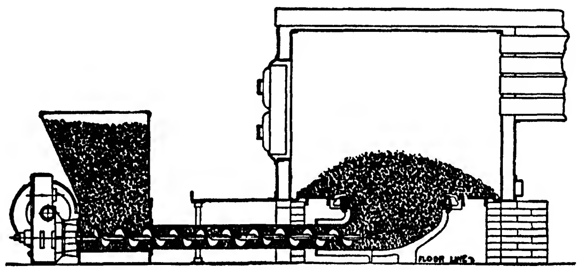

FIG. 5 UNDERFEED SCREW STOKER, HOPPER TYPE

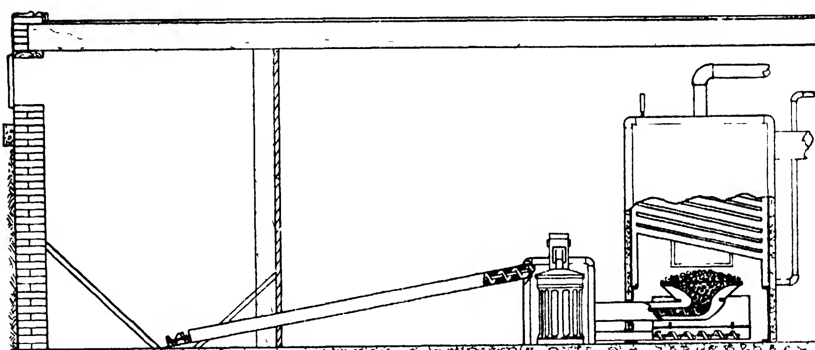

FIG. 6. UNDERFEED SCREW STOKER WITH AUTOMATIC ASH REMOVAL

A common type of stoker in this class consists, essentially of a coal reservoir or hopper, a screw for conveying the fuel from the hopper to the burner head or retort, a fan which supplies the air for combustion, a transmission for driving the coal feed worm, and an electric motor or motors for supplying the motive power for both coal feed and air supply as indicated in Fig. 5. The shape of the retort in this class of stokers is usually round although rectangular retorts are favored by some manufacturers. In all cases, however, the retort incorporates tuyeres through which the air for combustion is admitted.

Some household stokers are provided with an automatic grate-shaking mechanism together with screw conveyors for removing the ash from the ashpit (Fig. 6) and depositing it in an ash receptacle outside the boiler.

Certain types can also be provided with a coal conveyor which takes coal from the storage bin and maintains a full hopper at the stoker. In some cases the coal bin functions as the stoker hopper as shown in Fig. 7, and an extended worm is used to convey the fuel to the combustion furnace.

Domestic stokers may feed coal to the furnace either intermittently or with a continuous flow regulated automatically to suit conditions.

Household stokers are made for all classes of fuel; anthracite, bituminous and semi-bituminous coals, and coke. The *United States Department of Commerce* has issued commercial standards for household anthracite burners, which may be obtained by application. Standards of performance for bituminous coal stokers are also being developed by *Bituminous Coal Research, Inc.* The standards for anthracite stokers are described in the next paragraphs.

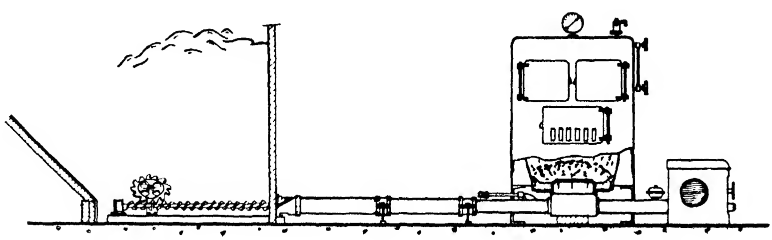

FIG. 7. UNDERFEED SCREW STOKER, BIN FEED TYPE

Operating Requirements for Anthracite Stokers

Efficiency. The over-all efficiency of the unit at all points above 50 per cent of maximum coal feed shall be above 50 per cent when installed in a round sectional cast-iron boiler having three intermediate sections and 1½ in of asbestos insulation or its equivalent in good condition of repair, operating at 50 per cent or more of the boiler capacity. The efficiency shall be maintained for any continuous period of 4 hours during any test or observation run.

Ash Loss. Combustible in ash shall not exceed 7.5 per cent of the Btu content of the coal as fired at any rate of coal feed above 50 per cent of maximum. Subsequent to the issuance of these standards the Society adopted a Code[1] which should be followed in all details applicable to stoker testing

Clinker. Ash removing systems should at all times be capable of disposing of any clinker which may be formed under any conditions of operation with the coals prescribed.

Combustion Rate. A combustion rate of at least 13 lb per square foot of horizontal projected area of ash ring per hour must be continuously maintained for at least 9 hours with the above conditions of efficiency, ash and clinker.

Flue Gas. Flue gas shall be not below 6 per cent in carbon dioxide with a reasonably tight boiler at any rate of operation above 50 per cent of maximum coal feed.

Maximum Rating. The maximum rating, in terms of gross square feet of water or steam radiation which the burner will supply, when intended for installation in the average existing cast-iron boiler, shall be 90 per cent of the maximum steam produced in a round cast-iron boiler in good repair having three intermediate sections and the equivalent of 1½ in. of asbestos insulation. However, in no case shall the maximum

[1]A S H V E Standard Code for Testing Stoker-Fired Steam-Heating Boilers (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1938, p 613)

rating be greater than 29 sq ft of direct steam radiation for each pound of coal fired per hour, and in no case shall ratings be based upon efficiency figures below 50 per cent.

The maximum rating as defined in the preceding paragraph shall be based upon combustion of Pennsylvania anthracite having the following approximate analysis:

Volatile matter 3.5 to 9 per cent, ash content not to exceed 15 per cent; sulphur content under 1.5 per cent; ash fusing temperature 2750 F, or above (volatile, ash and sulphur content on dry basis in accordance with *A.S.T.M.* method D271-33); Btu content 12,000 or above; properly sized as follows: A No. 1 buckwheat should pass through a round mesh screen having 9/16 in. holes and over a similar screen having 5/16 in. holes. The undersizing should not exceed 15 per cent and the oversizing should not exceed 10 per cent. A No. 2 buckwheat (rice) should pass through a round mesh screen having holes 5/16 in. in diameter and over a like screen having holes of 3/16 in. in diameter. The undersizing should not exceed 15 per cent and the oversizing should not exceed 10 per cent

Coal Storage It is recommended that the coal bin or closet be constructed so as to be dustproof.

Electrical Consumption. The electrical consumption shall not exceed 18 kwh per 2000 lb of coal burned at any rate of coal feed above 50 per cent of the maximum.

Operation Upon Other Sizes of Coal. The foregoing specifications have been drafted for operating with the Nos. 1 and 2 buckwheat sizes of anthracite. In the event that other sizes are recommended, ratings shall be based upon the same efficiency and ash loss requirements.

Banking. The burner shall be so constructed or controlled as to maintain a fire during an indefinite banking period.

Acceleration When the burner resumes operation after a 12 hour banking period, the time required for the stack temperature to reach a normal maximum shall not exceed 60 min.

Stoker-Fired Units

Boilers and air conditioners especially designed for stokers are now becoming available. Present designs feature better coordination between the heat absorber and the stoker. Increased setting height and furnace volume and more heating surface than in conversion installations are usually to be found in these stoker units.

Class 3 Stokers, Apartment House, Small Commercial

This class is used extensively for heating plants in apartments and hotels, and also for small industrial plants such as laundries, bakeries, and creameries. The majority of stokers used in this field are of the underfeed type. The principal exception is an overfeed type having step action grates in a horizontal plane and so arranged that they are alternately moving and stationary, and are designed to advance the fuel during combustion to an ash plate at the rear.

All of the stokers are provided with a coal hopper outside of the boiler. In the underfeed types, the coal feed from this hopper to the furnace may be accomplished by a continuously revolving screw or by an intermittent plunger. The drive for the coal feed may be an electric motor, or a steam or hydraulic cylinder. With an electric motor, the connection between the driver and the coal feed may be through a variable speed gear train which provides two or more speeds for the coal feed; or it may be through a simple gear train and a variable speed driver for the change in speed of the coal feed; or a simple gear train with a coal feed having an adjustment for varying the travel of the feeding device. With a steam or hydraulic cylinder, the power piston is connected directly to the coal feeding plunger.

The stokers in this class vary also in their retort design according to the fuels and load conditions. The retort is placed approximately in the middle of the furnace and is provided with tuyere openings at the top on all sides. In the plunger-feed type the retort extends from the inside of the front wall entirely to the rear wall or to within a short distance of the rear wall. This type of retort has tuyeres on the sides and at the rear.

These stokers also differ in the grate surface surrounding the retort. In many of the worm-feed stokers this grate is entirely a dead plate on which the fuel rests while combustion is completed. In the dead-plate type, all of the air for combustion is furnished by the tuyeres at the retort. Because of this, combustion is well advanced over the retort so that it may easily be completed by the air which percolates through the fuel bed. With the dead-plate type of grate the ash is removed through the fire doors and it is therefore desirable that the fuel used shall be one in which the ash is readily reduced to a clinker at the furnace temperature, in order that it may be removed with the least disturbance of the fuel bed.

In other stokers in this class, the grates outside of the retort are air-admitting and some stokers have shaking grates. These grates permit a large part of the ash to be shaken into the ashpit beneath, while the clinkers are removed through the fire doors. With this type of grate, the main air chamber extends only under the retort while the side grates receive air by natural draft from the ashpit.

In still other stokers of this class, the main air chamber extends beyond the retort and is covered with fuel-bearing, air-supplying grates. With this type of grate, the fuel is supplied with air from the main air chamber throughout combustion. Also with this type of grate, dump plates are provided beyond the grates where the ash accumulates and from which it can be dropped periodically into the ashpit beneath.

Stokers in this class are compactly built in order that they may fit into standard heating boilers and still leave room for sufficient combustion space above the grates. The height of the grate is approximately the same as that of the ordinary grates of boilers, so that it is usually possible to install such stokers with but minor changes in the existing equipment. In some districts, there are statutory regulations governing such settings.

These stokers vary in furnace dimensions from 30 in. square to approximately 66 in. square. The capacity of the stokers is measured by the amount of coal that can be burned per hour. In general, manufacturers recommend that, for continuous operation, the coal burning rate shall not exceed 25 lb of coal per square foot of grate per hour, while for short peaks this rate may be increased to 30 lb per hour. Although these stokers were designed to burn bituminous coal, types are available for the semi-bituminous coals such as Pocahontas and New River. They can also be used to burn the small sizes of anthracite but at a somewhat lower rate.

Class 4 Stokers, Medium Commercial

These stokers are usually of the screw feed type without auxiliary plungers or other means of distributing the coal. Rectangular retorts with sectional tuyeres and dead plates without air ports are employed.

The unit type of construction is almost universally used, the unit incorporating the hopper, the transmission for driving the feed screw, and the fan for supplying air for combustion.

Class 4 Stokers, Large Commercial, Small High Pressure Plants

Stokers in this group vary widely in details of mechanical design and the several methods of feeding coal previously described may be employed Such methods of applying power to the fuel conveying mechanism as, continuous gear train transmission, ratchet-type speed reducer, hydraulic cylinder and steam cylinder are used. Varying methods of ash disposal are found in this class.

Large Stokers

This class includes stokers with hourly burning rates of over 1200 lb of coal per hour. The prevalent stokers in this field are:

a. Overfeed flat grate stokers
b. Overfeed inclined grate stokers.
c. Underfeed side cleaning stokers.
d. Underfeed rear cleaning stokers.

Overfeed inclined grate stokers are seldom built in sizes of over 500 hp and are not as extensively used as other types of stokers.

Underfeed side cleaning stokers are made in sizes up to approximately 500 hp and in this field are extensively used. These stokers are not so varied in design as those in the smaller classes although the principle is much the same. Practically all of them are of the front coal feed type, either power driven or steam driven. Dump plates at the side are manually operated. These stokers are heavily built and designed to operate continuously at high boiler ratings with a minimum amount of attention. Because of the fact that all volatile gases must pass through the fire before reaching the combustion chamber, these stokers will operate smokelessly under ordinary conditions. Also because of the fact that these stokers are always provided with forced draft, they are the most desirable type for fluctuating loads or high boiler ratings.

In the design of the grates for supporting the fuel between the retort and the ash plates, the stokers differ in providing for movement of the fuel during combustion. Some stokers are designed with fixed grates of sufficient angle to provide for this movement as the bed is agitated by the incoming fuel, while others have alternate moving and stationary bars in this area and provide for this movement mechanically. In either type, with proper operation, all refuse will be deposited at the dump plate. Recent developments in this type of stoker provide for sliding distributor blocks along the bottom of the retorts which give flexibility in providing proper distribution of fuel over the grate area and assist in preventing coke masses when strong coking coals are used. Another difference in these stokers is that some use a single air chamber under the whole grate area thus having the same air pressure under the ignition area as under the rest of the grate, while others have a divided air chamber using the full air pressure under the ignition area and a reduced air pressure under

the remainder of the grate. These stokers vary in size from approximately 5 sq ft to a maximum of 8½ sq ft.

The most prevalent type of rear cleaning underfeed stoker is the multiple retort design. Occasionally double or triple retort side cleaning underfeeds are made. The multiple retort underfeed stoker is made for the largest sizes of boilers for large industrial plants and central stations. This stoker has reached a very fine stage of development mechanically and in the matter of air supply and control. In some instances zoned air control has been applied both longitudinally and transversely to the grate surface. Ash dumps on smaller sizes are sometimes manually operated.

The Combustion Process

Due to the marked differences in design and operating characteristics of stokers and the widely differing characteristics of stoker fuels, it is difficult to generalize on the subject of combustion in automatic stokers.

In anthracite stokers, which are almost exclusively of the small (Class 1) underfeed type, burning takes place within the stoker retort. The ash and refuse of combustion spills over the edge of the retort into an ashpit or receptacle from which it may be removed either manually or automatically. Anthracite is usually supplied for stoker firing in No. 1 buckwheat or No. 2 buckwheat size. Those stokers burning coke operate in a similar manner to anthracite stokers.

Since the majority of bituminous coal stokers used in heating plants operate on the underfeed principle some general observations of their operation are given.

When the coal is fed from the hopper or bin into the retort it is generally degraded to some extent and some segregation of sizes occurs. Because of these factors there may be some difference in the actions occurring in the various portions of the retort.

The coal moving upward in the retort toward the zone of combustion established by previous kindling of the fire is heated by conduction and radiation from the zone of combustion. As the temperature of the coal rises it first gives off moisture and occluded gases, which are largely non-combustible. When the temperature increases to around 700 or 800 F, the coal particles become plastic, the degree of plasticity varying with the type of coal.

A rapid evolution of combustible volatile matter occurs during and directly after the plastic stage of the coal. The distillation of volatile matter continues above the plastic zone and the coal is coked. The strength and porosity of the coke formed will vary according to the size and characteristics of the coal used.

As more coal is fed from below the mass of coke continues to grow forming a coke *tree*, *plug* or *spar* as it is variously designated. After a period of time, dependent upon the strength of the coke formed, pieces of the coke *tree* break off and fall upon the hearth surrounding the retort or within the retort itself where they are burned.

While part of the ash fuses into particles at the surface of the coke as it is released, most of it is freed in unfused flakes or grains. The greater part of this unfused ash remains on the hearth or dead plates although a part may be expelled from the furnace with the gases.

The ash layer becomes thicker with time and that near the retort, being exposed to temperatures which are high enough at times, fuses into a clinker. The temperature attained in the fuel bed, the chemical composition and homogeneity of the ash, and the time of heating are factors which govern the degree of fusion.

Bituminous coal stokers of the Class 1 type operate on the principle of the removal of ash as clinker and clinker tongs are provided to facilitate this purpose. Typical representations of underfeed bituminous stoker fuel beds are shown in Figs. 8 and 9.

The appearance of such fuel beds is very ragged at times, and large masses of coke build up, surrounded by blowholes with intense white flame indicating the presence of excess air. There is a natural tendency for users to disturb the fuel bed and make it conform to the conventional

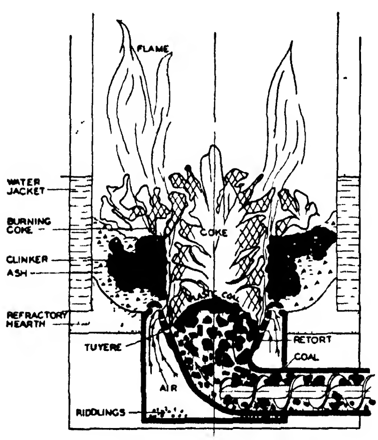

FIG. 8. CROSS-SECTION OF FUEL BED WITH WEAKLY COKING COAL

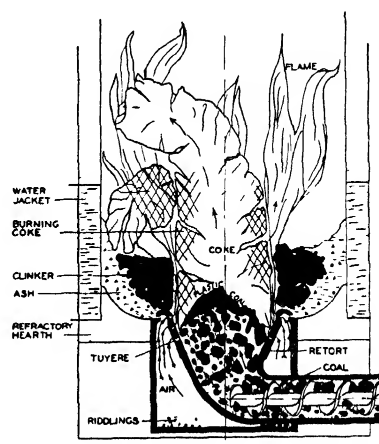

FIG. 9. CROSS-SECTION OF FUEL BED WITH STRONGLY COKING COAL

representation or ideal fuel bed. Such attention should not be required, as usually the fuel bed tends to correct its own faults as the cycles of plasticity, coke *tree* formation and ash fusion recur.

There are a number of factors which materially affect the rate and type of combustion obtained in stoker usage, the most important of these being: the type and design of stoker, the type and characteristics of the fuel, the method of stoker installation and the method of stoker operation.

Furnace Design

The burning of the fuel on the grate or in the retort will be influenced directly by the stoker design. The burning of the volatile gases above the fuel bed is a matter of furnace design. Proper care should be taken to provide furnaces sufficiently liberal in volume and with the grates or retorts at a sufficient distance from the heating surface to permit proper combustion of the gases. Smoke and low efficiency will result if the furnace is too small to permit proper mixing of the gases and completion of combustion.

The standard that has been most commonly used for the proportioning of furnaces for bituminous coal stokers is the code of the *Steel Heating Boiler Institute* (see Chapter 13).

Furnace volume is not an important item in anthracite stoker installations. Due care should be exercised for both anthracite and bituminous stokers to prevent intense heat application on the metal surfaces of the combustion chamber. The installation of a baffle or adjustment in setting height of the stoker may be desirable in some cases.

The prime essentials of good furnace design are: correct proportions, moderate combustion rate, adequate furnace volume and sufficient flame clearance. If these factors are properly compensated for and provision is made for the proper mixing of the gases bituminous coal stokers will operate smokelessly. In those stokers which are operated intermittently, however, some smoke may be produced during the *off* periods.

Combustion Adjustments

Satisfactory stoker performance may be secured by regulating the coal feed and the air supply so as to maintain, as nearly as possible, an ideal balance between the load demand and the heat liberated by the fuel. When the coal is consumed at about the same rate as that which it is fed this balance exists and uniform fuel bed conditions will be found. Under such conditions no manual attention to the fuel bed should be required other than the removal of clinker in those stokers which operate on this principle of ash removal.

Since complete combustion is not obtained in stoker furnaces receiving only the air theoretically required, it is necessary, even under the best of conditions, to supply from 30 to 50 per cent excess air to obtain desired combustion results. Due to the variable characteristics of solid fuels in burning, consideration must be given to a number of factors which affect the maintenance of the combustion conditions wanted.

The specified rate of coal feed of a stoker may vary due to changes in the bulk density of the coal dependent upon: (*a*) the size of coal, (*b*) distribution of size in the coal, (*c*) segregation of coal in the stoker hopper, and (*d*) friability of the coal.

The following factors may affect the rate of air supply: (*a*) changes in fuel bed conditions and resistance, (*b*) changes in furnace draft due to a variety of causes, i.e., changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney and changes in draft resistance of boiler due to partial stoppage of the flues, and (*c*) changes in air inlet adjustments to the fan.

Many domestic bituminous stokers now incorporate some method of automatic control which compensates for changes in fuel bed resistance. Since a secondary source of air due to leakage is present in most installations, the use of an automatic draft regulator to maintain the furnace draft at about 0.05 in. of water is desirable. This is quite important with intermittently operated stokers. Some fuel is burned by natural draft in the *off* periods, when fuel is not being fed, and it is essential that the burning in these periods be controlled. With excessive draft, due either

to fan pressure or chimney pull, an increase in the discharge of soot and fly ash from the combustion chamber will result.

Measurement of the Efficiency of Combustion

As efficient combustion is based upon a certain percentage of excess air, it is possible to determine the results by analysis of the gases formed by the combustion process. An Orsat apparatus can be used to determine the percentage (by volume) of the carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. Due to variations in the fuel bed and rate of burning of stoker-fired solid fuels it is not sufficient to analyze a *grab* sample. A continuous gas sample drawn at a constant rate throughout an operating period of reasonably long duration should be used.

A CO_2 reading of 12.5 to 14 per cent indicates that the excess air supplied is in the range of 30 to 50 per cent. The presence of CO indicates a loss due to improper mixing of the air and the gases of combustion. As an increase in excess air maintained during *on* periods will decrease the tendency toward smoke in the *off* periods of intermittently operated bituminous coal stokers, care should be taken that the delivery of air by the fan is great enough to avoid smoke

Controls

The industry developed by stokers in Classes 1, 2, 3 and 4 has been due as much to the application of proper controls as to the stoker itself. This is especially true of Class 1 stokers because of their application to residential heating, a field wherein the majority of owners and users are not familiar with control problems or stoker operation.

The usual controls applied are as follows:

a. Thermostats (plain and clock)
b Limit Controls (steam, vapor, vacuum, hot water, or air)
c. Stack Temperature or Time Controls (for actuating fires periodically)
d. Relay (for low voltage controls).
e. Safety or Overload Cutout (for protection against overload)
f Low Water Cutout (steam, vapor and vacuum)

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically and safely from liquid fuels. This heat is produced in the furnace or firepot of hot water or steam boilers or warm air furnaces and is absorbed by the boiler, and thus made available for distribution to the house through the heating system.

The number of combinations of the characteristic elements of domestic oil burners is rather large and accounts for the variety of burners found in actual practice. Domestic oil burners may be classified as follows:

1. AIR SUPPLY FOR COMBUSTION
   a. Atmospheric—by natural chimney draft
   b. Mechanical—electric-motor-driven fan or blower.
   c. Combination of (a) and (b)—primary air supply by fan or blower and secondary air supply by natural chimney draft.

2. METHOD OF OIL PREPARATION
   a. *Vaporizing*—oil distills on hot surface or in hot cracking chamber.
   b. *Atomizing*—oil broken up into minute globules.
      (1) Centrifugal—by means of rotating cup or disc.
      (2) Pressure—by means of forcing oil under pressure through a small nozzle or orifice.
      (3) Air or steam—by high velocity air or steam jet in a special type of nozzle.
      (4) Combination air and pressure—by air entrained with oil under pressure and forced through a nozzle.
   c. *Combination of (a) and (b).*

3. TYPE OF FLAME
   a. *Luminous*—a relatively bright flame. An orange-colored flame is usually best if no smoke is present.
   b. *Non-luminous*—Bunsen-type flame (*i.e.*, blue flame).

4. METHODS OF IGNITION
   a. *Electric.*
      (1) Spark—by transformer producing high-voltage sparks. Usually shielded to avoid radio interference. May take place continuously while the burner is operating or just at the beginning of operation
      (2) Resistance—by means of hot wires or plates.
   b. *Gas.*
      (1) Continuous—pilot light of constant size.
      (2) Expanding—size of pilot light expanded temporarily at the beginning of burner operation.
   c. *Combination*—electric sparks light the gas and the gas flame ignites the oil.
   d. *Manual*—by manually-operated gas torch for continuously operating burners.

5. MANNER OF OPERATION
   a. *On and off*—burner operates only a portion of the time (intermittent).
   b. *High and low*—burner operates continuously but varies from a high to a low flame.
   c. *Graduated*—burner operates continuously but flame is graduated according to needs by regulating both air and oil supply.

A trade classification of oil burners consists of the following general types: (*a*) gun or pressure atomizing, (*b*) rotary and (*c*) pot or vaporizing.

The gun type, illustrated in Fig. 10 is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a spray of atomized oil is introduced and mixed in the combustion chamber with the air stream emerging from the air tube. A variety of patented shapes are employed at the end of the air tube to influence the direction and speed of the air and thus the effectiveness of the mixing process.

The most distinguishing feature of vertical rotary burners is the principle of flame application. These burners are of two general types; the center flame and wall flame. In the former type, (Fig. 11) the oil is atomized by being thrown from the rim of a revolving disc or cup and the flame burns in suspension with a characteristic yellow color. Combustion is supported by means of a bowl-shaped chamber or hearth. The wall flame burner (Fig. 12) differs in that combustion takes place in a ring of

refractory material, which is placed around the hearth. These types of burners are further characterized by their installation within the ashpit of the boiler or furnace.

The pot type burner (Fig. 13) can be identified by the presence of a metal structure, called a pot or retort, in which combustion takes place.

When gun type (pressure atomizing) or horizontal rotary burners are

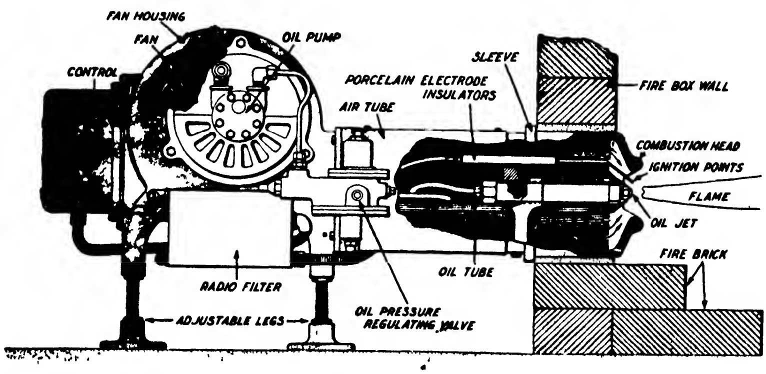

FIG. 10. GUN TYPE PRESSURE ATOMIZING OIL BURNER

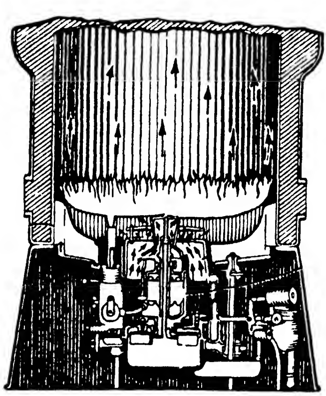

FIG. 11. CENTER FLAME VERTICAL ROTARY BURNER

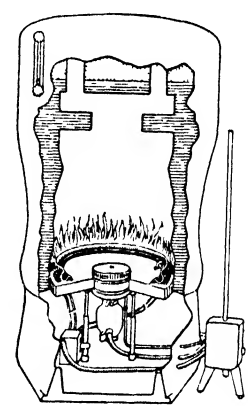

FIG. 12. WALL FLAME VERTICAL ROTARY BURNER

used the combustion chamber is usually constructed of firebrick or other suitable refractory material, and is part of the installation procedure.

The oil burners are operated by a small electric motor which pumps the oil and some or all of the air required. The smallest sizes can generally burn not much less than 1 gal of oil per hour. The grade of oil burned ranges from No. 1 to No. 3. No. 3 oil is the heaviest and most viscous of the various grades mentioned. An oil burner satisfactory for No. 3 oil can burn any of the lighter grades easily but an oil burner recommended for No. 2 oil should never be supplied with the heavier grades. It has been

found that while the heavier grades of oil have a smaller heat value per pound, they have, due to greater density, a larger heat value per gallon. The relative economy of the various grades must be based upon price and the amount of excess air required for clean and efficient combustion.

Boiler-Burner Units

Boilers and air conditioners especially designed for oil burners are available to the purchaser of this type of equipment. They are used for replacements as well as for new installations. This type of equipment usually has more heating surface than the older coal-burning designs. Flue proportions and gas travel have been changed with beneficial results.

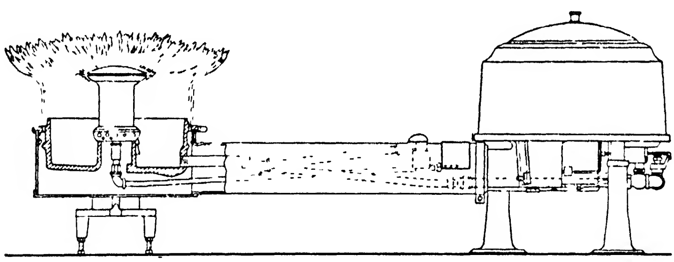

FIG. 13. POT TYPE VAPORIZING BURNER

All problems of combustion chamber design, capacities, efficiencies, etc., have been solved. The selection of the proper size of unit should be a simple process.

COMMERCIAL OIL BURNERS

Liquid fuels are used for heating apartment buildings, hotels, public and office buildings, schools, churches, hospitals, department stores, as well as industrial plants of all kinds. Contrary to domestic heating, convenience seldom is a dominating factor, the actual net cost of heat production usually controlling the selection of fuel. Some of the largest office buildings have been using oil for many years. Many department stores have found that floor space in basements and sub-basements can be used to better advantage for merchandising wares, and credit the heat producing department with this saving.

Wherever possible, the boiler plant should be so arranged that either oil or solid fuel can be used at will, permitting the management to take advantage of changes in fuel costs if any occur. Each case should be considered solely in the light of local conditions and prices.

Burners for commercial heating may be either large models of types used in domestic heating, or special types developed to meet the conditions imposed by the boilers involved. Generally speaking, such burners are of the mechanical or pressure atomizing types, the former using rotating cups producing a horizontal torch-like flame. (Fig. 14). As much as 350 gal of oil per hour can be burned in these units, and frequently they

are arranged in multiple on the boiler face, from two to five burners to each boiler.

The larger installations are nearly always started with a hand torch, and are manually controlled, but the use of automatic control is increasing, and completely automatic burners are now available to burn the two heaviest grades of oil. Nearly all of the smaller installations, in schools, churches, apartment houses and the like, are fully automatic.

Because of the viscosity of the heavier oils, it is customary to heat them before transferring by truck tank. It also has been common practice to preheat the oil between the storage tank and the burner, as an aid to movement of the oil as well as to atomization. This heating is accomplished by heat-transfer coils, using water or steam from the heating boiler, and heating the oil to within 30 deg of its flash point.

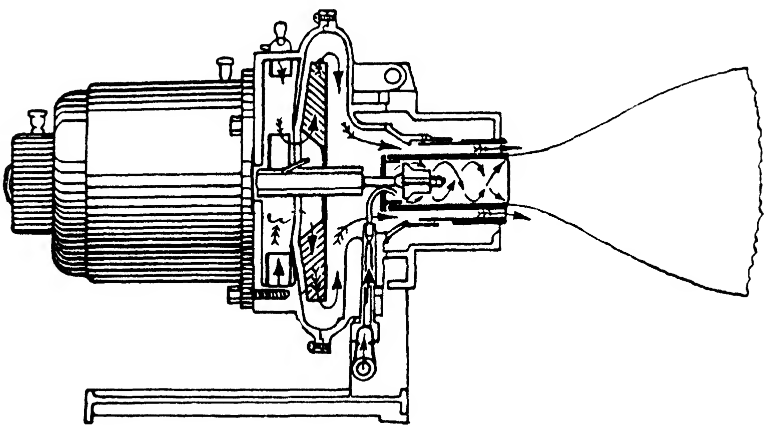

FIG 14 HORIZONTAL ROTATING-CUP OIL BURNER

Unlike the domestic burner, units for large commercial applications frequently consist of atomizing nozzles or cups mounted on the boiler front with the necessary air regulators, the pumps for handling the oil and the blowers for air supply being mounted in sets adjacent to the boilers. In such cases, one pump set can serve several burner units, and common prudence dictates the installation of spare or reserve pump sets. Pre-heaters and other essential auxiliary equipment also should be installed in duplicate.

Boiler Settings

As the volume of space available for combustion is the determining factor in oil consumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ashpit volume; in new installations the boiler should be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2 lb of oil can properly be burned. This corresponds to a maximum liberation of about 38,000 Btu per cubic foot per hour. There are indications that at times much higher fuel rates may be

satisfactory. This in turn suggests that the value of 38,000 Btu per cubic foot per hour might be adjusted according to good engineering judgment. For best results, care should be taken to keep the gas velocity below 40 ft per second. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or fire brick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

The Combustion Process

Efficient combustion, as previously indicated, must produce a clean flame and must use relatively small excess of air, i.e., between 25 and 50 per cent. This can be done only by vaporizing the oil quickly, completely, and mixing it vigorously with air in a combustion chamber hot enough to support the combustion. A vaporizing burner prepares the oil, for combustion, by transforming the liquid fuel to the gaseous state through the application of heat. This is accomplished before the oil vapor mixes with air to any extent and if the air and oil vapor temperatures are high and the fire pot hot, a clear blue flame is produced. There may be a deficiency of air as shown by the presence of carbon monoxide (CO) or an excessive supply of air, depending upon burner adjustment, without altering the clean, blue appearance of the flame.

An atomizing burner i.e., gun and rotary types is so named because the oil is mechanically separated into very fine particles so that the surface exposure of the liquid to the radiant heat of the combustion chamber is vastly increased and vaporization proceeds quickly. The result of such practice is the ability to burn more and heavier oil within a given combustion space or furnace volume. Since the air enters the fire pot with the liquid fuel particles, it follows that mixing, vaporization and burning are all occurring at once in the same space. This produces a luminous instead of a blue or non-luminous flame. In this case a deficient amount of air is indicated by a dull red or dark orange flame with smoky flame tips.

An excessive supply of air may produce a brilliant white flame in some cases or, in others, a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be easily detected, it is generally not possible to distinguish, by the eye alone, the finer adjustment which competent installation requires.

Certain tests indicate that there is no difference in economy between a blue flame and a luminous flame if the position, shape and the per cent of excess air of both flames are about the same.

Furnace or Combustion Chamber Design

The furnace or combustion chamber may be defined as that part of a boiler or conditioner in which combustion is established. With burners requiring a refractory combustion chamber the size and shape should be in accordance with the manufacturer's instructions. It is important that

the chamber shall be as nearly air tight as is possible, except when the particular burner requires a secondary supply of air for combustion.

It is evident that the atomizing burner is dependent upon the surrounding heated refractory or fire brick surfaces to vaporize the oil and support combustion. While the importance of the combustion chamber is obvious, its design has been troublesome. Unsatisfactory combustion may be due to inadequate atomization and mixing. A combustion chamber can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the fire brick surface, a carbon deposit will result. Fundamentally, the combustion chamber should enclose a space having a shape similar to the flame but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the combustion chamber flat but far enough below the nozzle to avoid flame contact, the sides tapering from the air tube at the same angle as the nozzle spray and the back wall rounded. A plan view of the combustion chamber thus resembles in shape the outline of the flame. In this way as much fire brick as possible is close to the flame so it may be kept quite hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead or inactive spaces in the combustion chamber. An overhanging arch at the back of the fire pot is sometimes used to increase the flame travel and give more time for mixing and burning and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, combustion chamber design becomes a less critical matter. Where secondary air is used, combustion chamber design is quite important. With some of the vertical rotary burners considerable care must be exercised in definitely following the manufacturers instructions when installing the hearth as in this class successful performance depends upon this factor.

Combustion Adjustments

Where adjustments of oil and air have been made which give efficient combustion, the problem of maintaining the adjustments constant becomes an important one. Particularly is this true when the change causes the per cent of excess air to decrease below allowable limits of the burner. A decrease in air supply while the oil delivery remains constant or an increase in oil delivery while the air supply remains constant will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment (*i.e.*, 25 per cent excess air) the more critical it will be of variations. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (*a*) changes in oil viscosity due to temperature change or variations in grade of oil delivered, (*b*) erosion of atomizing nozzle, (*c*) fluctuations in by-pass relief pressures and (*d*) possible variations in methods 2*b* (3) and 2*b* (4) listed in the previous classification table. Note that any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy and possibly a complete interruption of service but usually no soot will form.

The following factors may influence the air supply: (*a*) changes in combustion draft due to a variety of causes (*i.e.*, changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other

purposes, possible stoppage of the chimney and changes in draft resistance of boiler due to partial stoppage of the flues), and (*b*) changes in air inlet adjustments to the fan.

It is recognized that a secondary source of air due to leakage in the boiler setting is present in many installations and it is highly desirable that this leakage be reduced to a minimum. Obviously the amount of air leakage will be determined by the draft in the combustion chamber. It is important that this draft should be reduced as low as is consistent with the proper disposal of the gases of combustion. When using mechanical draft burners with average conditions, the combustion chamber draft should not be allowed to exceed 0.02-0.05 in. water. An automatic draft regulator is very helpful in maintaining such values.

Measurement of the Efficiency of Combustion

Efficient combustion being based upon a clean flame and certain proportions of oil and air employed, it is possible to determine the results by analyzing the gases formed by the combustion process. An Orsat apparatus is a device which measures the volume of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. Except in the case of a non-luminous flame it is usually sufficient to analyze only for carbon dioxide (CO_2). A showing of 10 to 12 per cent indicates the best adjustment if the flame is clean. Most of the good installations at the present time show from 8 to 10 per cent CO_2. Taking into account the potential hazard of oil or air fluctuations with low excess air (high CO_2) a setting to give 10 per cent CO_2 constitutes a reasonable standard for most oil burners. This is particularly true of non-luminous flame burners which will not function properly with less than 10 per cent CO_2.

Controls

Oil burner controls may be divided into two parts: (*a*) devices to regulate burner operation so the desired house heating result may be obtained, and (*b*) devices for the safety and protection of the boiler and burner. For control devices generally consult Chapter 37. The room thermostat has recently been improved to provide more frequent burner operation and greater uniformity of room temperature. Class (*b*) controls comprises a device to shut off the burner if the oil fails to ignite or if the flame should cease due to lack of oil; a device actuated by steam boiler pressure to shut off the burner when the pressure reaches some predetermined value; a device on the boiler to shut off the burner if the water level acts too low for safety or one which automatically feeds additional water to the boiler; a device on warm air furnaces to shut off the burner if the air temperature gets too high; a valve in the oil supply line which automatically closes in the event of fire in or near the cellar; and a device to keep the temperature of the boiler water within certain limits when it is being used to heat domestic hot water. These devices are all tested and approved by the Underwriters' Laboratory of Chicago, Ill., before they are offered to the purchaser. The selection of class (*b*) devices is made by the oil burner manufacturer.

Fuel Oil Gages

To insure a constant supply of fuel oil and to check deliveries and consumption it is essential to have accurate means of readily determining

the quantity of oil in the storage tank. For this purpose various types of indicating or recording gages are used, the simplest forms being the glass level gage, and a float-and-dial arrangement having a graduated dial face indicating the proportion of the tank containing liquid. Other more accurate and dependable devices are designed to operate by hydraulic action or by hydrostatic impulse. These instruments may be attached to the tank, giving a direct reading of the liquid contents; or the instrument itself may be located at a convenient point remote from the tank and connected with the tank by pipe or tubing. The quantity readings may be in gallons of liquid, height of liquid level in feet and inches, or in other desired units of measurement.

GAS-FIRED APPLIANCES

The increased use of gas for house heating purposes has resulted in the production of such a large number of different types of gas heating systems and appliances that today there is probably a greater variety of them than there is for any other kind of fuel.

Gas-fired heating systems may be classified as follows:

- I. Gas-Designed Heating Systems.
  - A. Central Heating Plants.
    - 1. Steam, hot water, and vapor boilers.
    - 2. Warm air furnaces.
  - B. Unit Heating Systems.
    - 1. Warm air floor furnaces
    - 2. Industrial unit heaters.
    - 3. Space heaters.
    - 4. Garage heaters.
- II. Conversion Heating Systems.
  - A. Central Heating Plants.
    - 1. Steam, hot water and vapor boilers.
    - 2. Warm air basement furnaces.

These systems are supplied with either automatic or manual control. Central heating plants, for example, whether gas designed or conversion systems, may be equipped with room temperature control, push-button control, or manual control.

Gas-Fired Boilers

Information on gas-fired boilers will be found in Chapter 13. Either snap action or throttling control is available for gas boiler operation. This is especially advantageous in straight steam systems because steam pressures can be maintained at desired points, while at the same time complete cut-off of gas is possible when the thermostat calls for it.

Gas-Fired Warm Air Furnaces

Warm air furnaces are variously constructed of cast-iron, sheet metal and combinations of the two materials. If sheet metal is used, it must be of such a character that it will have the maximum resistance to the corrosive effect of the products of combustion. With some varieties of manufactured gases, this effect is quite pronounced. Warm air furnaces

are obtainable in sizes from those sufficient to heat the largest residence down to sizes applicable to a single room. The practice of installing a number of separate furnaces to heat individual rooms is peculiar to mild climates. Small furnaces, frequently controlled by electrical valves actuated by push-buttons in the room above, are often installed to heat rooms where heat may be desired for an hour or so each day. These furnaces are used also for heating groups of rooms in larger residences. In a system of this type each furnace should supply a group of rooms in which the heating requirements for each room in the group are similar as far as the period of heating and temperature to be maintained are concerned.

The same fundamental principle of design that is followed in the construction of boilers, that is, breaking the hot gas into fine streams so that all particles are brought as close as possible to the heating surface, is equally applicable to the design of warm air furnaces.

Codes for proportioning warm air heating plants, such as that formulated by the *National Warm Air Heating and Air Conditioning Association* are equally applicable to gas furnaces and coal furnaces. Recirculation should always be practiced with gas-fired warm air furnaces. It not only aids in heating, but is essential to economy. Where fans are used in connection with warm air furnaces for residence heating, it is well to have the control of the fan and of the gas so coordinated that there will be sufficient delay between the turning on of the gas and the starting of the fan to prevent blasts of cold air being blown into the heated rooms. An additional thermostat in the air duct easily may be arranged to accomplish this.

Floor Furnaces

Warm air floor furnaces are well adapted for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility as any number of rooms may be heated without heating the others. With the usual type the register is installed in the floor, the heating element and gas piping being suspended below. Air is taken downward between the two sheets of the double casing and discharged upward over the heating surfaces and into the room. The appliance is controlled from the room to be heated by means of a control lever located near the edge of the register. The handle of the control is removable as a precaution against accidental turning on or off of the gas to the furnace.

Space Heaters

Space heaters are generally used for auxiliary heating, but may be, and are in many cases, installed for furnishing heat to entire buildings. With the exception of wall heaters, they are portable, and can be easily removed and stored during the summer season. Although they should be connected with solid piping it is sometimes desirable to connect them with flexible gas tubing in which case a gas shut-off on the heater is not permitted, and only *A.G.A.* approved tubing should be used.

Parlor furnaces or *circulators* are usually constructed to resemble a cabinet radio. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base and passes up inside the jacket

around a drum or heating section, and out of the heater at or near the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation. The burner or burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass up around baffles within the heating element or drum, and out the flue at the back near the top. They are well adapted not only for residence room heating but also for stores and offices.

Radiant heaters make admirable auxiliary heating appliances to be used during the occasional cool days at the beginning and end of the heating season when heat is desired in some particular room for an hour or two. The radiant heater gives off a considerable portion of its heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. Some have sheet-iron bodies finished in enamel or brass while others have cast-iron or brass frames with heavy fire-clay bodies. An atmospheric burner is supported near the center of the base, usually by set screws at each end. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are supported on legs and are portable; however, there are also types which are encased in a jacket which fits into the wall with a grilled front, similar to the ordinary wall register. Others are encased in frames which fit into fireplaces.

Gas-fired steam and hot water radiators are popular types of room heating appliances. They provide a form of heating apparatus for intermittently heated spaces such as stores, small churches and some types of offices and apartments. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator connected to a basement boiler. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to the steam or hot water radiators. They are usually constructed of pressed steel or sheet metal hollow sections. The hot products of combustion circulate through the sections and are discharged out a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Garage heaters are usually similar in construction to the cabinet circulator space heaters, except that safety screens are provided over all openings into the combustion chamber to prevent any possibility of explosion from gasoline fumes or other gases which might be ignited by an open flame. They are usually provided with automatic room temperature controls and are well suited for heating either residence or commercial garages.

Conversion Burners

Residence heating with gas through the use of conversion burners installed in coal-designed boilers and furnaces represents a common type of gas-fired house heating system. In many conversion burners radiants or refractories are employed to convert some of the energy in the gas to radiant heat. Others are of the blast type, operating without refractories.

Many conversion units are equipped with sheet metal secondary air

ducts which are inserted through the ashpit door. The duct is equipped with automatic air controls which open when the burners are operating and close when the gas supply is turned off. This prevents a large part of the circulation of cold air through the combustion space of the appliance when not in operation. By means of this duct the air necessary for proper combustion is supplied directly to the burner, thereby making it possible to reduce the amount of excess air passing through the combustion chamber.

Conversion units are made in many sizes both round and rectangular to fit different types and makes of boilers and furnaces. They may be secured with manual, push-button, or room temperature control.

The Combustion Process

Because of the varying composition of gases used for domestic heating it is difficult to generalize on the subject of gas burner combustion. Refer to the section on Gas Classification, in Chapter 9.

Combustion Adjustments

Little difficulty should be experienced in maintaining efficient combustion conditions when burning gas. The fuel supply is normally held to close limits of variation in pressure and calorific value and, therefore, the rate of heat supply is nominally constant. Since the force necessary to introduce the fuel into the combustion chamber is an inherent factor of the fuel, no draft by the chimney is required for this purpose. The use of a draft diverter insures the maintenance of constant low draft condition in the combustion chamber with a resultant stability of air supply. A draft diverter is also helpful in controlling the amount of excess air and preventing back drafts which might extinguish the flame.

Measurement of the Efficiency of Combustion

It is possible to determine the results of combustion by analyzing the gases of combustion with an Orsat apparatus. It is desirable to determine the percentage of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. While ultimate CO_2 values of 10 to 12 per cent may be obtained from the combustion of gases commonly used for domestic heating, a combustion adjustment which will show from 8 to 10 per cent CO_2 represents a practical value. Under normal conditions no CO will be produced by a gas-fired boiler or furnace. Limitations as to output rating by the *A.G.A.* are based upon operation with not more than 0.04 per cent CO in the products of combustion. This is too small an amount to be determined by the ordinary flue gas analyzer.

Controls

Gas burner controls may be divided into two parts: (*a*) devices to regulate burner operation so that the desired house heating results may be obtained, and (*b*) devices for the safety of the boiler and burner. Control devices are treated in detail in Chapter 37. A room thermostat may be used as a control of house heating effect. These may be obtained in a number of types. Some central heating plants are equipped with push-button or manual control. Class (*b*) controls include a device to shut off the burner if the gas fails to ignite, a device actuated by boiler pressure,

water temperature, or furnace bonnet temperature to shut off the burner when the pressure or temperature becomes excessive, a device on the boiler to shut off the burner if the water level falls below safe limits or one which automatically feeds additional water to the boiler, and a device for controlling the gas pressure within desired limits. The main gas valve may be either of the snap action or throttling type.

Sizing Gas-Fired Heating Plants

While gas-burning equipment can be and usually is so installed as to be completely automatic, maintaining the temperature of rooms at a predetermined and set figure, there are in use installations which are manually controlled. Experience has shown that, in order to effectively overcome the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard cast-iron column radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up or starting loads. Consequently, it is possible to use a much lower selection, or safety factor. A gas-fired boiler under thermostatic control is sensitive to variations in room temperatures so that in most cases a factor of 20 per cent is sufficient for pick-up load.

The factor to be allowed for loss of heat from piping, however, must vary somewhat, the proportionate amount of piping installed being considerably greater for small installations than for large ones. Consequently a selection factor for thermostatically controlled boilers must be variable. Liberal selection factors to be added to the installed steam radiation under thermostatic control are given in Fig. 3 of Chapter 13.

Appliances used for heating with gas should bear the approval seal of the *American Gas Association* Testing Laboratory. Installations should be made in accordance with the recommendations shown in the publications of that association.

Ratings for Gas Appliances

Since a gas appliance has a heat-generating capacity that can be predicted accurately to within 1 or 2 per cent, and since this capacity is not affected by such things as condition of fuel bed and soot accumulation, makers of these appliances have an opportunity to rate their product in exact terms. Consequently all makers give their product an hourly Btu output rating. This is the amount of heat that is available at the outlet of a boiler in the form of steam or hot water, or at the bonnet of the furnace in the form of warm air. The output rating is in turn based upon the Btu input rating which has been approved by the *American Gas Association* Testing Laboratory and upon an average efficiency which has been assigned by that association.

In the case of boilers, the rating can be put in terms of square feet of equivalent direct radiation by dividing it by 240 for steam, and 150 for water. This gives what is called the *American Gas Association* rating, and is the manner in which all appliances approved by the *American Gas Association* Laboratory are rated. To use these ratings it is only necessary to increase the calculated heat loss or the equivalent direct radiation load by an appropriate amount for starting and piping, and to select the boiler or furnace with the proper rating.

The rating given by the *American Gas Association* Laboratory is not only a conservative rating when considered from the standpoint of capacity and efficiency, but is also a safe rating when considered from the standpoint of physical safety to the owner or caretaker. The rating that is placed upon an appliance is limited by the amount of gas that can be burned without the production of harmful amounts of carbon monoxide. This same limitation applies to all classes of gas-consuming heating appliances that are tested and approved by the Laboratory. Gas boilers are available with ratings up to 14,000 sq ft of steam, while furnaces with ratings up to about 500,000 Btu per hour are available. (See Chapter 20.)

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PROBLEMS IN PRACTICE

1 ● List some factors which might account for higher efficiencies with stoker firing than with hand firing.

a The uniform rate of coal feed, *b*. Better distribution in the fuel bed, and *c*. Positive control of the air supplied for combustion

2 ● Classify stokers as to construction and operation.

a. Overfeed flat grate, *b*. Overfeed inclined grate, *c*. Underfeed side cleaning type, and *d*. Underfeed rear cleaning type.

3 ● What main parts are found in an underfeed residential stoker?

A *hopper* is supplied to hold coal which is fed by a *screw* or *plunger* into a *retort* provided with air openings called *tuyeres*. A *blower* supplies air under pressure for combustion, and a *gear case* provides for changes in coal feeding rates.

4 ● What is a dead-plate?

A dead-plate is a flat surface without air supply openings upon which the fuel rests while combustion of the fixed carbon is completed Generally the ash is removed from the dead-plate.

5 ● What features of furnace design are essential for the proper burning of the volatile coal gases above the fuel bed?

Adequate provisions should be made so that the furnace volume is sufficiently liberal and that the grates are a sufficient distance from the heating surfaces to permit the proper combustion of gases.

6 ● What methods of oil atomization are used?

a. Throwing the oil from a rotating cup or disc, *b*. Forcing the oil under high pressure through a nozzle, *c*. Propelling the oil with a high velocity jet of air or steam, and *d*. Forcing an oil and air mixture through a nozzle.

7 ● Is the furnace of much importance in oil burning?

In most cases it is very important. It is the function of the oil burner to supply the air and fuel in correct proportions; the furnace must provide heated space for proper mixing and combustion.

8 ● Which flame is considered better, the luminous or the non-luminous?

Laboratory tests show that they are equally efficient in the usual installation.

9 ● What CO_2 content should be attained in oil burning?

Ten per cent CO_2 is considered good practice, for it indicates the supplying of 50 per cent excess air

10 ● Name five types of gas-fired space heaters.

a. Parlor furnaces or circulators, *b*. radiant heaters, *c*. gas-fired steam or hot water radiators, *d*. warm air radiators, and *e*. garage heaters.

11 ● How are gas heating units rated?

Gas-fired units are rated on the basis of output in Btu per hour.

12 ● What safety consideration is noted in establishing the ratings of gas-fired units?

The rating is limited by the amount of gas that can be burned without the liberation of harmful amounts of carbon monoxide.

Chapter 12

HEAT AND FUEL UTILIZATION

Fuel Consumption Records, Calculated Heat Loss Estimation Method, Maximum Rate of Fuel Burning, Degree-Day Method, Unit Fuel Consumptions per Degree-Day, Maximum Demands and Load Factors

MANY methods are in use for estimating in advance of actual operation the anticipated heat or fuel consumption of heating plants over long or short periods. With suitable modification in procedure these same general methods are frequently useful in checking the degree of effectiveness with which heat or fuel is utilized during plant operation.

In applying any of these estimating methods to the consumption of a particular building plant it should be noted that (a) reliable records of past heat or fuel consumptions of *this* building will usually produce more trustworthy estimates of future consumptions than will any data obtained by averages or from other similar buildings; (b) where no past records exist useful data can sometimes be obtained from records of similar buildings with similar plants *in the same locality;* (c) records of consumption which are averages from many types of plants in many types of buildings in various localities, can produce no better than an average estimate which may be far from accurate; (d) estimates based on computed heat losses without the benefit of operating data are wholly dependent on how well the computation represents the actual facts.

Where records of past consumptions are available they should be examined for reliability to be sure that the records show fuel or heat for the heating plant only, or else make a suitable allowance for fuel used for other purposes, such as heating service water. Weights and measures shown on invoices may not always agree with fuel used, for residues left in bins or tanks may represent a considerable fraction of the fuel charged to a building. Generally, plant operating records of fuel used are to be preferred to those obtained from accounting or bookkeeping offices from fuel invoices.

Records from similar buildings even in the same locality should be examined with care before being used as the basis of estimates. The type of heating system, the quality of supervision in manual plants, the kind of control in automatic plants, and the attention given to the plant operation are all factors in fixing the consumption in any building. Many times these factors do not show up in superficial examination and are even difficult to evaluate when known to be present. Especially check the

records to be sure that they do not include energy or fuel used for other purposes than heating the building.

Estimates based on computed heat losses alone are frequently the only ones possible to obtain especially where new equipment is put into unusual buildings and there is a scarcity of records and an absence of experience data. Such estimates also have to be made where direct information is not obtainable as, for example, if a survey is being made without the assistance or knowledge of the building operator and thus without information as to the actual consumption. Estimates of this kind are also useful in some cases where a *relative* standard of performance is desired to serve as a base of comparisons in a campaign of fuel utili-

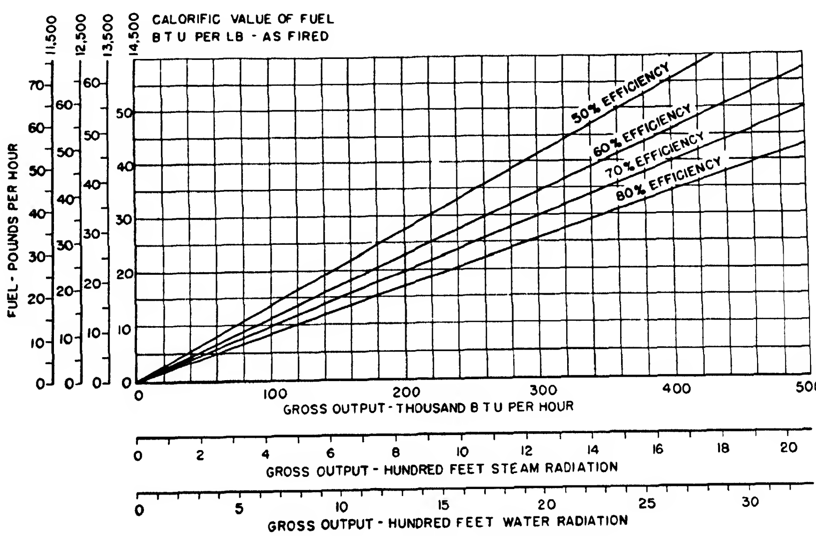

FIG. 1. COAL FUEL BURNING RATE CHART

zation. In such situations it can be plausibly argued that an estimate based on computed heat quantities is to be preferred to one which is related to operating methods.

In interpreting and evaluating heat or fuel consumption estimates as well as in their preparation, it is well to realize that any estimating method used will produce a more reliable result over a long period operation than over a short period. Nearly all of the methods in common use will give trustworthy results over a full annual heating season, and in some cases such estimates will prove consistent within themselves for monthly periods. As the period of the estimate is shortened there is more chance that some factor not allowed for in the estimating method will become controlling and thus give discrepant and even ridiculous results.

Of the various estimating methods in use attention is directed in this discussion to but two as they are illustrative of all, viz: (1) calculated heat loss method, and (2) degree-day method.

CALCULATED HEAT LOSS METHOD

This method is theoretical and assumes constant temperatures for very definite hours each day throughout the entire heating season. It does not take into account factors which are difficult to evaluate such as opening of windows, abnormal heating of the building, sun effect, poor heating systems, and many others.

In order to apply this method the hourly heat loss from the building under maximum load, or design condition is computed following the

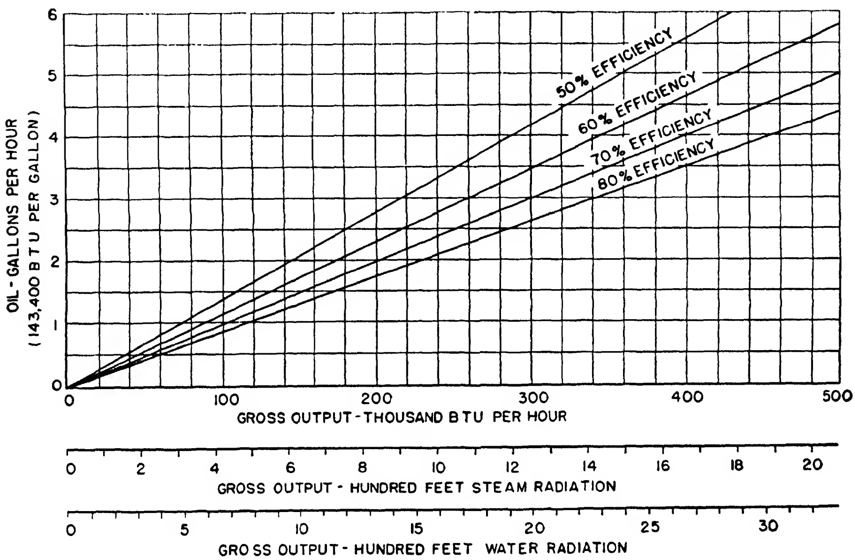

FIG 2. OIL FUEL BURNING RATE CHART[a]

[a]This chart is based upon No 3 oil having a heat content of 143 400 Btu per gallon If other grades of oil are used multiply the value obtained from this chart by the following factors No 1 oil (139,000 Btu per gallon) 1 032, No 2 oil (141 000 Btu per gallon) 1 017, No 4 oil (144 500 Btu per gallon) 0 992, No 5 oil (146,000 Btu per gallon) 0 982, and No 6 oil (150,000 Btu per gallon) 0 956

principles discussed in Chapters 5 and 6 and the method described and illustrated in Chapter 7.

In some cases, however, depending on the presence of interior partitions, the computed heat loss is modified when used for estimating the heat or fuel consumption. If the building has no interior walls or partitions then, by the method of Chapters 6 and 7, the infiltration losses are calculated by using only half the total window crack. In such a building the calculated loss need not be modified in order to prepare heat or fuel estimates by this method. Where the building does contain interior walls or partitions instead of using as the calculated heat loss (H) which is equal to the sum of the transmission losses (H_t) and the infil ration losses (H_i), it is more desirable to let $H = H_t + \frac{H_i}{2}$.

In predicting fuel consumption for building heating by the Calculated Heat Loss Method, the general formula is:

$$F = \frac{H\,(t - t_a)\,N}{E\,(t_d - t_o)\,C} \tag{1}$$

where

F = quantity of fuel or energy required (in the units in which C is expressed).

H = calculated heat loss, Btu per hour, during the design hour, based on t_o and t_d (generally $H = H_t + H_l$ but may on occasion equal $H_t + \frac{H_l}{2}$).

t = average inside temperature maintained over estimate period, degrees Fahrenheit.

t_a = average outside temperature through estimate period, degrees Fahrenheit (Table 2, Chapter 7).

t_d = inside design temperature, degrees Fahrenheit (usually 70 F).

t_o = outside design temperature, degrees Fahrenheit (see Outside Temperatures, Chapter 7).

N = number of heating hours in estimate period (for an Oct 1—May 1 heating season, 5088).

E = efficiency of utilization of the fuel over the period, expressed as a decimal; not the efficiency at peak or rated load condition

C = heating value of one unit of fuel or energy.

Example 1. A residence in Philadelphia is to be heated to 70 F from 6 A M to 10 P M and 55 F from 10 P M. to 6 A M The calculated hourly heat loss is 120,000 Btu per hour based on 70 F inside at −5 F outside If the building is to be heated by metered steam, how many pounds would be required during an average heating season?

Solution The heating value of steam may be taken at 1000 Btu per pound, and since it is purchased steam, the efficiency can be assumed as 100 per cent From Table 2, Chapter 7, t_a = 42 7 F. The average inside temperature is

$$\frac{(16 \times 70) + (8 \times 55)}{24} = 65 \text{ F}$$

Substituting in Equation 1:

$$F = \frac{120{,}000\ (65 - 42\ 7)\ 5088}{1\ 00\ [70 - (-5)]\ 1000} = 181{,}239 \text{ lb}$$

Example 2. How much would the fuel cost to heat the building in Example 1 during an average heating season with coal at \$8 per ton and with a calorific value of 11,000 Btu per pound, assuming that the seasonal efficiency of the plant was 55 per cent?

Solution. $F = \frac{120{,}000\ (65 - 42\ 7)\ 5088}{0.55\ [70 - (-5)]\ 11{,}000} = 30{,}013 \text{ lb} = 15 \text{ ton}$, which, at \$8 per ton, costs \$120.

Equation 1 can be expressed as:

$$F = \frac{H}{EC} \times \frac{(t - t_a)\ N}{(t_d - t_o)} \tag{2}$$

where the expression $\frac{H}{EC}$ is the rate at which fuel is burned during the design hour. Values of this rate are plotted as ordinates in Figs. 1, 2 and 3 for coal, oil and gas. For a given efficiency, the rate of fuel burning is directly proportional to the load and therefore these charts can be extended by moving the decimal points the same number of digits in both vertical and horizontal scales. Use of these charts thus expedites the estimate.

TABLE 1. HEAT CARRYING CAPACITY OF GRAVITY WARM AIR FURANCE ROUND LEADER PIPES

180 F Register Temperature

LEADER PIPE	BTU PER HR AT DESIGN CONDITIONS PER SQ IN OF LEADER PIPE
First floor	111
Second floor	167
Third floor	200

The charts are plotted so that the load is expressed in three terms (a) hourly heat loss at design conditions, (b) square feet of steam radiatoı surface (240 Btu per hour), and (c) square feet of hot water radiatoı surface (150 Btu per hour). By entering the chart at the correct point or the abcissa corresponding to the calculated heat loss (H), following vertically to the seasonal efficiency assumed and thence horizontally to the

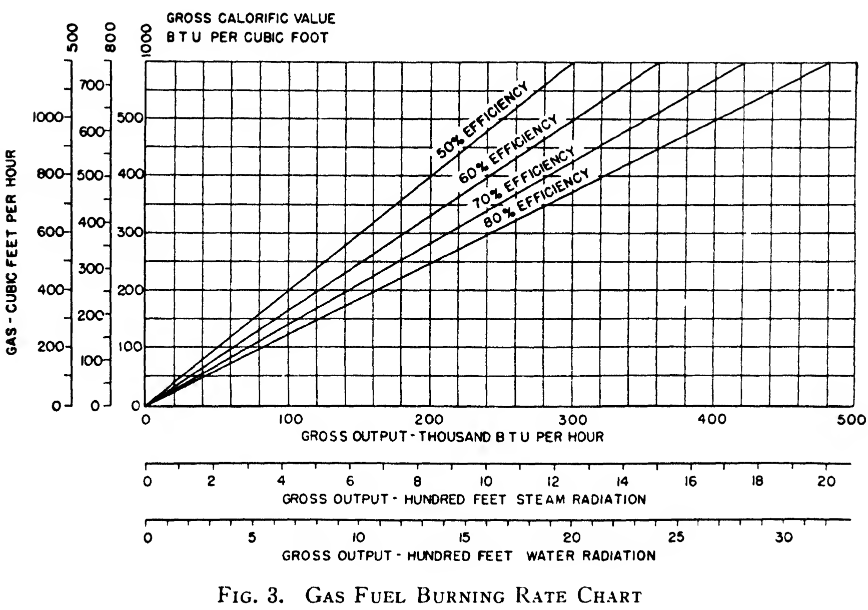

FIG. 3. GAS FUEL BURNING RATE CHART

fuel rate, the rate of fuel burning during a maximum or design hour will be found along the left hand scale for various calorific values of the fuel.

In the case of gravity warm air heating installations, the load is usually expressed in square inches of leader pipe. This can be converted into hourly heat loss by multiplying by the factors in Table 1.

Example 3. A building located in Salt Lake City with an oil-burning heating plant has a calculated hourly heat loss of 260,000 Btu per hour. The plant is designed to maintain a temperature of 70 F inside during all 24 hours of the day, the outside design temperature is −5 F, the average outside temperature 40 F, the heating season 5088 hours long, the assumed efficiency 60 per cent, and the oil has a calorific value of 143,400 Btu per gallon What will be the seasonal fuel consumption?

Solution Enter Fig 2 at 260,000 on the upper horizontal scale, move vertically to the 60 per cent efficiency curve, and horizontally to the vertical scale where the firing rate $\left(\frac{H}{EC}\right)$ is found as 3 0 gal per hour.

Substituting this in Equation 2 for $\left(\frac{H}{EC}\right)$:

$$F = 3\ 0 \times \frac{(70 - 40)\ 5088}{70 - (-5)} = 6106 \text{ gal.}$$

Example 4 What would be the total gas consumption over a full heating season of a gas-fired gravity warm air furnace designed according to the Code[1], and with four 12 in and two 8 in round leaders to the first floor and six 10 in leaders to the second floor, if the gas has a heating value of 500 Btu per cubic foot, the plant operates at a 70 per cent seasonal efficiency and is designed to maintain an average inside temperature of 65 F when it is 10 F outside in a city where the average outside temperature is 45 F and the heating season is 5088 hours long?

Solution The area of the round leaders is. 12 in , 113 sq in , 10 in , 79 sq in , and 8 in , 50 sq in From Table 1 the total Btu transmitted is:

First Floor.	[(4 × 113) + (2 × 50)] × 111	= 61,272 Btu per hour.
Second Floor	(6 × 79) × 167	= 79,158 Btu per hour
	Total	140,430 Btu per hour

Allowing 10 per cent for duct and furnace losses, the gross output would be 154,500 Btu per hour.

Enter Fig 3 at 154 5 on the upper horizontal scale, move to the 70 per cent efficiency curve and thence to the 500 Btu per cubic foot vertical scale and find $\left(\frac{H}{EC}\right)$ to be approximately 440 cu ft per hour

Substituting in Equation 2:

$$F = 440 \times \frac{(65 - 45)\ 5088}{(70 - 10)} = 746{,}428 \text{ cu ft.}$$

Maximum Rate of Fuel Burning

The rate at which fuel is burned during the maximum, or design hour is frequently useful in setting, or adjusting, the fuel feed devices attached to stokers, oil-burners, and gas burners. This rate is $\left(\frac{H}{EC}\right)$ and can be found from the charts of Figs. 1, 2 and 3 in the same way as outlined in the Examples 3 and 4. In using the charts for this purpose, however, it should be noted that the efficiency (E) is the overall efficiency of the boiler or furnace at the time of peak load This efficiency is generally considerably greater than the value selected for E when the seasonal efficiency of utilization is used in making seasonal fuel estimates. Failure to distinguish between the two essentially different meanings attached to E may result in grossly inaccurate estimates.

The correct fuel burning rate can be determined directly from the several charts for oil or gas burning installations, as these customarily operate on a strictly intermittent basis. These fuel burning devices

[1]Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences (9th edition), and the Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems, may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 50 W Broad St , Columbus, Ohio

usually introduce the fuel at a single fixed rate during the *on* periods and this rate should be sufficient to carry the gross or maximum design load. In the case of coal stokers, which are usually capable of variable rates of firing, it is desirable to operate at as low a rate as weather conditions will permit, but the maximum firing rate of the stoker should be sufficient to carry the gross load. This rate may be determined by the same method as used for oil or gas.

Example 5 The estimated net load (including domestic hot water supply) as calculated for a residence is 1500 sq ft of hot water radiation Determine the firing rates for various mechanically fired fuels assuming an overall boiler efficiency of 70 per cent, using coal with a calorific value of 12,500 Btu per pound, No 3 fuel oil and natural gas having a gross heating value of 1000 Btu per cubic foot

Solution Referring to Fig 3, Chapter 13, a piping and pick-up factor for a net load of 1500 sq ft is found to be 43 per cent or the gross output is equivalent to 1500 × 1 43 = 2145 sq ft hot water radiation.

Using the charts in Figs 1, 2 and 3 project vertically from the gross output value on the proper horizontal scale to the intersection of the 70 per cent efficiency line From the intersection of this line proceed horizontally to the proper vertical scale where a direct value of the required fuel burning rate is given These values are rates of burning while firing device is in operation and are not indicative of hourly fuel consumption

By use of the respective charts the firing rates for the various fuels will be found to be: coal 36 8 lb per hour, oil 3 2 gal per hour, and gas 460 cu ft per hour.

DEGREE-DAY METHOD

This method is based on consumption data which have been taken from buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the Calculated Heat Loss Method, it is of more value for practical use.

The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. Some years ago the *American Gas Association*[2] determined from experiment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the outside temperature. In other words, on a day when the temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F. The degree-day is defined in Chapter 45.

Recently the *National District Heating Association* studied the metered steam consumption of 163 buildings[3] in 22 different cities and published data substantiating the fact that the 65 F base originally chosen by the gas industry is approximately correct. (See Table 2).

If the degree-days occurring each day are totaled for a reasonably long period, the fuel consumption during that period as compared with another period will be in direct proportion to the number of degree-days in the two periods. Consequently, for a given installation, the fuel consumption can be calculated in terms of fuel used per degree-day for any sufficiently long period and compared with similar ratios for other periods to determine the relative operating efficiencies with the outside temperature variable eliminated.

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[2]See Industrial Gas Series, House Heating, (*third edition*) published by the *American Gas Association*
[3]These buildings are all served with steam from a district heating company

Predictions of fuel consumption are generally based on the average number of degree-days which have occurred over a long period of years, and such averages, by months, on a 65 F base, are given by months and heating season totals, for various United States and Canadian cities in Table 3. In general, attempts to apply the degree-day method to fuel consumptions over a period of less than a month are of questionable value.

Formula for Degree-Day Method

The general formula for predicting fuel consumption by the Degree-Day Method is:

$$F = U \times N \times D \tag{3}$$

where

F = fuel consumption for the estimate period.

U = unit fuel consumption, or quantity of fuel used per degree-day per building load unit.

N = number of building load units.

D = number of degree-days for the estimate period.

Values of D for use in Equation 3 are given in Table 3. Values of N depend on the particular building for which the estimate is being prepared and must be found by surveying plans, by observation, or by measurement of the building. Values of U for use in this equation are the Unit Fuel Consumptions per Degree-Day and are obtained as a result of the collection of operating information. Certain of this information is presented later but before referring to these data attention is directed to the nature of the unit.

TABLE 2. BASE TEMPERATURE FOR THE DEGREE-DAY[a]

Type of Building	No of Buildings Analyzed	Temperature F Corresponds to Zero Steam Consumption
Office	60	66.2
Office and Bank	4	65.8
Bank	3	66.2
Office and Telephone Exchange	2	65.5
Office and Stores	6	67.4
Stores	11	64.0
Department Stores	12	64.3
Hotels	7	66 5
Apartments	14	68.8
Residences	8	66.9
Clubs	4	65.5
Lodges	5	64.9
Theatres	3	67.6
Churches	2	65 8
Garages	2	64.8
Auto Sales and Service	4	61.2
Newspaper and Printing	3	67.7
Warehouse and Loft	3	67.7
Office and Loft	2	65.2
Manufacturing	8	65.4
Average for 163 Buildings		66.0 F

[a]Report of Commercial Relations Committee, *Proceedings, National District Heating Association*, 1932.

Unit Fuel Consumptions per Degree-Day

The quantity of fuel used per degree-day in a given heating plant car be reduced to a unit basis in terms of quantity of fuel (or steam) pei degree-day per square foot of radiator, per cubic foot of heated building space, or per thousand Btu hourly heat loss at design conditions. A less frequently used basis is quantity of fuel per degree-day per square foot ol air flow area. In fact any convenient unit can be used to relate the consumption to the degree-day and to the building.

The choice of these units requires explanation and some discrimination and judgment. The use of heated space in preference to the gross building cubage used by architects is obviously more accurate for this purpose. The architects' cubage includes the outer walls and certain percentages of attic and basement space which are usually unheated. The net heated space is usually about 80 per cent of the gross cubage and can be calculated from the latter if it cannot be measured. The cubical content is somewhat inaccurate as a basis of comparison due to differences in types of construction, exposure, and ratio of exposed area to cubical contents. Use of equivalent radiator surface figures is fundamentally the same as using a calculated heat loss and therefore units in terms of fuel per degree-day per equivalent square foot of calculated radiator surface, per 1000 Btu of calculated heat loss, or per Btu of heat loss at design conditions are all of equal accuracy and desirability. It is doubtful if installed radiator surface as determined by count should be used at all. Radiator units are also of questionable value where there is fan coil surface or warm air systems. In view of all these considerations it is believed that the unit based on *thousands of Btu of hourly calculated heat loss for the design hour* is probably the most desirable although the one most widely used seems to be *units of fuel (or heat) per degree-day per square foot of equivalent direct radiator surface.*

Since this unit is the one most widely used at present the unit fuel consumptions given in succeeding paragraphs of this chapter make use of this unit to a considerable extent, although it should be understood that most of these units of consumption can be transposed as desired.

Estimating Gas Consumption

Values of the Unit Fuel Consumption Constant (U) for gas are given in Table 4 for various gas heating values, and different types and sizes of heating plants. *They are based on an inside design temperature of 70 F and an outside design temperature of 0 F and apply only to these conditions.* For other design conditions corrections must be made as given in Table 5. Estimates for industrial buildings where low inside temperatures are maintained cannot be made from this table.

The factors in Table 4, as corrected if necessary, are satisfactory for regions having 3500 to 6500 degree-days per heating season. In regions with less than 3500 degree-days the unit gas consumption is higher than given; where over 6500, the unit is less than given. Ten per cent addition or deduction in these cases is recommended by *American Gas Association* publications.

For gas heating values other than those given in Table 4, simply interpolate or extrapolate. It will also be noted that Table 4 applies only to

TABLE 3. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a]

State	City	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec	Total
Ala.	Birmingham	589	521	260	69							318	595	2352
	Mobile	428	311	152								186	394	1471
Ariz	Flagstaff	1153	969	896	654	465	171		40	252	577	840	1128	7145
	Tucson	459	325	257	87							252	465	1845
Ark	Little Rock	719	582	353	78						47	381	651	2811
Calif	Los Angeles	326	266	239	159	90						123	301	1504
	San Francisco	465	356	353	294	264	195	202	186	114	146	261	428	3264
Colo	Col. Springs	1085	991	852	612	369	90			162	502	789	1066	6518
	Denver	1079	918	800	534	267				72	428	759	1017	5874
Conn	New Haven	1110	1011	899	543	223				39	360	693	1017	5895
D C	Washington	970	848	694	348	25					251	594	896	4626
Fla	Jacksonville	285	207	56								75	267	890
Ga	Atlanta	682	557	388	132						96	396	639	2890
	Savannah	409	316	167								201	397	1490
Idaho	Boise	1097	848	651	435	236				102	434	738	1011	5552
Ill	Chicago	1262	1095	911	549	248				3	353	756	1113	6290
	Springfield	1181	1008	760	366	56					282	681	1039	5373
Ind	Evansville	949	854	620	276						155	528	862	4244
	Indianapolis	1128	969	756	384	59					298	687	1017	5298
Iowa	Des Moines	1392	1173	890	441	118					357	798	1215	6384
	Sioux City	1435	1260	967	489	164				33	415	870	1265	6898
Kans	Dodge City	1116	890	688	342	47					276	672	1104	5035
	Topeka	1221	980	747	339						270	699	1051	5307
Ky	Lexington	973	868	648	342	25					245	612	905	4618
	Louisville	939	801	589	264						186	552	849	4180
La	New Orleans	332	230	59								102	301	1024
Me	Eastport	1380	1232	1110	786	543	300	143	136	276	543	843	1228	8520
	Portland	1321	1168	1017	642	329	39			120	443	780	1153	7012
Md	Baltimore	955	843	701	348	22					223	567	874	4533
Mass	Boston	1150	1042	908	570	245				48	363	693	1026	6045
Mich	Detroit	1252	1134	973	573	226				42	400	777	1113	6490
	Marquette	1500	1361	1249	804	496	186		43	225	567	960	1302	8693
Minn	Duluth	1727	1473	1277	810	524	198		37	261	620	1062	1491	9480
	Minneapolis	1609	1400	1094	570	236				93	481	963	1404	7850
Miss	Vicksburg	521	384	195								252	471	1823
Mo	Kansas City	1094	958	676	303	12					214	612	983	4852
	St Louis	1060	854	657	276						205	597	936	4585
Mont	Billings	1318	1120	955	534	316	60			189	524	909	1194	7119
	Havre	1624	1450	1169	630	369	144			270	620	1041	1383	8700
Nebr	Lincoln	1311	1131	840	417	99					316	753	1132	5999
	Omaha	1355	1126	868	414	84					329	780	1175	6131
Nev	Reno	1042	823	753	534	366	90			144	453	714	973	5892
N H	Concord	1349	1240	1011	669	298	54			168	484	846	1234	7353
N J	Atlantic City	992	904	806	519	220					254	588	893	5176
	Trenton	1014	941	735	402	81					242	588	930	4933
N M	Santa Fe	1110	902	775	543	298	3			120	459	780	1073	6063
N Y	Albany	1287	1142	980	549	183				72	446	774	1147	6580
	Buffalo	1240	1156	1032	675	335	12			75	419	774	1104	6822
	New York	1060	960	837	486	155					276	618	955	5347
	Utica	1248	1181	989	588	253				182	430	781	1144	6796
N. C.	Raleigh	722	630	446	183						130	429	694	3234
	Wilmington	555	468	322	108						19	303	527	2302
N D	Bismarck	1807	1548	1283	657	338	45			222	626	1113	1553	9192
Ohio	Cincinnati	1011	871	679	339	19					248	615	921	4703
	Cleveland	1181	1075	930	564	220				27	366	732	1060	6155
	Columbus	1113	980	778	420	87					313	690	1017	5398
Okla.	Oklahoma City	865	742	465	162						105	459	815	3613
Ore.	Portland	806	644	558	402	245	90			105	332	558	729	4469
	Salem	778	633	586	426	285	111			102	350	600	747	4618

TABLE 3. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a] (Concluded)

STATE	CITY	JAN	FEB	MAR	APR	MAY	JUNE	JULY	AUG	SEPT	OCT	NOV	DEC	
Pa.	Philadelphia	1001	893	756	402	68					242	588	905	[illegible]
	Pittsburgh	1054	944	787	423	78					313	669	967	[illegible]
R. I	Providence	1116	1070	890	558	251				63	348	693	1026	[illegible]
S. C.	Charleston	487	372	242	36							207	425	
	Spartanburg	725	622	431	147						121	429	716	3
S D.	Sioux Falls	1547	1358	1045	564	217				93	484	945	1404	7
Tenn	Memphis	744	599	384	96						62	402	663	2
	Nashville	812	675	477	180						136	483	744	3
Texas	Austin	487	330	133								201	434	1
	Dallas	617	493	267	9							303	567	2
	Houston	366	277	65								114	335	1
	San Antonio	381	274	74								126	347	1
Utah	Logan	1262	1072	893	525	329	48			114	468	819	1218	6
	Salt Lake City	1110	885	722	453	236				18	388	723	1020	5
Vt	Burlington	1429	1294	1088	654	273	3			144	481	861	1287	7
Va	Fredericksburg	887	820	583	303						223	549	877	4
	Norfolk	738	650	521	246						99	411	685	3
	Richmond	825	703	552	240						158	483	766	3
Wash	Seattle	775	652	623	465	319	168	40	43	192	403	570	716	4
	Spokane	1172	952	778	504	285	81			192	515	819	1057	6
W Va	Morgantown	1026	944	713	414	78					295	648	977	5
	Parkersburg	995	907	679	360	47					282	630	1048	4
Wis.	Fond du Lac	1507	1322	1048	603	276				117	493	921	1330	7
	Green Bay	1538	1358	1125	600	322				132	505	921	1324	7
	LaCrosse	1535	1265	1033	528	183				96	462	909	1311	7
	Milwaukee	1383	1201	1023	648	350	39			84	450	846	1221	7
Wyo	Cheyenne	1215	1075	995	720	446	126			240	605	900	1144	7

PROVINCE	CITY	JAN	FEB	MAR	APR	MAY	SEPT	OCT	NOV	DEC	TOTAL
B. C.	Victoria										5777
	Vancouver										5976
	Kamloops										6724
Alb.	Medicine Hat										8152
Sask.	Qu'Appelle										11,261
Man.	Winnipeg										11,166
Ont.	Port Arthur										10,803
	Toronto										7732
Que	Montreal	1615	1409	1219	720	309	190	372	961	1422	8417
	Quebec										8628
N. B.	Fredericton										9099
N. S.	Yarmouth										7694
P E. I	Charlottetown										8485

[a] Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C Strock and C H B Hotchkiss

small installations. In general the larger the installation the smaller the unit gas consumption becomes and the values in the table should be used with care, if at all, in large gas-burning installations.

Example 6 What would be the estimated average gas consumption of a residence in Albany, N Y, with 650 sq ft of steam radiation surface if the outside design temperature is −10 F, if 70 F is to be maintained inside and if 535 Btu per cubic foot gas is available?

Solution Find the value of U from Table 4 under, Steam, 300 to 700 sq ft, opposite 535 to be 0 215 Since the outside design temperature is other than 0 F, find the correction factor from Table 5 to be (opposite −10 F) 0 875 Multiplying 0 215 by 0.875

the corrected U is 0.188. The average number of degree-days in Albany, from Table 3, for a full average heating season is 6580, and the number of building load units is 650 Substituting in Equation 3:

$$F = 0.188 \times 6580 \times 650 = 804{,}076 \text{ cu ft of gas.}$$

Estimating Oil Consumption

Unit fuel consumption factors for oil, similar to those for gas in Table 4 are given in Table 6. *The factors in Table 6 apply only to an inside design temperature of 70 F and an outside design temperature of 0 F.* For other

TABLE 4. FACTORS FOR ESTIMATING GAS CONSUMPTION[a]

BTU VALUE OF GAS PER CU FT	HOT WATER			STEAM			WARM AIR	
	Cu Ft Gas per Degree-Day per Sq Ft Radiator			Cu Ft Gas per Degree-Day per Sq Ft Radiator			Cu Ft Gas per Degree-Day per 1000 Btu Hourly Design Heat Loss	
	Up to 500 Sq Ft	500 to 1200 Sq Ft	Over 1200 Sq Ft	Up to 500 Sq Ft	300 to 700 Sq Ft	Over 700 Sq Ft	Gravity	Fan Systems
500	0.142	0.135	0.128	0.242	0.231	0.220	0.855	0.820
535	0.132	0.126	0.120	0.226	0.215	0.206	0.800	0.766
800	0.089	0.085	0.081	0.151	0.144	0.137	0.534	0.513
1000	0.071	0.068	0.065	0.121	0.115	0.110	0.428	0.410
1 Therm = 100,000 Btu	Gas Consumption in Therms per Degree-Day							
	0.000708	0.000675	0.000642	0.00121	0.00115	0.00110	0.00428	0.00409

[a]Abstracted from Comfort Heating, *American Gas Association*, 1938

TABLE 5 CORRECTION FACTORS FOR OUTSIDE DESIGN TEMPERATURES

OUTSIDE DESIGN TEMP DEG F	INSIDE DESIGN TEMP DEG F	MULTIPLY VALUES IN TABLES 4, 6 AND 7 BY
−20	70	0.778
−10	70	0.875
0	70	0.000
+10	70	1.167
+20	70	1.400

outside design temperatures, the constants in Table 6 must be multiplied by the values in Table 5 as explained under Estimating Gas Consumption.

Table 6 assumes the use of oil with a heating value of 140,000 Btu per gallon. For other heating values, multiply the values in Table 6 by the ratio of 140,000 divided by the heating value per gallon of fuel being used.

Example 7. What would be the estimated seasonal oil consumption of a boiler-burner unit in Minneapolis of a building having a calculated heat loss of 192,000 Btu per hour, burning 144,000 Btu per gallon oil and operating at a seasonal efficiency of 60 per cent, if the outside design temperature for Minneapolis is −20 F, and the inside design temperature is 70 F?

Solution. From Table 6, under 60 per cent efficiency and opposite the bottom column, find the uncorrected U to be 0.00476 gal per 1000 Btu hourly heat loss.

From Table 5, the correction multiplier for −20 F outside design temperature is 0 778. Solving, 0 778 × 0 00476 = 0 00370. Making a further correction for the heating value:

$$0.0037 \times \frac{140{,}000}{144{,}000} = 0\ 0036 \text{ gal per 1000 Btu per hour calculated heat loss per degree-day.}$$

From Table 3, the average degree-days for Minneapolis number 7850, and from the problem $N = 192$. Substituting in Equation 3:

$$F = 0.0036 \times 7850 \times 192 = 5426 \text{ gal.}$$

TABLE 6. UNIT FUEL CONSUMPTION[a] CONSTANTS FOR OIL[b]

Unit	Efficiency in Per Cent				
	40	50	60	70	80
Gal Oil per Sq Ft Steam Radiator	0 00172	0 00137	0 00114	0 00098	0 00086
Gal Oil per Sq Ft Hot Water Radiator	0 00108	0 00086	0 00072	0 00062	0 00054
Gal Oil per 1000 Btu per Hour Heat Loss	0 00715	0 00571	0 00476	0 00409	0 00358

[a]Based on a heating value of 140,000 Btu per gallon

[b]Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C H B Hotchkiss

TABLE 7 UNIT FUEL CONSUMPTION[a] CONSTANTS FOR COAL[b]

Unit	Efficiency in Per Cent				
	40	50	60	70	80
Lb Coal per Sq Ft Steam Radiator	0 0200	0 0160	0 0133	0 0114	0 0100
Lb Coal per Sq Ft Hot Water Radiator	0 0125	0 0100	0 0084	0 0072	0 0063
Lb Coal per 1000 Btu per Hour Heat Loss	0 0825	0 0666	0 0550	0 0471	0 0412

[a]Based on a heating value of 12,000 Btu per pound.

[b]Abstracted by permission from *Degree-Day Handbook*, (Second Edition, 1937), by C Strock and C H B Hotchkiss

Estimating Coal or Coke Consumption

Coal or coke consumption estimates can be made in exactly the same way as for oil. The uncorrected values of U are given in Table 7. *These constants apply only to inside design temperatures of 70 F and an outside design temperature of 0 F*, and correction must be made for other conditions by use of the multiplying factors in Table 5. Table 7 is based on 12,000 Btu per pound coal and for other heating values of coal, values in Table 7 must be multiplied by the ratio of 12,000 divided by the heating value of fuel used.

Estimating Steam Consumption

In estimating steam consumption the efficiency is not ordinarily a factor and is assumed at 100 per cent. Ordinarily low pressure steam

TABLE 8. STEAM CONSUMPTION FOR VARIOUS CLASSES OF BUILDINGS[a]
(*Heating Season Only*)

Building Classification	No of Buildings Listed	Steam Consumption Pounds per Degree-Day—65 F Basis[d]		
		Per M Cu Ft of Heated Space	Per M Sq Ft of Radiator[c] Surface	Per M Btu per Hr of Heat Loss[b]
Apartments	16	1.78	97 5	0.359
Hotels	10	1.46	80.6	0.371
Residences	12	1 32	64.2	---- -----
Printing	7	1.25	105 5	----------
Clubs and Lodges	10	0 96	77.0	---- ----
Retail Stores	18	0.90	80 6	0 268
Theatres	6	0 90	75 0	0.498
Loft and Mfg	16	0.89	72.3	0 283
Banks	7	0 88	45.2	---- . .
Auto Sales and Service	8	0 83	62.2	---- . .
Churches	6	0.58	49 4	---- ---
Department Stores	14	0.57	60.7	0 238
Garages (Storage)[e]	6	0 42	72 3	---- ---
Offices (Total)	35	1 09	70.0	0 283
Offices (Heating only)	35	0.975	65.4	0 256

[a]Includes steam for heating domestic water for heating season only
[b]Heat loss calculated for maximum design condition (in most cases 70 F inside, zero outside)
[c]Equivalent steam radiator surface
[d]The figures are a numerical—not a weighted—average for the several buildings in each class
[e]Based on zero consumption at 55 F

with a heating value of 1000 Btu per pound is used so that no correction is necessary for heating value in the usual case. In comparing values from different cities, correction should be made for design temperature (see Table 5) when the unit figures are in terms of square foot of radiator or 1000 Btu per hour calculated heat loss, but not when the values are in terms of building volume or floor space.

Consideration has been given to the difference in steam utilization of different types of buildings and Table 8[4] shows actual average units for these various types. These figures are obtained from operating results in

TABLE 9 BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS[a]

Building Classification	Load Factor	Lb of Demand per Hr per Sq Ft of Equivalent Installed Radiator Surface
Clubs and Lodges	0.318	0 184
Hotels	0 316	0.207
Printing	0 287	0.217
Offices	0.263	0 209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0 248
Banks	0.203	0.158
Churches	0.158	0 152
Department Stores	0.138	0.145
Theatres	0.126	0.151

[a]Report of Commercial Relations Committee, *Proceedings, National District Heating Association*, 1932
[4]The Heat Requirements of Buildings, by J H Walker and G H Tuttle (A S H V E TRANSACTIONS, Vol 41, 1935, p 171)

196 buildings located in 21 different cities in the United States. Being averages, and for small groups in each type, the figures may need considerable modification to allow for local variations. It should be especially noted that the steam used for heating hot water is included in the values given in Table 8, but in the case of office buildings, the steam for heating only is also shown. Presentation of the unit consumption in three ways permits making the estimate if either the calculated heat loss or the volume of net heated space in the building is known.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 9.

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization habits. Thus, in Table 9, the theatres, operating for short hours, have a load factor of 0 126 as compared with the figure of 0 318 for clubs and lodges.

PROBLEMS IN PRACTICE

1 ● What will be the estimated fuel cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The seasonal efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 p.m. to 7 a.m. Assume that the price of gas is 7 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

The average hourly temperature is:

$$t_a = \frac{(72 \times 16) + (55 \times 8)}{24} = 66\ 3\ \text{F.}$$

The maximum hourly heat loss will be:

$$H = 92{,}000 - \frac{26{,}000}{2} = 79{,}000\ \text{Btu.}$$

$$M = \frac{79{,}000\ (66.3 - 36\ 4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2204\ 6\ \text{hundred thousand Btu.}$$

$2204\ 6 \times 0\ 07 = \$154.34$ = estimated fuel cost per year of heating building.

2 ● What factors should be taken into consideration when determining the efficiency at which a fuel will be burned?

Manufacturers' catalogs usually give equipment efficiencies obtained under test conditions. These values do not allow for poor attendance, defects in installation, or poor draft. Such efficiencies do not consider heat radiated from the outside of the equipment, but in many cases this heat is utilized Neither do they allow for the fact that the efficiency under less than rated capacity is frequently lower than at the rating point.

3 ● Make an estimate of the gas required to heat a building located in Chicago, Ill., assuming that the calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperature of 0 F and 70 F. Chicago has 800 Btu mixed gas, and 6290 degree-days.

Using Equation 3 and Table 4, the fuel consumption for a design temperature of 0 F with 800 Btu gas is found to be 0 085 cu ft of gas per degree-day per square foot of hot water radiation.

$$0.085 \times 1000 \times 6290 = 534{,}650 \text{ cu ft.}$$

4 ● A building in Marquette, Mich., has an hourly heat loss at design conditions of 240,000 Btu per hour. If the inside design temperature is to be 70 F and the outside design temperature is —10 F, what will be the estimated normal seasonal coal consumption for heating if 12,000 Btu per pound fuel is burned at a 50 per cent seasonal efficiency, and what part of the total will be used during November, December, and January?

From Table 7, U, uncorrected, is 0 0666 lb of coal per 1000 Btu per hour heat loss Correcting for the outside design temperature, Table 5, the corrected value of U is 0 875 × 0 0666 = 0 0583 From Table 3, D is 8693 and from the problem N is 240. Substituting in Equation 3.

$$F = 0.0583 \times 240 \times 8693 = 121{,}632 \text{ lb.}$$

Fuel used over any period is, according to the theory of the degree-day, proportional to the number of degree-days during the period. From Table 3, the average number of degree-days for November, December, and January in Marquette are 960, 1302, and 1500, a total of 3762. The yearly total is 8693, so that during these three months the estimated consumption is:

$$\frac{3762}{8693} \times 121{,}632 = 52{,}638 \text{ lb}$$

5 ● Careful estimates of the probable fuel consumption of a building in Baltimore based on average degree-days as shown in Table 3 indicated that the fuel consumed should total 13,600 gal of oil in a normal year. The first heating season the building was in operation, the winter was cold and the degree-days totaled 4741 and the oil consumed totaled 14,500 gal. Was the plant performing according to the estimate or not?

From Table 3, the average number of degree-days in Baltimore is 4533 During the period under consideration the degree-days were above normal, so that the calculated consumption would have been.

$$\frac{4741}{4533} \times 13{,}600 \text{ gal} = 14{,}223 \text{ gal}$$

Since the building consumed 14,500 gal the operating performance was slightly worse than expected after allowance was made for the severe winter

6 ● Which item may be determined more closely, the heating value of a fuel or the efficiency of its combustion?

The heating values of oil, gas, and solid fuels are closely determinable, whereas the efficiency of burning depends on the particular equipment chosen and the skill used in handling it.

Chapter 13

HEATING BOILERS

Cast-Iron Boilers, Steel Boilers, Special Heating Boilers, Gas-Fired Boilers, Hot Water Supply Boilers, Furnace Design, Heating Surface, Testing and Rating Codes, Output Efficiency, Selection of Boilers, Connections and Fittings, Erection, Operation and Maintenance, Boiler Insulation

STEAM and hot water boilers for low pressure heating work are built in a wide variety of types, many of which are illustrated in the *Catalog Data Section*, and are classified as (1) cast-iron sectional, (2) steel fire tube, (3) steel water tube, and (4) special.

CAST-IRON BOILERS

Cast-iron boilers may be of round pattern with circular grate and horizontal pancake sections joined by push nipples and tie rods, or of rectangular pattern with vertical sections. The latter type may be either of outside header construction where each section is independent of the other and the water and steam connections are made externally through these headers, or assembled with push nipples and tie rods, in which case the water and steam connections are internal.

Cast-iron boilers usually are shipped knocked down to facilitate handling at the place of installation where assembly is made. One of the chief advantages of cast-iron boilers is that the separate sections can be taken into or out of basements and other places more or less inaccessible after the building is constructed. This feature is of importance in making repairs to or replacing a damaged or worn-out boiler and should be given consideration in the original selection. Sufficient space should be provided in the boiler room for assembling the boiler and for disassembling it conveniently if repairs are needed. With the outside header type of boiler a damaged section in the middle of the boiler can be removed without disturbing the other sections so side clearance should be provided.

Capacities of cast-iron boilers range from that required for small residences up to about 18,000 sq ft of steam radiation. For larger loads, cast-iron boilers must be installed in multiple, or a steel boiler must be used. In most cases cast-iron boilers are limited to working pressures of 15 lb for steam and 30 lb for water. Special types are built for hot water supply which will withstand higher local water pressures.

STEEL BOILERS

Two general classifications may be applied to steel boilers: *first*, with regard to the relative position of water and hot gases, distinguished as fire tube or water tube; *second*, with regard to arrangement of furnace and

flues, as (1) horizontal return tubular (HRT) boilers, (2) portable (self-contained) firebox boilers with either water or fire tubes, and (3) water tube boilers of the power type.

Fire tube boilers are constructed so that the water available to produce steam is contained in comparatively large bodies distributed outside of the boiler tubes, the hot gases passing within the tubes. In *water tube* boilers, the water is circulated within the boiler tubes, heat being applied externally to them.

The *HRT boiler* is the oldest type and consists of a horizontal cylindrical shell with fire tubes, enclosed in brickwork to form the furnace and

Table 1. Practical Combustion Rates for Small Coal-Fired Heating Boilers Operating on Natural Draft of from 1/8 in. to 1/2 in. Water[a]

Kind of Coal	Sq Ft Grate	Lb of Coal per Sq Ft Grate per Hour
No 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15.5

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

combustion chamber. All heating surfaces and the interior of the boiler are accessible for both cleaning and inspection. Horizontal return tubular boilers, especially the larger sizes, should be suspended from structural columns and beams independent of the brick setting. Small HRT boilers sometimes are supported by brackets resting on the brick setting.

Portable firebox boilers are the more generally used type of steel heating boilers, their outstanding characteristic being the water-jacketed firebox which eliminates virtually all brickwork. They are shipped in one piece from the factory and come to the job ready for immediate hook-up to piping. They may be of welded or riveted construction and have either water or fire tubes. Manufacturers' catalogs usually list heating surface as well as grate area. The elimination of brickwork also makes this type the most compact of steel boilers as well as the lowest in first cost.

Water tube boilers. For large heating loads water tube boilers are quite frequently used. They usually require more headroom than other types of boilers but require considerably less floor space and make possible a

much higher rate of evaporation per square foot of heating surface, with proper setting, baffling and draft. Water tube boilers used for heating purposes are either completely supported, insulated and encased in steel, or else brick set, supported on structural steel columns and have the brick setting encased in an insulated steel housing to prevent air infiltration and to minimize heat losses. For large heating loads at a high rate of evaporation, such boilers should be operated at pressures above 15 lb per square inch with a pressure-reducing valve on the connection to the heating main.

SPECIAL HEATING BOILERS

A special type of boiler, known as the *magazine feed boiler*, has been developed for the burning of small sizes of anthracite and coke. These are built of both cast-iron and steel, and have a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types using buckwheat sizes of coal. Special attention must be given to insure adequate draft and proper chimney sizes and connections.

Oil-burner boiler units, in which a special boiler has been designed with a furnace shaped to meet the general requirements of oil burners or are specially adapted to one particular burner have been developed by a number of manufacturers. These usually are compact units with the burner and all controls enclosed within an insulated steel jacket. Ample furnace volume is provided for efficient combustion, and the heating surfaces are proportioned for effective heat transfer. Consequently, higher efficiencies are obtainable than with the ordinary coal-fired boiler converted to oil firing.

GAS-FIRED BOILERS

Gas boilers have assumed a well-defined individuality. The usual boiler is sectional in construction with a number of independent burners placed beneath the sections. In most boilers each section has its own burner. In all cases the sections are placed quite closely together, much closer than would be possible when burning a soot-forming fuel. The effort of the designer is always to break the hot gas up into thin streams, so that all particles of the heat-carrying gases can come as close as possible to the heat-absorbing surfaces. Because there is no fuel bed resistance and because the gas company supplies the motive power to draw in the air necessary for combustion (in the form of the initial gas pressure), draft losses through gas boilers are low. See Chapter 11.

HOT WATER SUPPLY BOILERS

Boilers for hot water supply are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-making properties of the water supplied. If water temperatures are maintained below 140 F the life of the heater will be much longer than if higher temperatures are used, owing to decreased scale formation and minimized corrosion below 140 F. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of the heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam heating system is installed, the domestic hot water usually is obtained from an indirect heater placed below the water line of the boiler.

FURNACE DESIGN

Good efficiency and proper boiler performance are dependent on correct furnace design embodying sufficient volume for burning the particular fuel at hand, which requires thorough mixing of air and gases at a high temperature with a velocity low enough to permit complete combustion of all the volatiles. On account of the small amount of volatiles contained in coke, anthracite, and semi-bituminous coal, these fuels can be burned efficiently with less furnace volume than is required for bituminous coal, the combustion space being proportioned according to the amount of volatiles present.

Combustion should take place before the gases are cooled by the boiler heating surface, and the volume of the furnace must be sufficient for this purpose. The furnace temperature must be maintained sufficiently high to produce complete combustion, thus resulting in a higher CO_2 content and the absence of CO. Hydrocarbon gases ignite at temperatures varying from 1000 to 1500 F.

The question of furnace proportions, particularly in regard to mechanical stoker installations, has been given some consideration by various manufacturers' associations. Arbitrary values have been recommended for minimum dimensions. A customary rule-of-thumb method of figuring furnace volumes is to allow 1 cu ft of space for a maximum heat release of 50,000 Btu per hour. This value is equivalent to allowing approximately 1 cu ft for each developed horsepower, and it is approved by most smoke prevention organizations.

The setting height will vary with the type of stoker. In an overfeed stoker, for instance, all the volatiles must be burned in the combustion chamber and, therefore, a greater distance should be allowed than for an underfeed stoker where a considerable portion of the gas is burned while passing through the incandescent fuel bed. The design of the boiler also may affect the setting height, since in certain types the gas enters the tubes immediately after leaving the combustion chamber, while in others it passes over a bridge wall and toward the rear, thus giving a better opportunity for combustion by obtaining a longer travel before entering the tubes.

To secure suitable furnace volume, especially for mechanical stokers or oil burners, it often is necessary either to pit the stoker or oil burner, or

where water line conditions and headroom permit, to raise the boiler on a brick foundation setting.

Smokeless combustion of the more volatile bituminous coals is furthered by the use of mechanical stokers. (See Chapter 11.) Smokeless combustion in hand-fired boilers burning high volatile solid fuel is aided (1) by the use of double grates with down-draft through the upper grate, (2) by the use of a curtain section through which preheated auxiliary air is introduced over the fire toward the rear of the boiler, and (3) by the introduction of preheated air through passages at the front of the boiler. All three methods depend largely on mixing secondary air with the partially burned volatiles and causing this mixture to pass over an incandescent fuel bed, thus tending to secure more complete combustion than is possible in boilers without such provision.

HEATING SURFACE

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed. The heat transfer capacity of a radiant heating surface may be as high as 6 to 8 times that of an indirect surface. This is one of the reasons why the water legs of some boilers have been extended, especially in the case of stoker firing where the extra amount of combustion chamber secured by an extension of the water legs is important. For the same reason, care should be exercised in building a refractory combustion chamber in an oil-burning boiler so as not to screen any more of this valuable surface with refractories than is necessary for good combustion.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. Investigations[1] by the U. S. Bureau of Mines show that:

1. A boiler in which the heating surface is arranged to give long gas passages of small cross-section will be more efficient than a boiler in which the gas passages are short and of larger cross-section.
2. The efficiency of a water tube boiler increases as the free area between individual tubes decreases and as the length of the gas pass increases.
3. By inserting baffles so that the heating surface is arranged in series with respect to the gas flow, the boiler efficiency will be increased.

The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed.

Heat Transfer Rates

Practical rates of heat transfer in heating boilers will average about 3300 Btu per sq ft per hour for hand-fired boilers and 4000 Btu per sq ft

[1]See *U. S Bureau of Mines Bulletin* No 18, The Transmission of Heat into Steam Boilers

per hour for mechanically fired boilers when operating at *design load.* When operating at *maximum load*[2] these values will run between 5000 and 6000 Btu per sq ft per hour. Boilers operating under favorable conditions at the above heat transfer rates will give exit gas temperatures that are considered consistent with good practice.

TESTING AND RATING CODES

The Society has adopted three solid fuel testing codes, a solid fuel rating code and an oil fuel testing code. A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June 1929)[3], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics. A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[3] is intended for use with A.S.H.V E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[4]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler. The A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[5] is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers.

Steel Heating Boilers Ratings

The *Steel Heating Boiler Institute* has adopted a method for the rating of low pressure boilers based on their physical characteristics and expressed in square feet of steam or water radiation or in Btu per hour as given in Table 2. The following requirements are included in this Code:

1. One square foot of steam radiation is to be considered equal to the emission of 240 Btu per hour and one square foot of water radiation is to be considered equal to emission of 150 Btu per hour.

2. The rating of a boiler expressed in square feet of steam radiation in which solid fuel hand fired is used is based on the amount equal to 14 times the heating surface of the boiler in square feet.

3 The rating of a boiler expressed in square feet of steam radiation in which solid fuel mechanically fired, or in which oil or gas is burned, is based on the amount equal to 17 times the heating surface of the boiler in square feet.

4. Heating surface is to be expressed in square feet and include those surfaces in the boiler which are exposed to the products of combustion on one side and water on the other. In measuring surfaces, the outer tube areas are to be considered. When a boiler has the water leg height increased the heating surface noted in the published ratings are not to be increased

5 A grate area is to be considered as an area of the grate surface expressed in square feet and measured in the plane of the top surface of the grate. For double grate boilers the grate surface is to be considered as the area of the upper grate plus one-quarter of the area of the lower grate

6. The grate area of a boiler for rating as determined in No. 2 is to be not less than that determined by the following formulae

For boilers with ratings 1800 sq ft to 4000 sq ft of steam radiation:

$$\text{Grate Area} = \sqrt{\frac{\text{Catalogue Rating (in square feet steam radiation)} - 200}{25.5}} \quad (1)$$

[2]For definitions of design load and maximum load see pages 257 and 258.
[3]See A S H V E TRANSACTIONS, Vol 35, 1929, p 12 Also Chapter 45
[4]See A S H V E TRANSACTIONS, Vol 36, 1930, p 35 Also Chapter 45
[5]See A.S H V E TRANSACTIONS, Vol 37, 1931, p 23 Also Chapter 45

TABLE 2. STANDARD STEEL HEATING BOILER RATINGS[a]

Hand Fired Capacity Rating					Mechanically Fired Capacity Rating			
Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hr	Heating Surface Sq Ft	Grate Area Sq Ft	Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hr	Furnace Volume Oil, Gas or Bituminous Coal Cu Ft
1,800	2,880	432,000	129	7.9	2,190	3,500	525,600	15.7
2,200	3,520	528,000	158	8.9	2,680	4,280	643,200	19.2
2,600	4,160	624,000	186	9.7	3,160	5,050	758,400	22.6
3,000	4,800	720,000	215	10.5	3,650	5,840	876,000	26.1
3,500	5,600	840,000	250	11.4	4,250	6,800	1,020,000	30.4
4,000	6,400	960,000	286	12.2	4,860	7,770	1,166,400	34.8
4,500	7,200	1,080,000	322	13.4	5,470	8,750	1,312,800	39.1
5,000	8,000	1,200,000	358	14.5	6,080	9,720	1,459,200	43.5
6,000	9,600	1,440,000	429	16.4	7,290	11,660	1,749,600	52.1
7,000	11,200	1,680,000	500	18.1	8,500	13,600	2,040,000	60.8
8,500	13,600	2,040,000	608	20.5	10,330	16,520	2,479,200	73.8
10,000	16,000	2,400,000	715	22.5	12,150	19,440	2,916,000	86.8
12,500	20,000	3,000,000	893	25.6	15,180	24,280	3,643,200	108.5
15,000	24,000	3,600,000	1,072	28.4	18,220	29,150	4,372,800	130.2
17,500	28,000	4,200,000	1,250	30.9	21,250	34,000	5,100,000	151.8
20,000	32,000	4,800,000	1,429	33.2	24,290	38,860	5,829,600	173.5
25,000	40,000	6,000,000	1,786	37.4	30,360	48,570	7,286,400	216.9
30,000	48,000	7,200,000	2,143	41.2	36,430	58,280	8,743,200	260.3
35,000	56,000	8,400,000	2,500	44.7	42,500	68,000	10,200,000	303.6

[a]Adopted by the *Steel Heating Boiler Institute* in cooperation with the *Bureau of Standards, United States Department of Commerce Simplified Practice Recommendation R 157-35*

For boilers with ratings 4000 sq ft of steam radiation and larger.

$$\text{Grate Area} = \sqrt{\frac{\text{Catalogue Rating (in square feet steam radiation)} - 1500}{16.8}} \qquad (2)$$

7. The volume for furnaces in which solid fuel is burned is to be considered as the cubical content of the space between the bottom of the fuel bed and the first plane of entry into or between the tubes Volume of furnaces in which pulverized liquid fuel or gaseous fuel is burned are to be considered as the cubical content of the space between the hearth and the first plane of entry into or between the tubes No minimum furnace volume is to be specified for mechanical-fired boilers burning anthracite

8 The furnace volume for a boiler, with a rating as determined in No 3 in which oil, gas or bituminous coal stoker fired is burned is not to be less than one cubic foot for every 140 sq ft of steam rating.

9. The average height of furnace for the rating determined in No 3 in which bituminous coal, stoker fired is burned is not to be less than that determined graphically in Fig. 1 or mathematically by the following formula

$$H = \sqrt{\frac{R}{22.5}} + \sqrt{\frac{R}{A}} \qquad (3)$$

where

H = average furnace height, inches as determined by the following formula

$$H = \frac{12F}{A} = \frac{12F}{WL}$$

R = stoker fired boiler rating, square foot steam radiation.
A = plan area of firebox, square feet measured at the bottom of the fuel bed
F = furnace volume, cubic feet.

W = average width of furnace, measured at the bottom of the fuel bed, feet.
L = length of furnace, feet. If the furnace is longer than the fuel bed or contains a bridge wall, the total length of the furnace may be used except that this length is not to exceed $2\frac{1}{2}$ W.

BOILER OUTPUT

Boiler output as defined in A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) is the quantity of heat available at the boiler nozzle with the boiler normally insulated. It should be based on actual tests conducted in accordance with this code. This output is usually stated in Btu and in square feet of equivalent heating surface (radiation). According to the A.S.H.V.E. Standard Code for

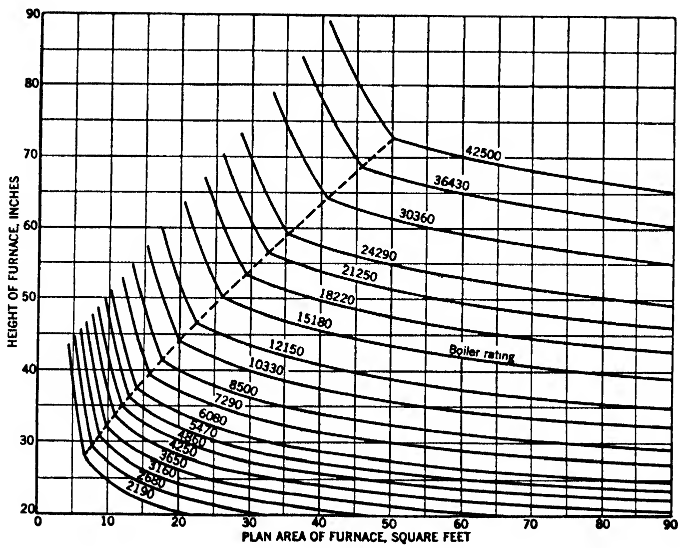

FIG. 1. FURNACE HEIGHTS FOR STOKER FIRED BOILERS AND BITUMINOUS COAL RATED IN SQUARE FEET STEAM RADIATION

Rating Steam Heating, Solid Fuel, Hand-Fired Boilers, the performance data should be given in tabular or curve form on the following items for at least five outputs ranging from maximum down to 35 per cent of maximum: (1) fuel available, (2) combustion rate, (3) efficiency, (4) draft tension, (5) flue gas temperature. The only definite restriction placed on setting the maximum output is that priming shall not exceed 2 per cent. These curves provide complete data regarding the performance of the boiler under test conditions. Certain other pertinent information, such as grate area, heating surface and chimney dimensions is desirable also in forming an opinion of how the boiler will perform in actual service.

The output of large heating boilers is frequently stated in terms of *boiler horsepower* instead of in Btu per hour or square feet of equivalent radiation.

Boiler Horsepower: The evaporation of 34.5 lb of water per hour *from and at 212 F* which is equivalent to a heat output of $970.2 \times 34.5 = 33{,}471.9$ Btu per hour.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at this same temperature and at atmospheric pressure.

It is usually considered that 10 sq ft of boiler heating surface will produce a rated boiler horsepower. A rated boiler horsepower in turn can carry a design load of from 100 to 140 sq ft of equivalent radiation. It is apparent, therefore, that 1 sq ft of boiler heating surface can carry a design load of from 10 to 14 sq ft of equivalent radiation, or somewhat more if the boiler is forced above rating. The application of these values is discussed under the heading Selection of Boilers.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

2. *Liquid Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel to the calorific value of 1 lb of fuel.

Solid fuel boilers usually show an efficiency of 50 to 75 per cent when operated under favorable conditions at their rated capacities. Information on the combined efficiencies of boiler, furnace and burner has resulted from research conducted at Yale University in cooperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*[6].

SELECTION OF BOILERS

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932).

The estimated design load is the sum of the following three items[7]:

1. The estimated heat emission in Btu per hour of the connected radiation (direct, indirect or central fan) to be installed.

2. The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler.

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[6]A S H V E RESEARCH REPORT No 907—Study of the Characteristics of Oil Burners and Heating Boilers, by L E Seeley and E J Tavanlar (A S H V E TRANSACTIONS, Vol 37, 1931, p 517) A S H V E RESEARCH REPORT No 925—A Study of Intermittent Operation of Oil Burners, by L E Seeley and J. H Powers (A S H V E TRANSACTIONS, Vol 38, 1932, p 317)

[7]A S H V E Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929)

3. The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or the equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932.)

The estimated maximum load is given by[8]:

4. The estimated increase in the normal load in Btu per hour due to starting up cold radiation. This percentage of increase is to be based on the sum of Items 1, 2 and 3 and the heating-up factors given in Table 3.

TABLE 3. WARMING-UP ALLOWANCES FOR LOW PRESSURE STEAM AND HOT WATER HEATING BOILERS[a, b, c]

Design Load (Representing Summation of Items 1, 2, and 3,[d]		Percentage Capacity to Add for Warming Up
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A S H V E Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C Taggert (A S H V E TRANSACTIONS, Vol 19, 1913, p 292), Report of A S H V E Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A S H V E TRANSACTIONS, Vol 36, 1930, p 35), Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932)

[c]This table refers to hand-fired, solid fuel boilers A factor of 20 per cent over design load is adequate when automatically-fired fuels are used (see Fig 3)

[d]240 Btu per square foot

Other things to be considered are·

5. Efficiency with hard or soft coal, gas, or oil firing, as the case may be
6. Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas
7. Combustion space in the furnace
8. Type of heat liberation, whether continuous or intermittent, or a combination of both.
9. Miscellaneous items consisting of draft available, character of attendance, possibility of future extension, possibility of breakdown and headroom in the boiler room

Radiation Load

The connected radiation (Item 1) is determined by calculating the heat losses in accordance with data given in Chapters 5, 6 and 7, and dividing by 240 to change to square feet of equivalent radiation as explained in Chapter 14. For hot water, the emission commonly used is 150 Btu per square foot, but the actual emission depends on the temperature of the medium in the heating units and of the surrounding air. (See Chapter 14.)

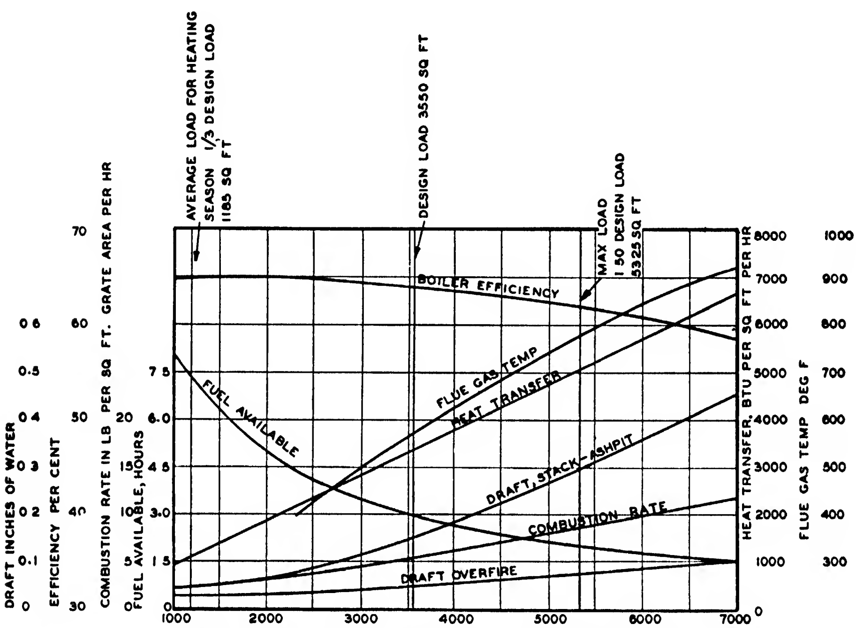
Although it is customary to use the actual connected load in equivalent square feet of radiation for selecting the size of boiler, this connected load usually represents a reserve in heating capacity to provide for infiltration in the various spaces of the building to be heated, which reserve, however,

[8]Loc Cit Note 7

is not in use at all places at the same time, or in any one place at all times. For a further discussion of this subject see Chapter 6.

Hot Water Supply Load

When the hot water supply (Item 2) is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. The



BOILER DATA—

GRATE AREA, SQ FT	18 0	FUEL—BITUMINOUS 3/4" LUMP		
HEATING SURFACE, SQ FT	254	ANALYSIS—	VOLATILE MATTER	34 68%
WEIGHT, LB	9160		FIXED CARBON	55 44
FUEL CAPACITY, LB	651		ASH	9 87
FUEL AVAILABLE, LB	414		SULPHUR	2 86
FUEL DEPTH, IN	10		MOISTURE	3 00
			BTU PER LB	13,655

FIG. 2 TYPICAL PERFORMANCE CURVES FOR A 36-IN. CAST-IRON SECTIONAL STEAM HEATING BOILER, BASED ON THE A S.H.V.E. CODE FOR RATING STEAM HEATING SOLID FUEL HAND-FIRED BOILERS

allowance to be made will depend on the amount of water heated and its temperature rise. A good approximation is to add 4 sq ft of equivalent radiation for each gallon of water heated per hour through a temperature range of 100 F. For more specific information, see Chapter 43.

Piping Tax (Item 3)

It is common practice to add a flat percentage allowance to the equivalent connected radiation to provide for the heat loss from bare and covered pipe in the supply and return lines. The use of a flat allowance of 25 per cent for steam systems and 35 per cent for hot water systems is preferable to ignoring entirely the load due to heat loss from the supply

and return lines, but better practice, especially when there is much bare pipe, is to compute the emission from both bare and covered pipe surface in accordance with data in Chapter 39. A chart is shown in Fig. 3 indicating percentage allowances for piping and warming-up which are applicable to automatically-fired heating plants using steam radiation. With direct radiation served by bare supply and return piping the percentages may be higher than those stated, while in the case of unit heaters where the output is concentrated in a few locations, the piping tax may be 10 per cent or less.

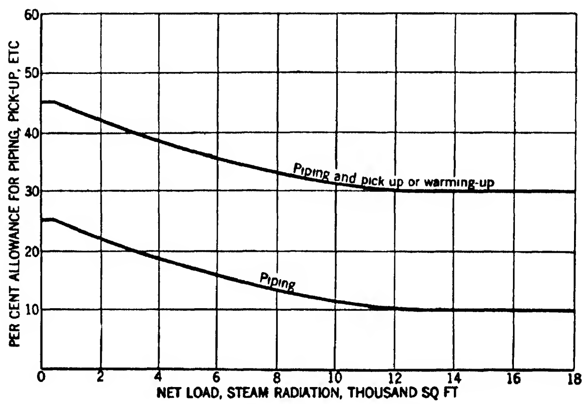


FIG. 3. PERCENTAGE ALLOWANCE FOR PIPING AND WARMING-UP

Warming-Up Allowance

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature and heating up cold radiation and piping. (See Item 4.) The factors to be used for determining the allowance to be made should be selected from Table 3 and should be applied to the estimated design load as determined by Items 1, 2 and 3. While in every case the estimated maximum load will exceed the design load if adequate heating response is to be achieved, there is however, no object in over-estimating the allowances, as the only effect would be to reduce the time of warming-up by a few minutes. Otherwise, it might result in firing the boiler unduly and increasing the cost of operation.

Performance Curves for Boiler Selection

In the selection of a boiler to meet the estimated load, the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers recommends the use of performance curves based on actual tests conducted in accordance with the A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3), similar to the typical curves shown in Fig. 2. It should be understood that performance data apply to test conditions and that a reasonable allowance should be made for decreased output resulting from soot deposit, poor fuel or inefficient attention.

Selection Based on Heating Surface and Grate Area

Where performance curves are not available, a good general rule for conventionally-designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load consisting of connected radiation, piping tax and domestic water heating load. As stated in the section on Boiler Output, this is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the maximum load including the warming-up allowance will be provided for by operating the boiler in excess of the design load, that is, in excess of the 100 per cent rating on a boiler-horsepower basis.

Due to the wide variation encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{4}$$

where

G = grate area, square feet.

H = required total heat output of the boiler, Btu per hour (see Selection of Boilers, p. 251).

C = combustion rate in pounds of dry coal per square foot of grate area per hour, depending on the kind of fuel and size of boiler as given in Table 1.

F = calorific value of fuel, Btu per pound.

E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required heat output of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Formula 4. With small boilers where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Selection of Steel Heating Boilers

Boiler ratings previously described under the *Steel Heating Boiler Institute's* Boiler Rating Code are intended to correspond with the estimated design load based on the sum of items 1, 2 and 3 outlined on pages 257 and 258. Insulated residence type boilers for oil or gas may carry a net load expressed in square feet of steam radiation of not more than 17 times the square feet of heating surface in the boiler, provided the boiler manufacturer guarantees the boiler to be capable of operating at a maximum output of not less than 150 per cent of net load rating with overall efficiency of not less than 75 per cent with at least two different makes of each type of standard commercial burner recommended by the boiler manufacturer. If the heat loss from the piping system exceeds 20 per cent of the installed radiation, the excess is to be considered as a part of the net load.

When the estimated heat emission of the piping (connecting radiation, and other apparatus to the boiler) is not known the net load to be considered for the boiler may be determined from Table 4.

Selection of Gas-Fired Boilers

Gas-heating appliances should be selected in accordance with the percentage allowances given in Fig. 3. These factors are for thermostatically-controlled systems; in case manual operation is desired, a warming-up allowance of 100 per cent is recommended by the *A.G.A.* A gas boiler selected by the use of the *A.G.A.* factors will be the minimum

TABLE 4. BOILER RATINGS BASED ON NET LOAD[a]

Hand Fired Ratings		Mechanically Fired Ratings	
Steam Radiation Sq Ft	Net Load[b] Steam Radiation Sq Ft	Steam Radiation Sq Ft	Net Load[b] Steam Radiation Sq Ft
1,800	1,389	2,190	1,695
2,200	1,702	2,680	2,089
2,600	2,020	3,160	2,461
3,000	2,335	3,650	2,853
3,500	2,732	4,250	3,335
4,000	3,135	4,860	3,830
4,500	3,540	5,470	4,330
5,000	3,945	6,080	4,834
6,000	4,770	7,290	5,850
7,000	5,608	8,500	6,885
8,500	6,885	10,330	8,490
10,000	8,197	12,150	10,125
12,500	10,417	15,180	12,650
15,000	12,500	18,220	15,183
17,500	14,584	21,250	17,708
20,000	16,667	24,290	20,242
25,000	20,834	30,360	25,300
30,000	25,000	36,430	30,359
35,000	29,167	42,500	35,417

[a]Adopted by the *Steel Heating Boiler Institute* in cooperation with the *Bureau of Standards, United States Department of Commerce Simplified Practice Recommendation R 157-35*

[b]The net load is made up by the sum of the estimated design load, items 1 and 2 (pages 257 and 258) All net loads are expressed in 70 F For hand-fired boiler ratings less than 1800 sq ft of steam or 2880 sq ft of water and mechanically-fired boiler ratings of 2190 sq ft of steam or 3500 sq ft of water, apply the factor 1 3 to the net load to determine the boiler size For water boilers use the equivalent net load for steam boilers of similar physical size.

size boiler which can carry the load. From a fuel economy standpoint, it may be advisable to select a somewhat larger boiler and then throttle the gas and air adjustments as required. This will tend to give a low stack temperature with high efficiency and at the same time provide reserve capacity in case the load is under-estimated or more is added in the future.

Conversions

The conversion of a coal or oil boiler to gas burning is simpler than the reverse since little furnace volume need be provided for the proper combustion of gas. When a solid fuel boiler of 500 sq ft (or less) capacity is converted to gas burning, the necessary gas heat units should be approximately double the connected load. The presumption for a conversion job is that the boiler is installed and probably will not be made larger.

therefore, it is a matter of setting a gas-burning rate to obtain best results with the available surface. Assuming a combustion efficiency of 75 per cent for a conversion installation the boiler output would be $2 \times 0.75 = 1.5$ times the connected load, which allows 50 per cent for piping tax and pickup. In converting large boilers, the determination of the required Btu input should not be done by an arbitrary figure or factor but should be based on a detailed consideration of the requirements and characteristics of the connected load.

An efficient conversion installation depends upon the proper size of flue connection. Often the original smoke breeching between the boiler and chimney is too large for gas firing, and in this case, flue orifices can be used. They are discs provided with an opening of the size for the gas input used in this boiler. The size should be based on 1 sq in. of flue area for each 7500 hourly Btu input.

If dampers are found in the breeching they should be locked in position so that they will not interfere with the normal operation of the gas burners at maximum flow. In the case of large boiler conversions, automatic damper regulators proportion the position of the flue dampers to the amount of gas flowing and may be substituted for existing dampers. Generally in residence conversions automatic dampers are not of the proportioning type but close the flue during the off periods of the gas burners. Automatic shutoff dampers should be located between the back draft diverter and the chimney flue. Automatic dampers are usually designed to operate with electric contact mechanism, but frequently an arrangement is utilized which functions with mechanical fluid or gas pressure.

Physical Limitations

As it will usually be found that several boilers will meet the specifications, the final selection of the boiler may be influenced by other considerations, some of which are:

1. Dimensions of boiler.
2. Durability under service.
3. Convenience in firing and cleaning.
4. Adaptability to changes in fuel and kind of attention.
5. Height of water line.

In large installations, the use of several smaller boiler units instead of one larger one will obtain greater flexibility and economy by permitting the operation, at the best efficiency, of the required number of units according to the heat requirements.

Space Limitations

Boiler rooms should, if possible, be situated at a central point with respect to the building and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flues, and should be at least 3 ft greater than the length of the boiler firebox.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various

dampers, cleanouts and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts. With large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories and for proper installation of piping. In general the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

When steel boilers are used, space should be provided for the removal and replacement of tubes.

CONNECTIONS AND FITTINGS

The velocity of flow through the outlets of low pressure steam heating boilers should not exceed 15 to 25 fps if fluctuation of the water line and undue entrainment of moisture are to be avoided. Steam or water outlet connections preferably should be the full size of the manufacturers' tapping and should extend vertically to the maximum height available above the boiler. For gravity circulating steam heating systems, it is recommended that a Hartford Loop, described in Chapter 16, be utilized in making the return connection.

Particular attention should be given to *fitting connections* to secure conformity with the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections and safety valve capacity.

Steam gages should be fitted with a water seal and a shut-off consisting of a cock with either a tee or lever handle which is parallel to the pipe when the cock is open. *Steam gage connections* should be of copper or brass when smaller than 1 in. *I.P.S.*[9] if the gage is more than 5 ft from the boiler connection, and also in any case where the connection is less than ½ in. *I.P.S.*

Each steam or vapor boiler should have at least one *water gage* glass and two or more *gage cocks* located within the range of the visible length of the glass. The water gage fittings or gage cocks may be direct connected to the boiler, if so located by the manufacturer, or may be mounted on a separate water column. No connections, except for combustion regulators, drains or steam gages, should be placed on the pipes connecting the water column and the boiler. If the water column or gage glass is connected to the boiler by pipe and fittings, a cross, tee or equivalent, in which a cleanout plug or a drain valve and piping may be attached, should be placed in the water connection at every right-angle turn to facilitate cleaning. The water line in steam boilers should be carried at the level specified by the boiler manufacturer.

Safety valves should be capable of discharging all the steam that can be generated by the boiler without allowing the pressure to rise more than 5 lb above the maximum allowable working pressure of the boiler. This should be borne in mind particularly in the case of boilers equipped with mechanical stokers or oil burners where the amount of grate area has little significance as to the steam generating capacity of the boiler.

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[9] *A S M E* Code, Identification of Piping Systems

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent clean-out plug should be provided in the case of a single return connection.

Blow-off or drain connections should be made near the boiler and so arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

For further data on pipe connections for steam and hot water heating systems, see Chapters 16 and 17 and the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining and where practicable it should be grouted from the inside of the chimney. A thimble or sleeve grout usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer always should be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement which cannot be reached after assembly is complete must be finished as the assembly progresses.

The following precautions should be taken in all installations to prevent damage to the boiler:

1. There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather.

2. Strains on the boiler due to movement of piping during expansion should be prevented by suitable anchoring of piping and by proper provision for pipe expansion and contraction.

3. Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material

4. Condensation must flow back to the boiler as rapidly and uniformly as possible Return connections should prevent the water from backing out of the boiler.

5. Automatic boiler feeders and low water cut-off devices which shut off the source

of heat if the water in the boiler falls below a safe level are recommended for boilers mechanically fired.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing, (*d*) boiler too small, (*e*) improper piping; (*f*) improper arrangement of sections, (*g*) heating surfaces covered with soot; and (*h*) insufficient radiation installed.

2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; and (*c*) boiler operating at excessive output.

3. *Water disappears from gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output, and (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft resulting from air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; and (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft, (*b*) smoky combustion; (*c*) too low a rate of combustion, and (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; and (*e*) improper reduction in breeching size.

Cleaning Steam Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities have a tendency to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least 1¼ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure and while fire is burning briskly open valve in blow-off line. When

pressure recedes close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use in a boiler, as given by the boiler manufacturer, should be carefully followed.

It is common practice when starting new installations to discharge heating returns to the sewer during the first week of operation. This prevents the passage of grease, dirt or other foreign matter into the boiler and consequently may avoid the necessity of cleaning the boiler. During the time the returns are being passed to the sewer, the feed valve should be cracked sufficiently to maintain the proper water level in the boiler.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulphur from the fuel with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.
2. All machined surfaces should be coated with oil or grease.
3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.
4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard of some one inadvertently building a fire in a dry boiler, however, it is safer to keep the boiler filled with water. A hot water system usually is left filled to the expansion tank.
5. The grates and ashpit should be cleaned.
6. Clean and repack the gage glass if necessary.
7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.
8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings and regulator parts.

BOILER INSULATION

Insulation for cast-iron boilers is of two general types: (1) plastic material or blocks wired on, cemented and covered with canvas or duck; and (2) blocks, sheets or plastic material covered with a metal jacket furnished by the boiler manufacturer. Self-contained steel firebox boilers usually are insulated with blocks, cement and canvas, or rock wool blankets; HRT boilers are brick set and do not require insulation beyond that provided in the setting. It is essential that the insulation on a boiler

and adjacent piping be of non-combustible material as even slow-burning insulation constitutes a dangerous fire hazard in case of low water in the boiler.

PROBLEMS IN PRACTICE

1 ● What basic requirements of boiler design are to be accomplished with a combination boiler and oil burner unit?

Combination units vary widely but in general, the basic requirements of design depends upon a combustion chamber of proper design and arranged for the flame shape with adequate heating surface for the complete combustion of the fuel.

2 ● What is the normal rating range of each type of boiler?

a. Cast-iron boilers are rated at from 200 to 18,000 sq ft EDR.
b. Steel boilers are rated at from 300 to 50,000 sq ft EDR

3 ● What factors contribute to economical fuel operation in low pressure boilers burning coal or oil?

a. Proper furnace volume for complete combustion.
b. Arrangement of heating surfaces in series to create a turbulent and scrubbing contact of gases against the convective surfaces.
c. Rapid internal water circulation which will remove steam bubbles from the water side of heating surfaces and allow other steam bubbles to be formed. Rapid disengagement of steam bubbles increases the steam generating efficiency of each unit area of heating surface, and thereby lowers flue gas temperatures.

4 ● What equipment is usually directly attached to a low pressure heating boiler?

For coal burning steam boilers: water column, water gage, tri-cocks, steam gage, lever pop safety valve, boiler damper regulator.

For coal burning hot water boilers: damper regulator, altitude gage, thermometer, relief valve.

For oil burning boilers, the damper regulators are omitted and the following additional equipment is usually attached automatic water feeder, low water cutout, a pressure control, and a water temperature control. These are generally furnished by the oil burner manufacturer and do not come with the boiler

5 ● What general precautions regarding the boiler should be taken to make sure a proposed heating installation will work properly?

a. Select the right size and type of boiler
b. Be sure the combustion space is proper for the type of fuel burned.
c. Allow sufficient space around the boiler for cleaning.
d. Secure proper height and area of chimney and connecting breeching.
e. Clean the boiler thoroughly and provide surface blowoff connections and bottom blowoff connections for periodic cleaning after operation is begun.
f. See that the boiler heating surface is cleaned at regular periods.
g. Check flue gas temperatures and make a flue gas analysis at least once a month.
h. Secure information and advice from boiler manufacturer.

6 ● What is the average heat transmission rate in heating boilers in Btu per sq ft of heating surface per hour?

3500 for coal burning boilers, 4200 for oil burning boilers.

Chapter 14

RADIATORS AND GRAVITY CONVECTORS

Heat Emission of Radiators and Convectors, Types of Radiators, Output of Radiators, Heating Effect, Heating Up the Radiator and Convector, Enclosed Radiators, Convectors, Selection, Code Tests, Gravity-Indirect Heating Systems

THE accepted terms for heating units are: (1) *radiators*, for direct surface heating units, either exposed, enclosed, or shielded, which emit a large percentage of their heat by radiation; and (2) *convectors*, for heating units having a large percentage of extended fin surface and which emit heat principally by convection. Convectors are dependent upon enclosures to provide the circulation by gravity of large volumes of air.

HEAT EMISSION OF RADIATORS AND CONVECTORS

All heating units emit heat by *radiation* and *convection*. The resultant heat from these processes depends upon whether or not the heating unit is exposed or enclosed and upon the contour and surface characteristics of the material in the units.

An exposed radiator emits less than half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, radiation is further reduced. The balance of the emission is by conduction to the air in contact with the heating surface, and the resulting circulation of the air warms by convection.

A convector emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible.

TYPES OF RADIATORS

Present day radiators may be classified as tubular, wall, or window types, and are generally made of cast-iron. Catalogs showing the many designs and patterns available now include a junior size sometimes known as slim tube radiation. The tubes in these radiators are materially smaller, and they are compactly assembled in less space than those of the standard radiator.

Pipe Coils

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used

in factory buildings, but now wall type radiators are most frequently used for this service. When coils are used, the miter type assembly is to be preferred as it best cares for expansion in the pipe. Cast manifolds or headers, known as branch tees, are available for this construction.

OUTPUT OF RADIATORS

The output of a radiator can be measured only by the heat it emits. The old standard of comparison used to be square feet of *actual surface*, but since the advance in radiator design and proportions, the surface area alone is not true index of output. (The engineering unit of outputs is the *Mb* or 1000 Btu.) However, during the period of transition from the old to the new, radiators may be referred to in terms of *equivalent square feet*. For steam service this is based on an emission of 240 Btu per hour per square foot and for hot water service 150 Btu per hour per square foot.

TABLE 1. VARIATION IN DIMENSIONS AND CATALOG RATINGS OF 10-SECTION TUBULAR RADIATORS (STEAM)

No. of Tubes		3	4	5	6	7
Width of Radiator	Inches	4.6–5.1	6.0–7.0	8.0–8.9	9.1–10.4	11.4–12.8
Length per Section	Inches	2.5	2.5	2.5	2.5	2.5–3.0
HEIGHT WITH LEGS—INCHES		HEAT EMISSION—EQUIVALENT SQUARE FEET				
13–14		....	....	....	20	25.0–32.5
16–18		....	....	28.5	....	30.0–38.3
20–21		15.0–17.5	20.0–22.5	25.0–31.2	30	36.7–45.0
22–23		20.0–21.3	25	30.0–33.9	35	40.0–45.2
25–26		20.0–26.7	25.0–27.5	32.5–39.8	37.5–40.0	50.0–53.5
30–32		25.0–30.9	33.3–35.0	40.0–48.6	50	63.3–62.5
36–38		30.0–36.7	40.0–42.5	50.0–56.5	60	70.0–75.4

Output of Tubular Radiators

Table 1 illustrates the difficulty in tabulating tubular radiator outputs since there is so much variation in design between the products of the different manufacturers. Only on the four-tube and six-tube sizes is there any practical agreement in output value. The heat emission values appear as square feet but are entirely empirical, being based on the heat emission of the radiator and not on the measured surface.

Output of Wall Radiators

An average value of 300 Btu per actual square foot of surface area per hour has been found for wall radiators one section high placed with their bars vertical. Several recent tests[1] show that this value will be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually so noticeable a difference in temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of a room satisfactorily.

[1]University of Illinois, *Engineering Experiment Station Bulletin* No. 223, p. 30

Output of Pipe Coils

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 2. This has been developed from available data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

TABLE 2. HEAT EMISSION OF PIPE COILS PLACED VERTICALLY ON A WALL (PIPES HORIZONTAL) CONTAINING STEAM AT 215 F AND SURROUNDED WITH AIR AT 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

SIZE OF PIPE	1 IN	1¼ IN	1½ IN
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

Effect of Paint

The prime coat of paint on a radiator has no material effect on the heat output, but the finishing coat may influence the radiation emission and thus affect the heat output. Within the range of temperatures at which radiators operate, color has no appreciable influence on the radiation emitted. Thus, finishing coats of oil paints of various colors will give the same results. However, a bronze paint, applied as the finish coat will change the character of the surface and reduce the amount of heat emitted by radiation. No paint has a noticeable effect on the portion of heat which is given off by convection. The larger the proportion of direct radiating surface, the greater will be the effect of any finish coat of paint which changes the character of the surface. Available tests are on old-style column type radiators which give results as shown in Table 3

TABLE 3. EFFECT OF PAINTING 32-IN. THREE COLUMN, SIX-SECTION CAST-IRON RADIATOR[a]

RADIATOR NO	FINISH	AREA SQ FT	COEFFICIENT OF HEAT TRANS BTU	RELATIVE HEATING VALUE PER CENT
1	Bare iron, foundry finish	27	1.77	100 5
2	One coat of aluminum bronze	27	1.60	90.8
3	Gray paint dipped	27	1.78	101 1
4	One coat dull black Pecora paint	27	1 76	100 0

[a]Comparative Tests of Radiator Finishes, by W H Severns (A S H V E TRANSACTIONS, Vol 33, 1927, p 41)

Effect of Superheated Steam

Available research data indicate that there is probably a decrease in heat transfer rate for a radiator or gravity convector with superheated steam in comparison with saturated steam at the same temperature. The decrease is probably small for low temperatures of superheats and

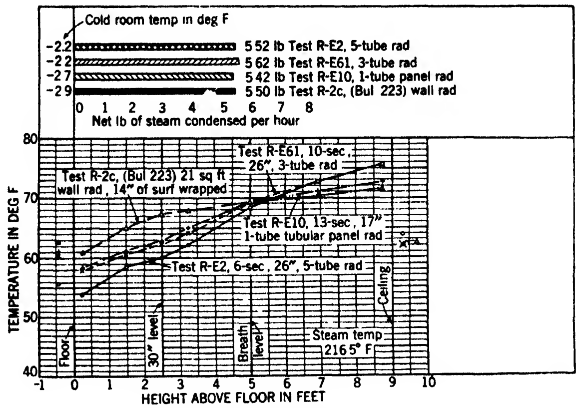


FIG. 1. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 60-IN LEVEL

Note that the steam condensations are practically the same for all four radiators *when the same air temperature of 69 F is maintained at the 60-in level.*

additional tests are necessary with varying degrees of superheat to establish accurate comparisons for all types of radiators and convectors[2].

HEATING EFFECT

For several years the *heating effect* of radiators has been considered by engineers in order to use it for the rating of radiators and in the design of heating systems. Heating effect is the *useful output* of a radiator, in the comfort zone of a room, as related to the total input of the radiator[3].

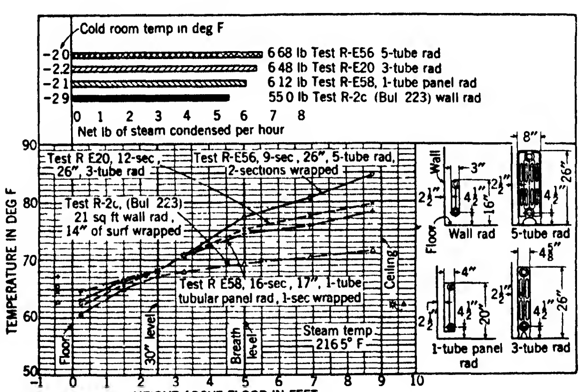


FIG 2. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 30-IN. LEVEL

Note that the steam condensations are different for all four radiators *when the same air temperature of 68 F is maintained at the 30-in. level*

[2]Tests of Radiators with Superheated Steam, by R C Carpenter (A S H.V.E. TRANSACTIONS, Vol 7, 1901, p 206)

[3]The Heating Effect of Radiators, by Dr Charles Brabbeé (A.S.H.V E TRANSACTIONS, Vol 33, 1927, p 33) A.S H V E RESEARCH REPORT No 962—The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A C Willard, A. P Kratz and M K. Fahnestock (A.S.H.V E TRANSACTIONS, Vol 39, 1933, p 303)

The results of tests conducted at the University of Illinois are shown in Figs. 1 and 2[4]. For the four types of radiators shown, the following conclusions are given:

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.
2. Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.
3 The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.
4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room The comfort-indicating temperature should be taken below the breathing line level.
5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath window openings[5].

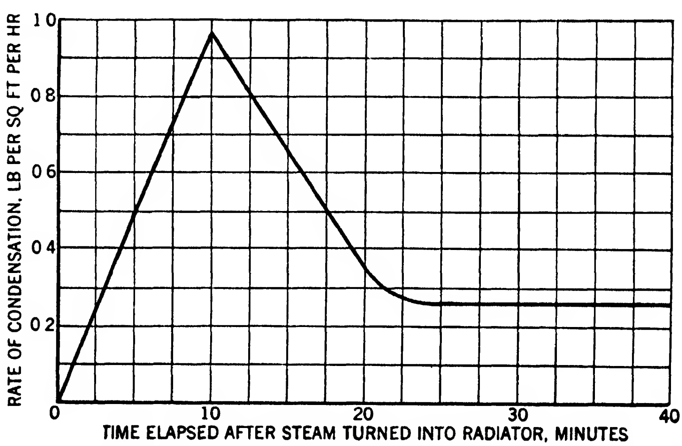


FIG. 3. CHART SHOWING THE STEAM DEMAND RATE FOR HEATING UP A CAST-IRON RADIATOR WITH FREE AIR VENTING AND AMPLE STEAM SUPPLY

HEATING UP THE RADIATOR AND CONVECTOR

The maximum condensation occurs in a heating unit when the steam is first turned on[6]. Fig. 3 shows a typical curve for the condensation rate in pounds per hour for the time elapsing after steam is turned into a cast-iron radiator. The data are from tests on old-style column type radiators. In practice the rate of steam supply to the heating unit while heating up is frequently retarded by controlled elimination of air through air valves or traps. Automatic control valves may also retard the supply of steam. Vacuum types of air venting valves may be used to reduce the length of the venting periods.

[4]A S.H V E RESEARCH REPORT No 905—Steam Condensation an Inverse Index of Heating Effect, by A P. Kratz and M. K Fahnestock (A S H V E TRANSACTIONS, Vol 37, 1931, p 475).

[5]Effect of Two Types of Cast-Iron Steam Radiators in Room Heating, by A. C. Willard and M K Fahnestock (*Heating, Piping and Air Conditioning*, March, 1930, p. 185).

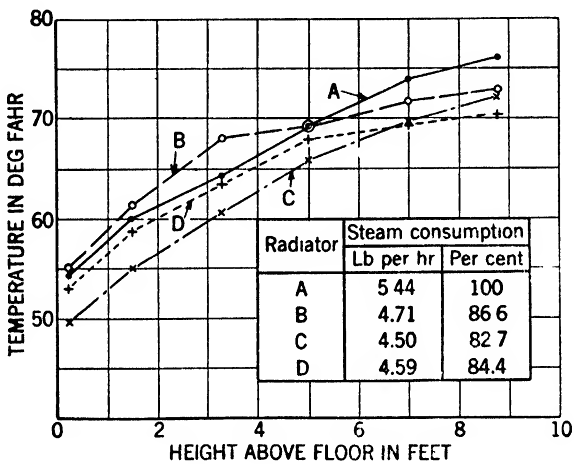
[6]The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P Kratz, M. K. Fahnestock and E L Broderick (A S H V E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1937, p 251)

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed, improve the heating effect. Recent investigations[7] indicate that in the design of the enclosure three things should be considered:

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures
2. The lessened steam consumption may not materially change the radiator heating performance.
3. The enclosed radiator may inadequately heat the space

A comparison between a bare or exposed radiator (*A*) and the same radiator with a well-designed enclosure (*B*), with a poorly-designed enclosure (*C*), and with a cloth cover (*D*) will illustrate the relative



Radiator	Steam consumption Lb per hr	Steam consumption Per cent
A	5 44	100
B	4.71	86 6
C	4.50	82 7
D	4.59	84.4

FIG. 4. STEAM CONSUMPTION OF EXPOSED AND CONCEALED RADIATORS

heating effects. In Fig. 4 the curve (*B*) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same heating effect. Curve (*C*) shows the unsatisfactory effects produced by improperly designed enclosures. Curve (*D*) shows that the effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

Practically all commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[8] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when the radiator is steam hot and the relative humidity

[7]University of Illinois, ***Engineering Experiment Station Bulletins*** Nos 192 and 223, and Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M K Fahnestock and S Konzo (A S H V.E TRANSACTIONS, Vol 35, 1929, p 77).

[8]University of Illinois, ***Engineering Experiment Station Bulletin*** No. 230, p 20.

in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

CONVECTORS OR CONCEALED HEATERS

Although any standard radiator may be concealed in a cabinet or other enclosure so that the greater percentage of heat is conveyed to the room by convection thereby resulting in a form of gravity convector, generally better results are obtained with specially designed units which permit a free circulation of a larger volume of air at moderate tempera-

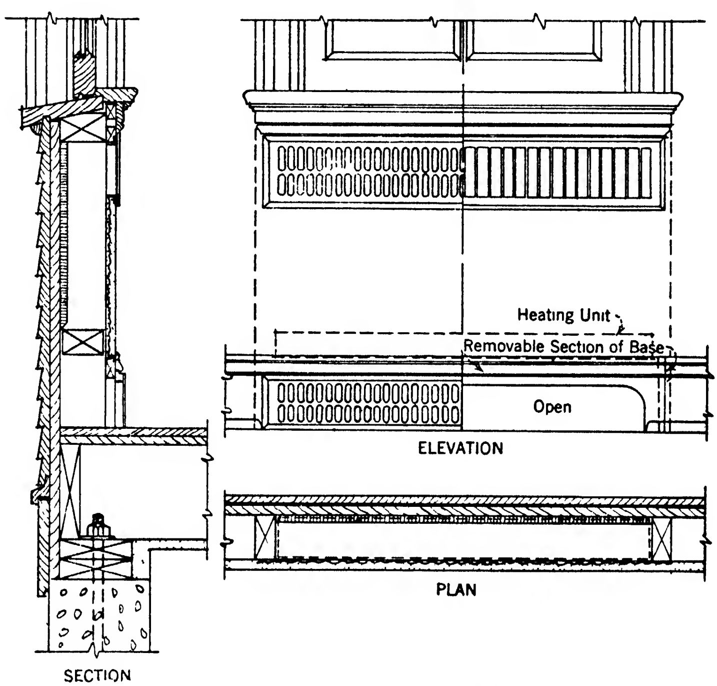


FIG. 5. TYPICAL CONCEALED CONVECTOR USING SPECIALLY DESIGNED HEATING UNIT

tures. Since air stratifies according to temperature, moderate delivery temperatures at the outlet of the enclosure reduce the temperature differential between the floor and ceiling and accordingly accomplish the desired heating effect in the living zone.

Fig. 5 shows a typical built-in convector. The heating element consisting of a large percentage of fin surface is usually shallow in depth and placed low in the enclosure in order to produce maximum chimney effect in the enclosure. The air enters the enclosure near the floor line just below the heating element, is moderately heated in passing through the core and delivered to the room through an opening near the top of enclosure. Since the air can only enter the enclosure at the floor line, the cooler air in the room which always lies at this level, is constantly being

withdrawn and replaced by the warmer air. This air movement accomplishes the desired reduction in temperature differentials and assures maximum comfort in the living zone.

The *Convector Manufacturers Association* has adopted the A.S.H.V.E. Standard[9] in the formulation of its ratings and has compiled a tentative standard of heating effect allowances for various enclosure heights to be included in the ratings by its members.

All published ratings bearing the title *C.M.C.* Ratings (*Convector Manufacturers Certified Ratings*) indicate that the convectors have been tested in accordance with the A.S.H.V.E. Code by an impartial and disinterested laboratory and that the ratings have been approved by the Standardization Committee of the *Convector Manufacturers Association.*

Concealed heaters or convectors are generally sold as completely built-in units. The enclosing cabinet should be designed with suitable air inlet and outlet grilles to give the heating element its best performance. Tables of capacities are catalogued for various lengths, depths and heights, and combinations are available in several styles for installations, such as the wall-hung type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille in a plane parallel with the floor, although the front outlet is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector. It is important that the enclosure or housing for the convector fit as snugly as possible so that the air to be heated must pass through the convector and cannot be by-passed in the enclosure.

The output of a convector, for any given length and depth, is a variable of the height. Published ratings are generally given in terms of equivalent square feet, corrected for heating effect. However, an extended surface heating unit is entirely different structurally and physically from a direct radiator and, since it has no area measurement corresponding to the heating surface of a radiator, many engineers believe that the performance of convectors should be stated in Btu's. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation. When more than one heating unit is used, one mounted above the other in the same cabinet, the output of the upper unit or units will be materially less than that of the bottom unit

RADIATOR AND CONVECTOR SELECTION

The capacity of a radiator varies as the 1.3 power, and that of a convector[10] as the 1.5 power of the temperature difference between the heating medium and the surrounding air in the case of the radiators, and the entering air in the case of the convector. It is obvious that for conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the

[9]A S H V E Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam). (A S H V E TRANSACTIONS, Vol 37, 1931, p 367); (Hot Water), (A S H V E TRANSACTIONS, Vol 39, 1933, p. 237)

[10]A S H V E RESEARCH REPORT No 998—Factors Affecting the Heat Output of Convectors, by A P Kratz, M K Fahnestock, and E L Broderick (A S H V E TRANSACTIONS Vol 40, 1934, p 443,

TABLE 4. CORRECTION FACTORS FOR DIRECT CAST-IRON RADIATORS AND CONVECTOR HEATERS[a]

Steam Press Approx: Gage Vacuum In Hg	Steam Press Approx: Abs Lb per Sq In	Heating Medium Temp. F Steam or Water	Factors for Direct Cast-Iron Radiators — Room Temperature F: 80	75	70	65	60	55	50	Factors for Convectors — Inlet Air Temperature F: 80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Lb per Sq In																
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0.95	0.91	0.87	0.83	0.79	0.76
15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	0.56	0.54	0.53	0.51	0.49	0.48	0.47

[a]To determine the size of a radiator or a convector for a given space, divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table

To determine the heating capacity of a radiator or a convector under conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector divide the heating capacities at the basic conditions by the proper factor from the above table

inlet air temperature at 65 F in the case of a convector, the heat emission will be other than 240 Btu per square foot of rating.

Table 4 shows factors by which radiation requirements, as determined by dividing heat load by 240, shall be multiplied to obtain proper radiator or convector sizes from published rating tables for room temperatures ranging between 50 and 80 F as well as for steam or water temperatures from 150 to 300 F. For other room and heating medium temperatures the factor is determined by the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{t_s - t_r}\right)^{1.3}$$

For convectors:

$$C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5}$$

where

C_s = correction factor
t_s = steam temperature, degrees Fahrenheit
t_r = room temperature, degrees Fahrenheit
t_i = average inlet air temperature, degrees Fahrenheit

CODE TEST FOR RADIATORS AND CONVECTORS

As previously indicated, the output of radiators and convectors is still designated by the terms of older practice, but this is gradually giving place to an engineering method of designating heat emission. The A.S.H.V.E. has adopted the following standards: Code for Testing Radiators (1927); Codes for Testing and Rating Concealed Gravity Type Radiation (Steam), 1931, and (Hot Water), 1933, see also (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 38).

For steam services the actual condensation weight is taken without any allowance for heating effect; for hot water services the weight of circulated water is used without allowance for heating effect. In all cases the total heat transmission varies as the 1.3 power for radiators[11] and the 1.5 power for convectors[12] of the temperature difference between that inside the radiator and the air in the room, and is expressed in Btu or *Mb* per hour.

Standard test conditions specify either a steam pressure of 1 lb gage 15.6 lb per sq in. absolute (215 F) or an average hot water temperature of 170 F and a room temperature of 70 F (5 ft above floor) for radiators, or an inlet air temperature of 65 F for convectors. The heating capacity of a *steam radiator* or *steam convector* is determined as follows:

$$H_t = W_s h_{fg} \tag{1}$$

where

H_t = Btu per hour under test conditions.
W_s = condensation in pounds per hour.
h_{fg} = latent heat in Btu per pound.

H_t may be converted to standard conditions of code ratings by using the proper correction factor from the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{T_s - T_r}\right)^{1.3} = \left(\frac{145}{T_s - T_r}\right)^{1.3} \tag{2}$$

For convectors:

$$C_s = \left(\frac{215 - 65}{T_s - T_i}\right)^{1.5} = \left(\frac{150}{T_s - T_i}\right)^{1.5} \tag{3}$$

The output under standard conditions will be:

$$H_s = C_s H_t \tag{4}$$

where

C_s = correction factor.
T_s = steam temperature during test, degrees Fahrenheit.
T_r = room temperature during test, degrees Fahrenheit
T_i = inlet air temperature during test, degrees Fahrenheit.
H_s = heat emission rating under standard conditions, Btu per hour.

Similarly, for *hot water convectors*, the output under test conditions may be determined as follows:

$$H = W(\theta_1 - \theta_2)\frac{3600}{t} \tag{5}$$

where

H = Btu per hour under test conditions.
W = pounds of water handled during test.
θ_1 = average temperature of inlet water, degrees Fahrenheit
θ_2 = average temperature of outlet water, degrees Fahrenheit.
t = duration of test, seconds.

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[11]Loc. Cit. Note 9.
[12]Loc Cit. Notes 9 and 10

To convert test results to standard conditions, the following correction factor is used:

$$C = \left(\frac{170 - 65}{\frac{\theta_1 + \theta_2}{2} - T_1}\right)^{1.5} = \left(\frac{105}{\frac{\theta_1 + \theta_2}{2} - T_1}\right)^{1.5} \qquad (6)$$

It has been shown that when the exponent 1.5 is used the range of error is less than 3 per cent[13] for convectors.

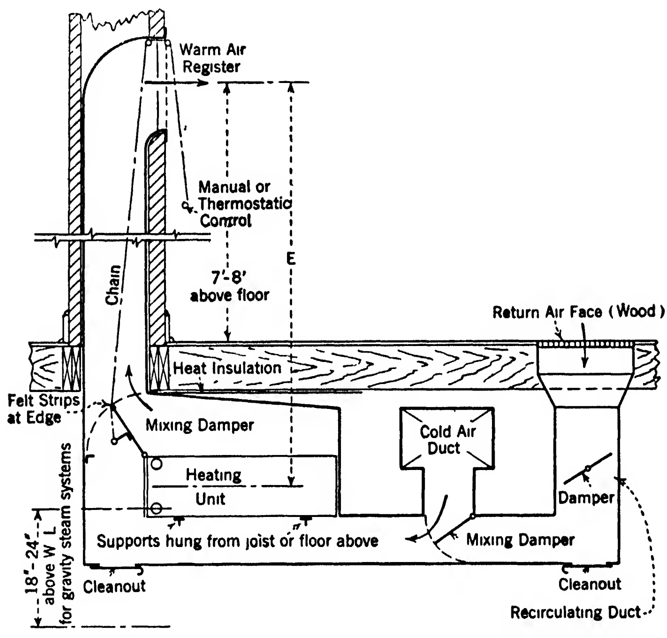


FIG. 6. GRAVITY-INDIRECT HEATING SYSTEM

GRAVITY-INDIRECT HEATING SYSTEMS

The heating units for this system are usually of the extended surface type for steam or hot water, and are installed about as shown in Fig. 6. The temperature and volume of the air leaving the register must be great enough so that in cooling to room temperature the heat available will just equal the heat loss during the same time. To establish and maintain a constant heat flow, provision must be made for removing the air in the room, after it has cooled to the desired room temperature, by a system of vent flues or ducts. As the air flow is maintained by natural draft and this gravity head is very slight, it is necessary to make all ducts as short as possible, expecially the runs from the heating units to the base of the vertical warm air flues. Gravity-indirect arrangements, such as illus-

[13]Loc Cit. Note 10.

trated in Fig. 6, are not to be generally recommended for hot water systems unless the water temperature can be maintained at a reasonably high temperature and rapid circulation of the water can be obtained.

PROBLEMS IN PRACTICE

1 ● What are the principal differences between a radiator and a convector?

A radiator is commonly thought of as a commercial heating unit having a maximum amount of direct heating surface, whereas a convector is a heating device in which the extended or secondary surface may be several times that of the prime surface and which is specially designed to utilize to the fullest extent the convection principal of heating The radiator ordinarily has vertical tubular chambers for the heating medium but most convectors have horizontal tubular chambers to which fins are attached so as to form vertical flues for the passage of air. While radiators are either exposed, enclosed, or shielded, convectors are concealed by means of a tight-fitting enclosure Radiators are commonly made of cast-iron but convectors may be made of a combination of metals, such as copper and brass, or copper and aluminum, as well as entirely of cast-iron

2 ● How did the term heating effect come into use?

It has been found that a room requiring a radiator of a certain determined capacity could under certain conditions be properly heated, with less temperature gradient between floor and ceiling and with less steam condensation, by the same radiator or by one of a different design having the same commercially rated capacity. This resulted in the use of the term *heating effect* to apply to the useful heat output of a radiator, in the comfort zone of a room, as related to the total input to the radiator.

3 ● On what basis are the capacities of convectors published?

Published ratings of convectors are expressed in equivalent square feet of direct cast-iron radiation. Some manufacturers have increased their ratings by as much as 30 per cent to allow for a supposed improved heating effect Tests indicate that the credit to be given heating effect is, in all cases, probably less than 10 per cent, and in many cases negligible.

4 ● How are fins of convectors attached to the tubes or prime surface?

Tubes or a solid core may be forced through piercings in the fins under pressure, or the tubes may be expanded into the holes through the fins. In addition a metallic bonding agent is sometimes used to insure permanent contact.

5 ● What is the procedure in selecting a convector when the required amount of radiation is known?

First the limiting factor or factors of the enclosure must be determined so the available size of the wall recess can be found Manufacturers' catalogs show capacities of convectors of each standard length and depth with varying enclosure heights. From these capacity tables, the proper convector of the required capacity can be selected for the available wall recess. If all three dimensions of the wall recess are insufficient to accommodate a convector of the required capacity, the available height and length can be maintained, but greater depth can be obtained by using a partially recessed enclosure

6 ● Given a room to be heated to 80 F with outside temperature at 0 F, assume the heat loss under these conditions to be 10,000 Btu per hour. Determine the size of the steam radiator to be installed.

A square foot of radiation is equivalent to a heat emission of 240 Btu per hour under standard conditions of steam at one pound gage pressure (215 F) and surrounding air at 70 F. With surrounding air at 80 F, the heat emission from a radiator will be less Under these conditions, the heat emission will not be 240 Btu per square foot of catalog rating per hour, but 240 C_s.

$$C_s = \left(\frac{t_s - t_i}{215 - 70}\right)^{1.3} = \left(\frac{215 - 80}{215 - 70}\right)^{1.3} = 0.912,$$

and 240 C_s = 240 × 0.912 = 218.5 Btu. Therefore, the size of the radiator to be selected shall have a catalog rating of 10,000 divided by 218.5 or 45.8 sq ft.

Chapter 15

STEAM HEATING SYSTEMS

Gravity and Mechanical Return, Gravity One-Pipe Air-Vent, Gravity Two-Pipe Air-Vent, Air Line Heating, One-Pipe Vapor, Two-Pipe Vapor, Atmospheric, Condensation Return, Vacuum, Sub-Atmospheric, Orifice, Zone Control, Condensation Return Pumps, Vacuum Heating Pumps, Traps

THE essential features of the common types of steam heating systems are described in this chapter together with some of the characteristics which influence their successful design and operation. The combination of equipment and piping by which steam is used for space heating, or to warm air for use in ventilating or air conditioning, is known as a steam heating system. They may be classified according to (*a*) the piping arrangement, (*b*) the service performed, such as the *split system* where direct radiators are used for space heating and the heat exchanging units are only used for central fan ventilating or air conditioning, (*c*) the accessories used, (*d*) the method of returning the condensate to the boiler, (*e*) the method of expelling air from the system, or (*f*) the type of control employed. The above classifications are used both where a boiler is included in the system and where the steam supply is from a district heating system.

After the selection of the most suitable type of steam heating system is made on the basis of its operating characteristics, the design of the system should be considered under four headings, namely, (1) determination of load and selection of heating units, (2) the arrangement of the general piping scheme, (3) the sizing of the piping, and (4) the details of connections. Specific information concerning the design and layout of steam heating systems will be found in Chapter 16.

GRAVITY AND MECHANICAL RETURN

When systems are classified according to the method of returning the condensate from the system to the boiler they are known as *gravity* or *mechanical* systems. In *gravity systems* the condensate is returned to the boiler by gravity due to the static head of water in the vertical portion of the return pipes or mains. The elevation of the boiler water line must be sufficiently below the lowest heating unit, steam pipe or dry return pipe to permit the return by gravity. The *water line difference* forming the *static head* must be sufficient to overcome the maximum pressure drop in the system, including the pressure drop due to the condensing effect of the radiation. When radiator and drip traps are used, as in two-pipe vapor

systems, the static pressure must also exceed the operating pressure of the boiler. The pressure drop caused by condensing rate of the radiation is especially important during those portions of the operating periods where changing pressure conditions prevail, as for example, when the system is being initially filled with steam. In systems where the condensate is wasted to the sewer no water line difference is required as is the case with closed systems. However, the waste of condensate may introduce conditions which warrant the use of an appropriate mechanical system. Whenever the conditions of a heating system are such that the returns from the radiation cannot gravitate to the boiler they must be returned by some mechanical means.

In *mechanical systems* the condensate flows to a receiver by gravity and is then forced into the boiler against its pressure. In all instances the

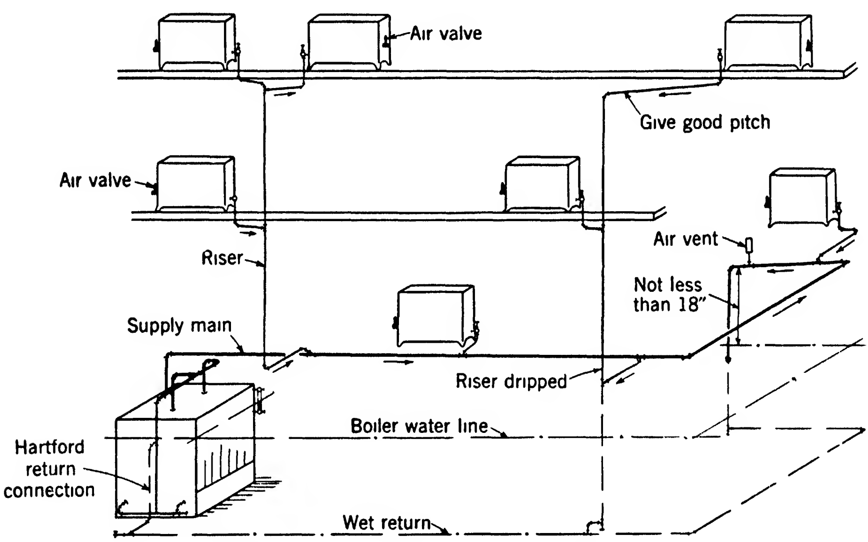


FIG. 1. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

preferable practice is to provide for gravity flow even where a vacuum pump is used. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system, and as long as this condition is obtained, the relative elevation of the boiler water line is unimportant.

There are three general types of mechanical return devices in common use, namely, (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum return line pump.

GRAVITY ONE-PIPE AIR-VENT SYSTEM

This system is the most common of all methods of steam heating, especially for small size installations, due largely to its low cost and simplicity. With larger sized systems the cost of one-pipe systems may

be greater than what are regarded as more expensive systems because of the increased cost of labor and materials for the relatively larger pipe sizes required. As seen by Fig. 1, the steam piping rises to a point as high as possible, at the boiler, and pitches downward from this location until the far end of the main is reached. At the low points and the far ends, drips are taken off and sealed below the water line before being connected to other drips and brought back to the boiler through a wet return. The radiators are supplied with steam by risers branched off the main or mains, the steam passing up the riser and the condensation flowing down it. In the riser, steam and condensation flow in opposite directions, but after the condensation enters the steam main it flows in the same direction as the steam and is removed from the main through the drip. Short mains are sometimes arranged for the condensate to flow in a direction opposite the steam by sizing them so the critical velocity is not exceeded. In buildings of several stories it is customary to drip the heel of each riser to avoid counter-flow of the steam and condensate in the riser spring piece so as to improve steam circulation, but in buildings of one or two stories

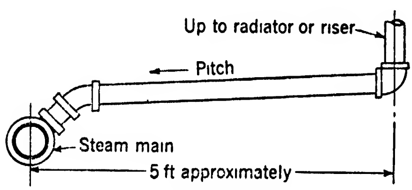


FIG. 2. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

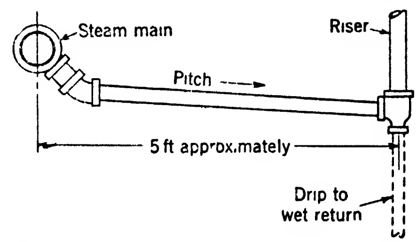


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

the condensate is carried back into the steam main instead of being emptied into the wet return through a riser drip. Both types of risers are shown in Fig. 1.

Horizontal branches to radiators and risers should be pitched not less than ½ in. per foot downward toward the riser or vertical pipe. Horizontal branches from steam mains should be graded at least this amount toward the main, except where the heel of the riser is dripped. Where the heel of the riser is dripped the direction of pitch should be toward the riser drip. (Figs. 2 and 3). Drips carried above the boiler water line or overhead should grade toward the boiler. They must be water sealed before being connected to another dry drip. Wet drips should preferably pitch toward the boiler so they may be completely drained if the building is to be left unoccupied in freezing weather. Wet returns or drips need not be pitched toward the boiler to maintain steam circulation and may be pitched in the opposite direction. In any case provision for drainage is important.

As the one-pipe system is relatively sluggish because of the slowness with which air is released from the system, provision should be made to promote it by providing air vents at the ends and at intermediate points where the steam main is brought to a higher elevation. Venting at drip point of mains is especially important. It is desirable to install the air-vent valves about a foot ahead of the drips, as indicated in Fig. 1, to

prevent possible damage to their mechanisms by water as may occur if the valves are installed directly above the drips. The air valves may be manual or automatic, with or without a check. Air-vent valves with checks prevent the ready re-entrance of expelled air.

The radiator valves may be the angle-globe, corner pattern or gate type. Straight-globe type should not be used since the damming effect of the raised valve seat would interfere with the flow of condensation through the valve. Graduated valves cannot be used since the steam valves on this system must be fully open or fully closed to prevent the radiators filling with water and creating a dangerous water line condition. An objection to the one-pipe steam system is that the heat cannot be regulated at the radiator. Regulating at the radiator is possible only by having the heat *all on* or *all off*, or by setting the valve in an intermediate position. Improved systems and devices are now available which make it possible to obtain a modulating effect from one-pipe heating systems. This is accomplished by the use of special *one-pipe regulating plates* and automatic control of the rate of steam supply which permits varying rates of steam supply and gives fair control during average and severe winter weather.

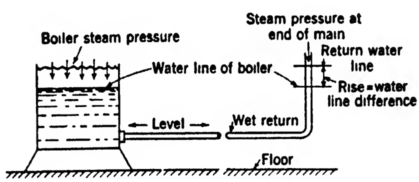


FIG. 4 DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

It is important to keep the lowest points of the steam mains and heating units sufficiently above the water line of the boiler to prevent flooding. Usually a distance of 18 in. is sufficient but construction limitations frequently make shorter distances necessary. The minimum distance which may be used can be checked in the following manner:

Referring to Fig. 4 it will be seen that the water in the wet return is really in an inverted siphon, or U-shaped container, with the boiler steam pressure on the top of the water at one end and the steam main pressure on the top of the water at the other end The difference between these two pressures is the *pressure drop* in the system, *i e.*, the friction of the steam in passing from the boiler to the far end of the main and the pressure reduction in consequence of the condensation occurring in the system. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise enough farther to produce a flow through the return into the boiler (usually about 3 in unless the pipes are small or full of sediment), and it will rise still farther if a check valve is installed in the return so as to obtain sufficient head to lift the tongue of the check (usually 4 in. will be necessary).

If a one-pipe steam system is designed, for example, for a total pressure drop of ⅛ lb, and utilizes an Underwriters' Loop instead of a check valve on the return, the rise in the water level at the far end of the return due to the difference in steam pressure would be ⅛ of 28 in , or 3½ in Adding 3 in. to this for the flow through the return main and 6 in. as a factor of safety gives 12½ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of ½ lb, and with a check in the

return, would require ½ of 28 in., or 14 in., for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

The overhead down-feed system varies basically only from the up-feed systems in the location of the steam main. The steam supply is taken from the boiler and carried to the top of the building, as near the boiler as possible as shown in Fig. 5. If the run to the main riser is long, or the

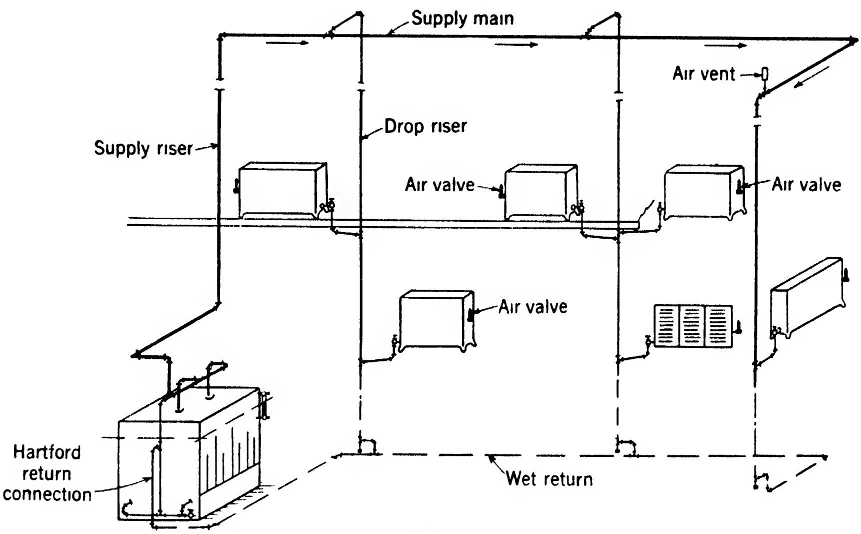


FIG. 5 TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

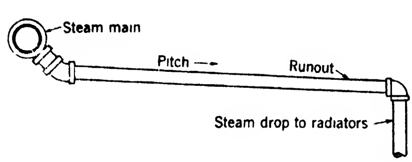


FIG. 6. STEAM RUNOUTS DRIPPING MAIN

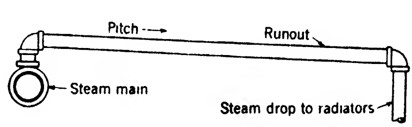


FIG. 7. STEAM RUNOUTS WITH MAIN DRIPPED AT END ONLY

riser extends several stories, the bottom of the riser should be dripped into the wet return. The horizontal main extends from the top of the riser and grades down from it toward all the drops or down-feed risers. The connections to the risers are taken from the bottom of the main, and each drop carries its share of the main's condensation (Fig. 6). All of the drops, except the last, may be taken from the top of the main (Fig. 7), the last drop being from the bottom and serving to drain the entire main. The overhead main does not carry condensation from the radiators The air vent may be located on the main before the last drop (Fig. 5) but the preferable location is at the bottom of the drop below the last radiator

connection and sufficiently above the water line of the boiler to prevent flooding.

The radiators, radiator valves, air valves and radiator runouts, as far back as the risers, are arranged the same as for the *up-feed system.*

GRAVITY TWO-PIPE AIR-VENT SYSTEMS

The gravity two-pipe system indicated in Fig. 8, is now considered obsolete although many of these systems are still in use in older buildings. The same general principles governing its piping design are used when connecting radiators as in other types of gravity systems where they must discharge their condensation to the wet return pipe. Separate supply and return mains and connections are required for each heating unit. Radi-

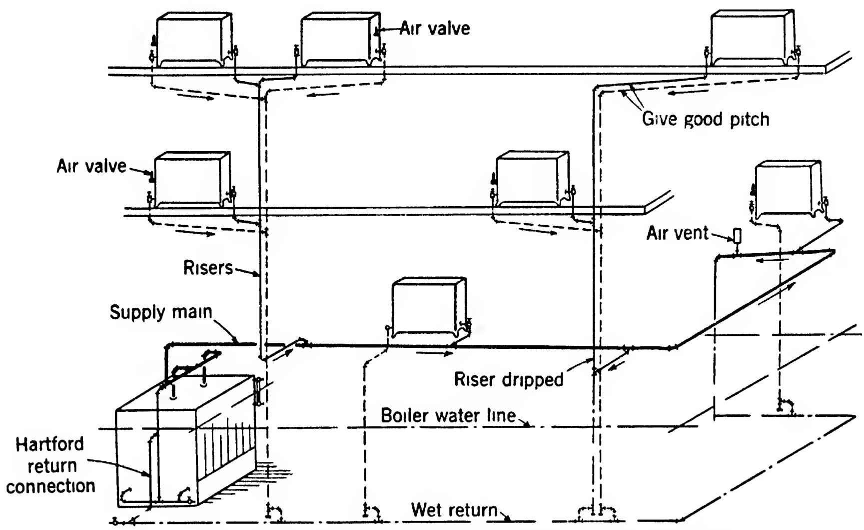


FIG. 8. TYPICAL UP-FEED GRAVITY TWO-PIPE AIR-VENT SYSTEM

ator valves are required in both the supply and return connection to the radiator, and air valves are installed on the heating units and the mains. The radiator valves are the same as described for the gravity one-pipe air-vent system. The steam main is run and pitched in the same manner as in the one-pipe system, but the returns from each radiator are connected into a separate return line system which has the base of the risers sealed either by carrying them down to a wet return below the boiler water line or by a water seal. Where the return has to be kept high to function as a dry return, it is advisable to connect the return risers to the dry return through water seals about 36 in. deep, as shown in Fig. 9, to prevent steam from one riser or radiator entering another through the outlet tapping and closing the air valves on the nearest radiator.

Down-Feed Gravity Two-Pipe System

The steam main in the down-feed system is carried to the top of the building, and the piping of the steam side is arranged practically as in the

down-feed one-pipe gravity system. The drips at the bottoms of the steam drops and the runouts to the radiators are similar to those shown in Fig. 8 for the up-feed gravity two-pipe system. On the return side of the system, the piping is arranged in exactly the same manner as the up-feed gravity two-pipe system.

AIR LINE HEATING SYSTEMS

Both one-pipe and two-pipe systems are at times provided with air valves, which instead of venting to the atmosphere direct, vent to a return pipe of small size. This pipe is then finally vented to atmosphere or connected to a vacuum pump. These are known as *one-pipe* and *two-pipe* air line systems depending upon the omission or provision of a return pipe for the condensate. Where the air line is exhausted by a vacuum pump

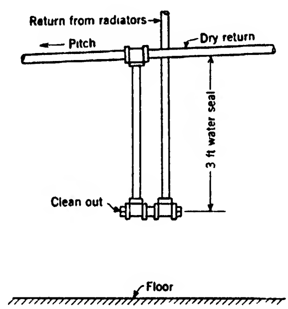


FIG. 9. METHOD OF CONNECTING TWO-PIPE GRAVITY RETURNS TO DRY RETURN MAIN

they are most generally termed *one-pipe* or *two-pipe vacuum air line systems*. Actually the one-pipe air line system has two pipes, the supply pipe and the air pipe (line) while the two-pipe air line system actually has three pipes. Such systems, when making use of a vacuum pump have, in the past, been termed *vacuum systems*. The history of steam heating reveals that the term *vacuum* has been very loosely used.

ONE-PIPE VAPOR SYSTEM

The one-pipe vapor system operates under pressures at or near atmospheric and returns its condensation to the boiler by gravity. In this system the automatic air valves are of special design to permit the ready release of air and prevent its ready return after it is expelled and the steam radiator valves are a type which, when opened, give a free and unobstructed passageway for water. The piping is the same as for the one-pipe gravity system so designed as to permit operation at a few ounces pressure.

TWO-PIPE VAPOR SYSTEM

Two-pipe vapor systems use separate supply and return pipes. In present practice the radiators discharge their condensation through ther-

mostatic traps to the dry return. They operate at a few ounces pressure and above, but those with mechanical condensate return devices may operate at pressures upward of 10 lb atmospheric pressure. Other essentials are packless graduated valves on the radiators, thermostatic traps to drip the ends of steam mains or to vent them where they are dripped to a wet return, and means of venting the system to the atmosphere. In the simplest cases the vent consists of a ¾ in. pipe with a check valve opening outward. Certain systems employ special patented forms of vent valves, designed to allow the air to readily pass out of the system and to prevent its return. These systems contemplate control of the radiator by varying the opening of the graduated radiator valves. The boiler pressure is

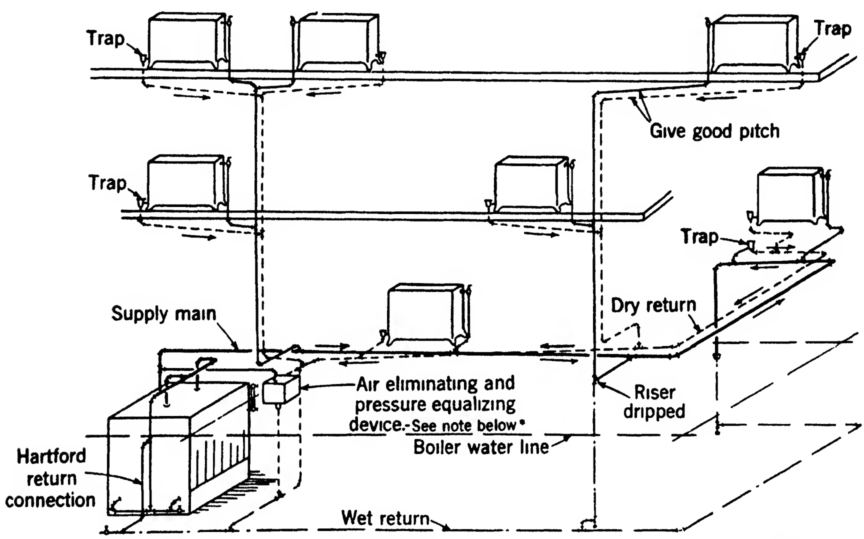


FIG. 10. TYPICAL UP-FEED VAPOR SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination

maintained at substantially constant pressure slightly above but close to atmospheric pressure.

These systems may be classified as (1) *closed systems*, consisting of those which have a device to prevent the return of air after it has once been expelled from the system, and which can operate at both super and subatmospheric pressures for a period of four to eight hours depending upon the tightness of the system and rate of firing, and (2) *open systems*, comprising those which have the return line constantly open to the atmosphere without a check or other means to prevent the return of air. The open systems are not so popular because they have the disadvantage of not holding heat when the rate of steam generation is diminishing. The two-pipe vapor system is shown in Fig. 10. Systems of this design should preferably be equipped with an automatic return trap to prevent water from backing out of the boiler and when so equipped have sometimes been known as *return heating systems*. In installing the return trap a check valve is inserted in the return main at a point near the boiler and a

vertical pipe is run up into the bottom of the return trap, which is usually located with the bottom about 18 in. above the boiler water line. Some traps are constructed so that they will operate when they are installed with their bottom as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler. Fig. 11 shows a typical connection for a automatic return trap.

Up-Feed Two-Pipe Vapor System

In the up-feed system the supply piping is carried to a high point directly at the boiler and is graded down toward the end or ends of the

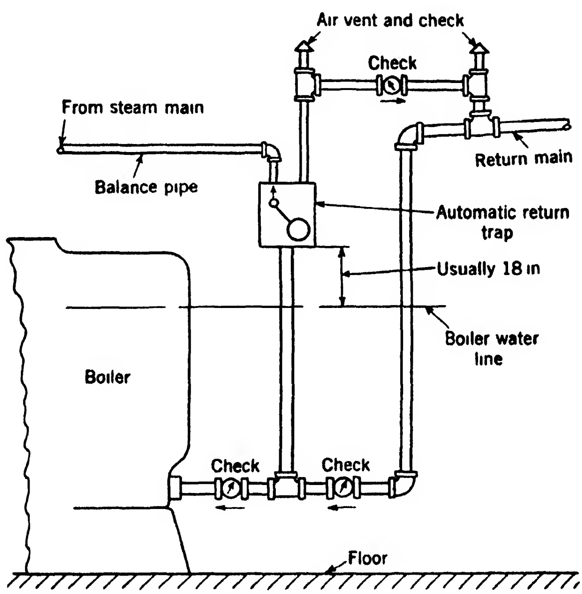


FIG. 11. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

supply main, each supply main being dripped at the end into the wet return or carried back to a point near the boiler where it drops down below the boiler water line and becomes a wet return. From this main, runouts are branched off to feed risers or radiator above, these being graded back toward the steam main; they are not dripped at the bottom of the riser, or toward the riser if the riser heel is dripped. Both types of connections are illustrated in Figs. 2 and 3.

The return risers are connected to each radiator on its return end through thermostatic traps. These are connected to the return main through runouts which slope toward the return main. The return main is run dry and slopes back toward the boiler. Wet returns, if used, are installed in the same manner as for one-pipe gravity systems. If the dry return becomes a wet return it should be vented at that point.

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The branches are taken off the main from the bottom or at a 45-deg angle downward, with the runouts sloped toward the drops (Fig. 6). Thus each branch from the main forms a drip and no accumulation of water is carried down any one drop. Another method of running the steam main, which is not considered as satisfactory but which is practical, is to take the branches off the top of the main (Fig. 7) and to drip the end of the main through the last riser, as illustrated in the down-feed one-pipe system detail shown in Fig. 6. If this is done, the pipe drop at the end or ends of the mains should be

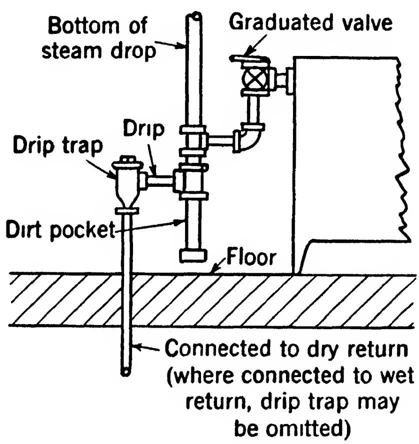


FIG. 12 DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

enlarged one pipe size to provide capacity for this concentration of the main drip.

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small and the normal size of drop required is 1 in. or less. The bottom of the steam drops should terminate with a dirt pocket as shown in Fig. 12. The returns on a down-feed vapor system are the same as on an up-feed system except that every steam drop must have a drip at the bottom connected either into the return through a trap or into a separate water-sealed drip line below the boiler water line, as illustrated in Fig. 10, in which case the thermostatic traps may be omitted. The runouts to the radiators and the radiator connections of the down-feed system are the same as those for the up-feed system already described.

ATMOSPHERIC SYSTEM

The distinguishing features of the atmospheric system are gravity return to the boiler or to waste, graduated or ordinary radiator valves, no automatic air valves on the radiators, thermostatic traps on the radiator returns, and the venting of all air from the system by means of pipes open to the atmosphere. The returns are open to the atmosphere at all times, usually by extending the return risers to the top of the building where they are either connected together in groups and carried through the roof or extended through the roof individually. Atmospheric systems, either up-feed or down-feed, are often used where the condensation is not

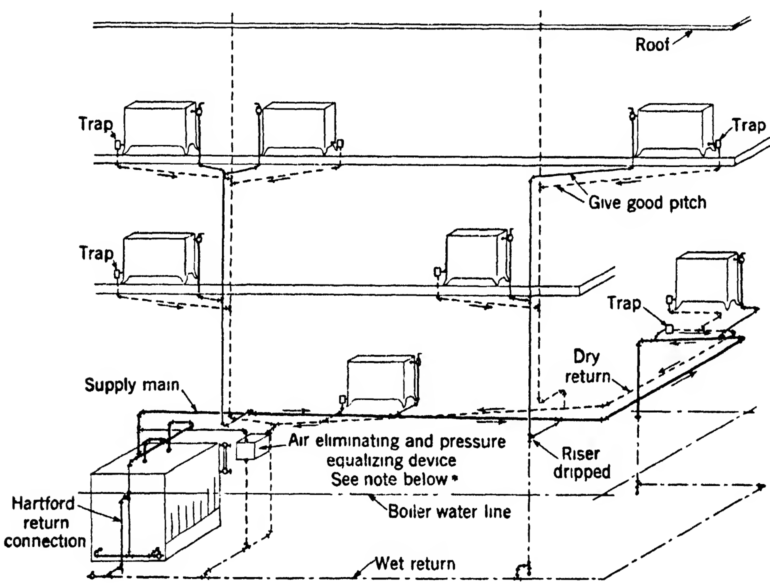


FIG 13. TYPICAL ATMOSPHERIC SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

returned to the boiler, as in heating systems supplied by high pressure steam through pressure-reducing valves at locations far from the boilers. The returns may be delivered back to the boiler, if desired, by condensation return pumps which are vented to the atmosphere. The return lines in such systems are simply gravity waste lines in which the condensation flows entirely by gravity and is not aided by any pressure difference.

Atmospheric systems contemplate maintaining a practically constant pressure in the steam pipe and atmospheric pressure in the return pipe. When graduated steam valves are provided, they enable the occupant of a room to vary the flow area to the radiator so as to obtain a greater or lesser heating effect.

The steam side may be run as that for either up-feed or down-feed two-pipe vapor systems, as the conditions require, and the radiator con-

nections are the same as for vapor systems in that they have graduated valves on the radiator supply ends and thermostatic traps on the radiator return ends. All drips from the supply main and the steam side of the system must pass through thermostatic drip traps before entering the return system where only atmospheric pressure exists. Fig. 13 illustrates a typical scheme of piping used on atmospheric systems. Such systems do not maintain heat in the radiators under declining fires. As the steam supply diminishes, air from the atmospheric re-enters through the open vent pipe retarding the inflow of steam and cooling the radiator.

CONDENSATION RETURN HEATING SYSTEMS

When an automatic condensation return pump is substituted for the gravity return of a two-pipe vapor system they are quite generally known

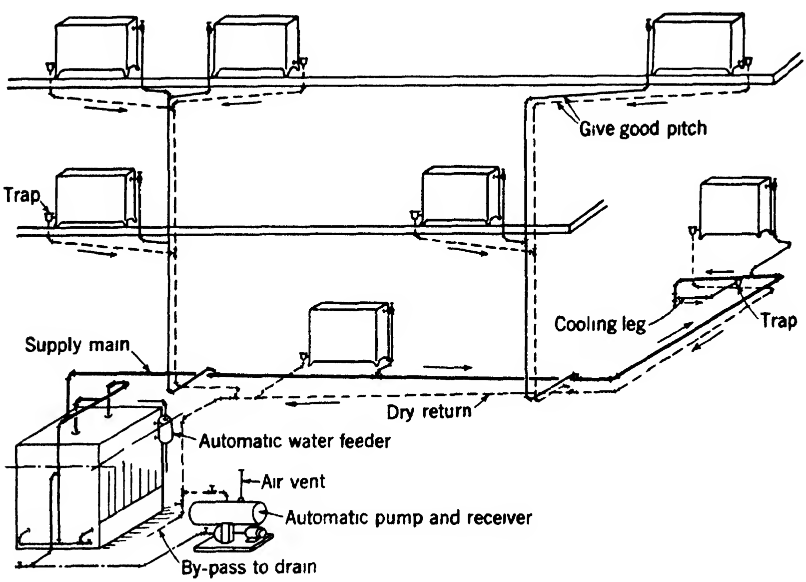


FIG. 14. TYPICAL INSTALLATION USING CONDENSATION PUMP

as *return systems* or *return pump heating systems*. A typical installation of a motor driven automatic condensation unit is illustrated in Fig. 14. It will be noted that the returns are graded to cause flow by gravity to the vented receiver. As the receiver is filled, the float mechanism operates either a pilot or an across-the-line switch to start the pump, and upon emptying the tank disconnects the power and stops it. The pump may be used to deliver the condensate direct to the boiler, to a feed water heater or to raise the water to any higher elevation or pressure than that of the return line. A useful application is to use a small condensation unit to handle a remote section of radiation that otherwise would be difficult to grade to the main return.

VACUUM SYSTEMS

In the vacuum system, a vacuum is maintained in the return line practically at all times but no vacuum is carried on the steam side. The pump is usually controlled by a vacuum regulator which operates the pump to maintain the vacuum within limits and operates in response to a pressure difference between the atmosphere and the return to control the vacuum in the return main. The source of steam supply may be from a low pressure boiler as shown in Fig. 15, or a high pressure line through a pressure reducing valve.

The supply main slopes down in the direction of flow; the runouts pitch down toward the riser if it is dripped (Fig. 3) or upward toward the riser if it is not dripped (Fig. 2). The dripping of the risers depends largely on

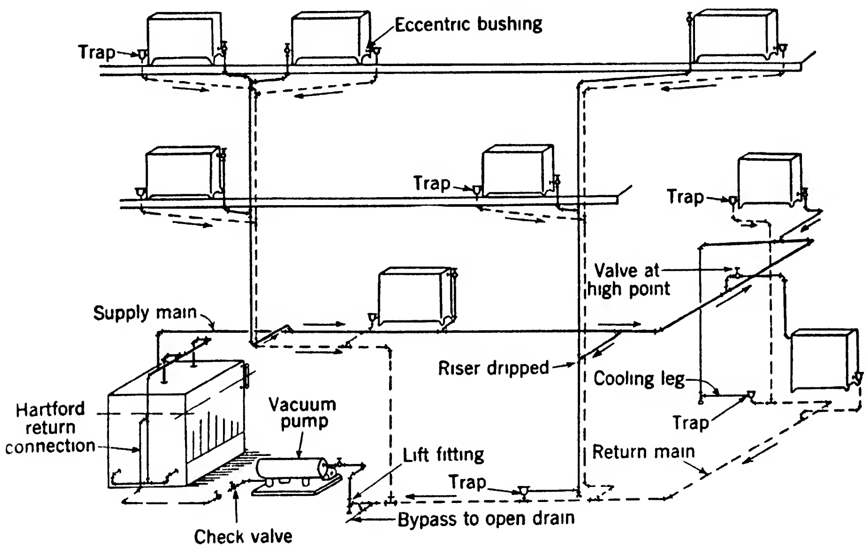


FIG. 15 TYPICAL UP-FEED VACUUM PUMP SYSTEM

the height of the riser. Ordinarily risers less than three stories high are not dripped, while those more than four stories high are usually dripped. When risers are dripped the runouts from the steam main may be taken from the bottom, if desired, and each runout then serves as a drip for the main. The runouts from the dripped risers slope back toward the riser.

Where graduated manual control of the radiators is desired, graduated valves may be used to control the supply of steam to the radiator. Angle or corner pattern radiator valves may be used. The radiators must be drained through thermostatic traps which pass air and water but prevent the passage of steam. Combination float and thermostatic traps are preferable for draining mains and risers.

The return risers are connected in the basement into a common return main which slopes downward toward the vacuum pump. The vacuum

pump withdraws the air and water from the system, separates the air from the water and expells it to atmosphere and pumps the water back to the boiler, or other receiver, which may be a feed-water heater or hot well. It is essential that no connection be made from the supply side to the return side at any point except through a trap. The best practice demands a return flowing to the vacuum pump by an uninterrupted downward slope. In some instances local conditions make it necessary to drop the return below the level of the vacuum pump inlet, before the pump can be reached. In such an event one of the advantages of the vacuum system is the ability to raise the condensate to a considerable height by the suction of the vacuum pump by means of a lift connection or fitting

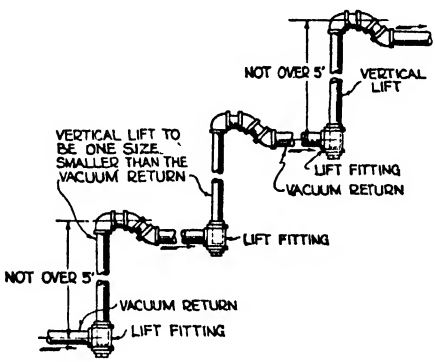


FIG. 16. METHOD OF MAKING LIFTS ON VACUUM SYSTEMS WHEN DISTANCE IS OVER 5 FT

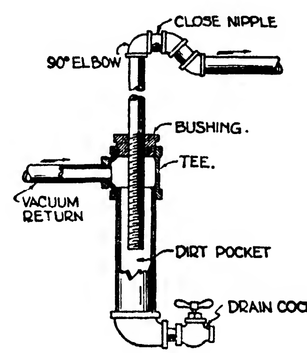


FIG. 17. DETAIL OF MAIN RETURN LIFT AT VACUUM PUMP

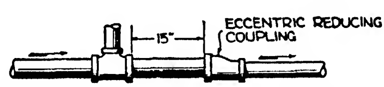


FIG. 18. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

inserted in the return. The height the condensate can be raised depends on the steam pressure and the amount of vacuum maintained. It is preferable to limit lift connections to a single lift at the vacuum pump. A still more preferable arrangement is the use of an accumulator tank, or receiver tank with a float control for the pump, at the low point of the return main located adjacent to the vacuum pump.

When the vertical lift is considerable, several lift fittings should be used in steps (Fig. 16). This permits a given lift to be secured with a somewhat lower vacuum than where the vertical distance is served by a single lift. Where several lifts are present in a given system at different locations, the lifting cannot occur until the entire system is filled with steam. A lift connection for location close to the pump, where the size may be above the commercial stock sizes, is shown in Fig. 17. It is desirable that means be provided for manually draining the low point of the lift fittings to eliminate from the return piping all water in danger of freezing in case the system is shut down for a considerable length of time.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return side of the system is exactly the same as the up-feed system except that the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45-deg angle down from the steam main (Fig. 6) so that they may serve as steam main drips. When this is done it is practical to run the steam main level if a runout is located at every

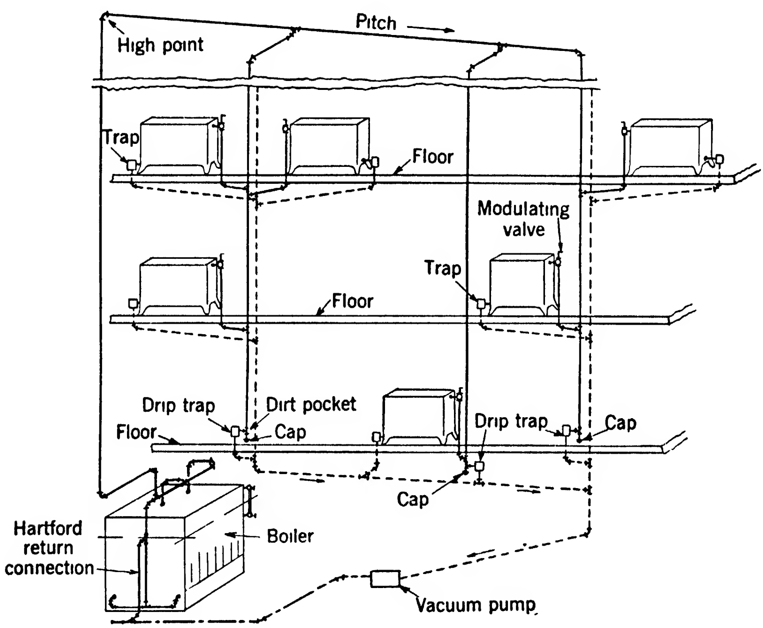


FIG. 19. TYPICAL DOWN-FEED VACUUM SYSTEM

change in pipe size, or if eccentric fittings are used (Fig. 18). A slight pitch in the steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 19.

SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems but, in contrast, provide control of building temperature by variation of the heat output from the radiators. The radiator heat emission is controlled by varying the pressure, temperature and volume of steam in circulation. These systems differ from the ordinary vacuum system in that they maintain a controllable partial vacuum on both the supply and return sides of the system, instead of only on the return side. In the vacuum system,

steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, atmospheric pressure of higher levels exists in the steam supply piping and radiators only during the most severe weather. Under average winter temperature the steam is under partial vacuum which in mild weather may reach as high as 25 in. Hg., after which further reduction in heat output is obtained by partially filling the radiation with steam.

The rate of steam supply is controlled by a control valve in the steam main or by thermostatically controlling the rate of steam production in the boiler. The control valve may be of the automatic *modulating* or *floating* type governed thermostatically from selected control points in the building or it may be a special pressure reducing valve which will maintain the desired sub-atmospheric pressures by continuous flow into the heating main. At all radiator supply tappings are radiator valves incorporating adjustable orifices or equipped with regulating orifice plates. The sizes of orifices used are larger than for orifice systems because for equal radiator sizes the volume flowing is larger. Radiator traps and drips are designed to operate at any pressure from 15 lb gage to 26 in. of Hg. Unit heaters, unit ventilators, cast-iron direct radiators and convectors may be used together in the sub-atmospheric system. A vacuum pump capable of operating at high partial vacua is preferable to promote accuracy in the distribution of steam throughout the system, particularly in mild weather. This vacuum is partially self induced by the condensation of the steam in the system under conditions of restricted supply for reduction of the radiator heat emission. The vacuum pump regulator is a diaphragm device subjected to the difference in pressure between the supply and return sides of the system. It starts the pump when the pressure drop through the system falls to a minimum and stops it when the pressure difference increases to a fixed maximum. The actual pressure difference (drop) maintained is only enough to secure adequate circulation and is often about 2 in. of Hg. The low pressure difference maintained permits the use of a wider range of sub-atmospheric pressure in the radiators and supply pipe and more precise control of distribution. This method of operation results in a diminution in heat output from steam mains and risers (as well as radiators) and gives a measure of control over this portion of the total system output. The decrease in condensation in the piping, as the temperature of the steam is reduced under vacuum, assists in securing control of building temperature and promoting economy. The orifices function to distribute steam proportionately when both complete and partial filling of the radiators is being employed in the cycle of heat output reduction. Individual thermostatic radiator control may be used with this system. With individual thermostatic radiator control the individual control makes fewer operations and the radiator follows a more even temperature without fluctuating from extreme hot to extreme cold, since operating the system with steam temperature inversely with the outside temperature removes a part of the load from the radiator temperature control.

The returns must grade downward constantly and uninterruptedly from the radiator return outlets to the inlet of the receiver of the vacuum pump. One radical difference between this system and the ordinary vacuum system is that no lifts should be made in the return line, except

at the vacuum pump. The receivers may be placed at a lower level than the pump and are equipped with float control so the pump may operate as a return pump during night operation. The system may be operated in the same manner as the ordinary vacuum system when desired.

To convert an ordinary vacuum return line system to a sub-atmospheric system, a control valve is inserted on the steam main near the boiler or the boiler is automatically controlled. The steam supply to each radiator is provided with a flow proportioning device, such as an orifice, a high-vacuum pump is substituted for the ordinary type and is supplied with a pressure difference control, and traps are placed on the radiators and drip points which will operate satisfactorily throughout the wide pressure and vacuum range.

Steam for heating domestic hot water should not be taken from the heating system side of the control valves of this system. It should be taken from the boiler header back of the control valve so that pressures sufficiently high for heating the water may be maintained on the heater. The sub-atmospheric method of heating can be used for the tempering and heating coils of ventilating and air conditioning systems. The flexible control of heat output secured by this method materially reduces the required size of by-pass around the heaters. Sub-atmospheric systems are proprietary.

ORIFICE SYSTEMS

Orifice systems of steam heating may have piping arrangements identical with vacuum systems. Some of these systems omit the radiator thermostatic traps but use thermostatic or combination float and thermostatic traps on all drip points. A return condensation pump with the receiver vented to atmosphere is generally used to return the condensation to the boiler, or place of final disposition, such as a feed water heater or hot well. They contemplate varying the heat emission from the radiators by varying the pressure maintained in the steam supply piping while the radiator pressure remains substantially equal to that of the atmosphere.

The principle on which they operate is based on the well known fact that the steam flow through an orifice will vary when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side, no further increase in flow will be obtained as a result of the increased pressure difference. If an orifice is so designed in size as to exactly fill a radiator with 2 lb gage on one side and $\frac{1}{4}$ lb gage on the other, the absolute pressure relation is:

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 0.90 \text{ or } 90 \text{ per cent.}$$

Should the steam pressure be dropped to $\frac{1}{4}$ lb on the supply pipe, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be apparent that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, reducing this steam main pressure will permit filling various desired portion of the radiator down to the point where the main pressure equals the back pressure in the radiator provided the supply pipe pressures may be controlled sufficiently closely. If orifices are designed on

a similar basis for a given system and proportioned to the heating capacity of the radiators they serve, all radiators will heat proportionately to the steam pressure. The range of pressure variation is limited by the permissible noise level of the steam flowing under the pressure difference required for maximum heat output. The control of the steam supply is obtained by a valve placed in the steam main, which maintains a determined pressure. These valves are frequently manually set from a remote location, guided by temperature indicating stations in the building; or thermostatically controlled from a thermostat on the roof, which automatically measures the differential of outside and inside temperatures; or by a boiler pressure control. Since the range through which the pressures may be varied is limited, usually from 0 to 4.0 lb gage, the control should be capable of maintaining close regulation to maintain the desired space temperatures, particularly in mild weather.

Some systems use orifices not only in radiator inlets but also at different points in the steam supply piping for the purpose of balancing the system to a greater extent. In this manner the difference between the initial and terminal pressure in the steam main may be compensated to a great extent. For example, if the initial pressure was 3 lb gage and the pressure at the end of the main was 2 lb, an orifice could be used in each branch for the purpose of obtaining a more uniform pressure throughout the system. Such a provision may be particularly useful in this system for branches close to the boiler where the drop in the main has not yet been produced Orifice systems are proprietary.

ZONE CONTROL

Often certain portions of a building may require more heat than others even if occupied for the same daily periods and the same maintained temperatures exist. Sometimes an entire building is on one general control, which results in overheating some sections when sufficient heat is supplied to accommodate the coldest portion.

By separating a building into *zones*, each with its own piping system, each *zone* may be controlled separately. Systems are zoned to care for exposure, hours of occupancy, stack effect and the requirements of occupancy activity.

In large buildings it is important to consider zoning for exposure because of the varying effects of the wind and sun. With the prevailing winter winds from the northwest, for example, a simple zoning would place the north and west sides of the building on one zone and the south and east sides on another. If the size of the building justifies the expenditure, a better arrangement would be to place all north walls on one zone, all west walls on a second, all east walls on a third, and all south walls on a fourth. Certain interior areas, such as basements, light well walls and other locations where sun and wind do not affect the conditions, should be placed in still another zone if the most economical operation is to be secured.

In high buildings it is often important to consider zoning for stack or *chimney* effect, caused by the difference in density between the warm air on the inside of a building and the colder air on the outside. Where the lowest eight or ten stories are protected from winds by surrounding buildings, it may accentuate the need for zoning to correct for the *chimney*

effect. On still days the heat demands vertically will vary little, but on windy days there will be a marked difference in the heat requirements for the different horizontal sections at different elevations. An arrangement to provide for difference in heat requirement for exposure and *chimney effect* would give 12 zones; namely, north, east, south, and west lower zones; similar middle zones; and similar top zones. Every type of steam heating system may be zoned. The extent to which any system may be zoned is governed by the limitations of the given installation and the particular type of system. Zone control is used principally with sub-atmospheric, orifice and vacuum heating systems.

Each zone should constitute an individual and separate system with its own control valve (controlled by thermostats in its respective zone), steam supply and return piping and preferably its own return pump or vacuum pump. It is possible for a single vacuum pump to serve several zones with a controller for each section connected in parallel so that the zone in which the lowest differential or vacuum is produced may start the pump.

Zoning has advantages even where individual thermostatic radiator control is installed, whether this be of pneumatic, electric, or the self contained radiator valve type. The control secured by zoning in supplying heat in parallel with its outside temperature and wind fluctuations removes a large part of the load from such individual thermostatic controls; they operate less frequently and the radiators follow a more even temperature instead of fluctuating from extreme hot to extreme cold.

CONDENSATION RETURN PUMPS

Condensation return pumps are used for gravity systems when the local conditions do not permit the condensation to return to the boiler under the existing static head. The return of the condensate permits the water to repeatedly go through the cycle of vaporization, with subsequent condensation and return to the boiler. During such repeated cycles any incrustants or other substances in solution are precipitated and the water de-activated to a considerable extent so that corrosion of a serious nature is seldom ever encountered where the condensate is repeatedly used. Serious corrosion is more frequently found in systems where the condensation is not repeatedly used but is wasted and fresh make-up water is continually being introduced.

The most generally accepted condensation pump unit for low pressure heating systems consists of a motor-driven centrifugal pump with receiver and automatic float control. Other types in use include rotary, screw and reciprocating pumps with steam turbine or motor drive, and direct-acting steam reciprocating pumps.

The receiver capacities of these automatic units should be sized so as not to cause too great a fluctuation of the boiler water line if fed directly to the boiler and at the same time not so small as to cause too frequent operation of the unit. The usual unit provides storage capacity between stops in the receiver of approximately 1.5 times the amount of condensate returned per minute and the pump generally has a delivery rate of 3 to 4 times the normal flow. This relation of receiver and pump size to heating system condensing capacity takes account of the peak condensation rate.

VACUUM HEATING PUMPS

On vacuum systems, where the returns are under a vacuum, and sub-atmospheric systems, where the supply piping, radiation and the returns are under a vacuum, it is necessary to use a vacuum pump to discharge the air and non-condensable gases to atmosphere and to dispose of the condensation. Direct-acting steam driven reciprocating vacuum pumps are sometimes used where high pressure steam is available or where the exhaust steam from the pump can be utilized. In general, however, these have been replaced by the automatic motor-driven return line heating pump especially developed for this service. Steam turbine drive is also frequently used where steam at suitable pressures is available, the steam being used afterward for building heating. The usual vacuum pump unit consists of a compact assembly of exhausting unit for withdrawing the air-vapor mixture and discharging the air to atmosphere and a water removal unit which discharges the condensate to the boiler furnished complete with receiver and separating tank and automatic controls mounted as an integrated unit on one base. There are also special steam turbine driven units which are operated by passing the steam to be used in heating the building through the turbine with only a 2 to 3 lb drop across the turbine required for its operation. Under special conditions such as installations where it is necessary to return the condensate to a high pressure boiler, auxiliary water pumps may be supplied. In some instances separate air and water pumps may be used.

Practically all automatic motor-driven return line vacuum heating pumps make use of a portion of the condensate to operate either as a liquid piston pump or as a kinetic exhauster (which operate on a modified ejector principle) to withdraw the air and condensate from the system, discharge the air to atmosphere and return the condensate to the boiler. Some type of hydraulic action is utilized to produce the suction. Such hydraulic evacuating devices may be classified as:

a. Water ring centrifugal displacement pumps.

b. Water piston pumps.

c. Stationary kinetic exhauster pumps.

d. Rotary kinetic ejector pumps.

The evacuating element is generally combined with a centrifugal water impeller for the delivery of the condensate to the boiler or feed-water heater.

The assembled units may be further grouped under two general classifications:

a. Those which perform the function of air separation under atmospheric pressure.

b. Those which perform the function of air separation under a partial vacuum.

Pumps coming under the first classification remove both the air and condensate from the returns by means of the hydraulic evacuator and deliver both to a separating tank under atmospheric pressure. From this tank the air and non-condensable vapors are vented to atmosphere while the condensate is removed and delivered to the boiler by means of the built-in boiler feed pump impeller.

In the second classification, the air and condensate are first separated under vacuum by means of the receiver which is directly connected to the returns. The hydraulic evacuator withdraws only the air and non-condensable vapors from the top of the receiver and delivers them to atmosphere. The built-in condensate pump impeller removes the condensate from the bottom of the receiver and delivers it direct to the boiler or feedwater heater.

Under special conditions such as returning the condensate to a high pressure boiler or the furnishing of large air removal units for high vacuum systems, it is customary to supply separate motor-driven air and water pumps.

For rating purposes[1] vacuum pumps are classified as *low vacuum* and *high vacuum*. Low vacuum pumps are those rated for maintaining 5½ in.

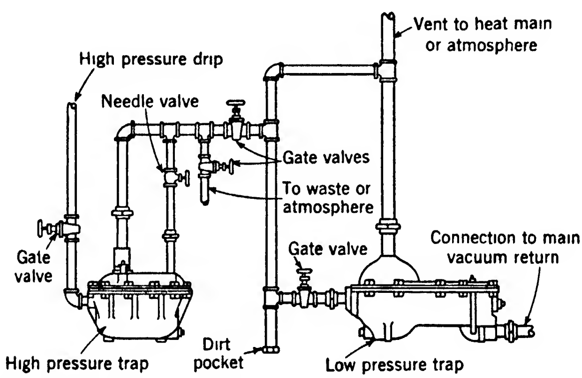


FIG. 20 METHOD OF DISCHARGING HIGH-PRESSURE APPARATUS INTO LOW-PRESSURE HEATING MAINS AND VACUUM RETURN MAINS THROUGH A LOW-PRESSURE TRAP

Hg. vacuum on the system, and high vacuum pumps are those rated to maintain vacuums above 5½ in.

The vacuum maintained in the returns of a system is affected by the steam pressure being carried on the system. Thus when pressures in the upper ranges used in low pressure steam heating are carried, the temperature at which the condensate enters the return piping increases. A portion of the condensate in flowing through the traps, upon entering the lower pressure region of the returns re-evaporates because it is at a temperature higher than the saturation temperature for the pressure existing in the return piping. This vaporization tends to increase the pressure existing in the return piping (reduce the vacuum). The amount vaporizing is proportional to the difference between the heat content of the condensate at the temperature it leaves the radiator and that corresponding to the vacuum in the return piping. This action also frequently results in the inability to produce as high vacuums in systems having covered return piping as would otherwise be the case even though the same steam pressure is carried on the radiation. It is obvious that conditions may be

[1]A S H V E Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps, (A.S H V E TRANSACTIONS, Vol 40, 1934, p 33)

obtained which limit the vacuum the pump can produce even though the traps are in normal operating condition and the system is reasonably tight. It is for this reason the condensate from equipment using steam at high pressures should not be connected directly to a vacuum return line but should drain to a receiver through a high pressure trap. The receiver should have an equalizing connection to a low pressure steam main and drain through a low pressure trap to the vacuum return main as indicated in Fig. 20.

Vacuum Pump Controls

In the ordinary vacuum system the pump is controlled by a regulator which cuts in when the vacuum drops to the lowest point desired and cuts out when the vacuum has been increased to the desired high point to economize on current consumption. The *cut-in* point is usually about 3 in. and the *cut-out* point approximately 8 in. This is done largely to eliminate frequent starting and stopping of the vacuum pump which would otherwise occur without serving any particular purpose in the ordinary vacuum return line system. In addition to this vacuum control, a float control is included which automatically starts the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum in the system. A selector switch is usually provided to allow operation at night as a *condensation* pump only and to permit *continuous* operation if desired.

In sub-atmospheric systems the vacuum pump control maintains a pressure difference between the supply and the return piping which is held within relatively close limits. Such limits permit securing higher vacua on the supply piping as the upper limit of the pump's vacuum is approached. A pilot switch is also provided to enable operation as a *condensation pump* or to give *continuous* operation. In those cases where the condensation returns at a level lower than the pump, an accumulator tank is provided with a float control to start and stop the pump whenever sufficient condensate accumulates.

Piston Displacement Vacuum Pumps

Piston displacement return vacuum heating pumps may be either power or steam driven. They should be provided with mechanical lubricators and their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. They are usually supplied with an air separating tank, open to atmosphere, placed on the discharge side of the pump and at an elevation sufficiently high to allow gravity flow of the condensate to the boiler. If the boiler pressure is too high for such gravity feed then an additional steam pump for feeding the boiler is desirable. The extra pump is sometimes avoided by using a closed separating tank with a float controlled vent. In both arrangements, the air taken from the system must be discharged against the full discharge pressure of the vacuum pump. In the case of high or medium pressure boilers, it is better to use the atmospheric separator and the second pump.

In figuring the required displacement for such pumps, a value of from 6 to 10 times the volumetric flow of condensation is used for average vacuums and systems.

TRAPS

Traps are generally classified as to function as (*a*) separating traps, (*b*) return, lifting or vacuum traps, and (*c*) air traps. Separating traps may be either float operated, thermostatically operated or float and thermostatically operated. Return traps for low pressure service have been referred to previously as *alternating receivers* in this chapter. Return traps may also operate to receive condensate under a vacuum and return it to atmosphere or a higher pressure. Air traps are generally float operated.

Separating traps are used to release water of condensation but to retain steam. The thermostatic, and float and thermostatic types release both condensate and air but retain steam. Separating traps are used for draining condensate from radiators, indirect air heaters, steam piping systems, kitchen equipment, laundry equipment, hospital equipment, drying equipment and many other kinds of apparatus. Air traps release air but retain water. Devices known as *quick vents* and *vent valves* are, in principle, traps which allow the passage of air but prevent the passage of either water or steam.

Return traps are used for returning condensate either by gravity, by steam pressure, or by both, to a boiler or other point of disposal, and for lifting condensate from a lower to a higher elevation, or for handling condensate from a lower to a higher pressure.

The fundamental principle upon which the operation of practically all traps depends is that the pressure within the trap at the time of discharge shall be equal to, or slightly in excess of, the pressure against which the trap must discharge, including the friction head, velocity head and static head on the discharge side of the trap. If the static head is in favor of the trap discharge it is a minus quantity and may be deducted from the other factors of the discharge head.

Traps may also be classified according to the principle of operating device which supplies the power to cause them to function as (1) float, (2) bucket, (3) thermostatic, (4) float and thermostatic, (5) impulse, or (6) tilting traps.

Float Traps. A discharge valve is operated by the rise and fall of a float due to the change of water level in the trap. When the trap is empty the float is in its lowest position, and the discharge valve is closed. A gage glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present.

Bucket Traps. Bucket traps are of two types, the upright and inverted, and although they are both of the open float construction, their operating principle is entirely different. In the *upright bucket* trap, the water of condensation enters the trap and fills the space between the bucket and the walls of the trap. This causes the bucket to float and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water to a discharge opening. When the bucket is emptied it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent.

In the *inverted bucket* trap, steam floats the inverted submerged bucket and closes the

valve. Water entering the trap fills the bucket which sinks and through compound leverage opens the valve, and the trap discharges. It is impossible to install a water gage glass on an inverted bucket trap, but if visual inspection is necessary, a gage glass can be placed on the line leading to the trap. No air relief cocks can be used but this is unnecessary, as the elimination of air is automatically taken care of by air passing through the vent in the top of the inverted bucket regardless of temperature.

Thermostatic Traps. Thermostatic traps are of two types, those in which the discharge valve is operated by the relative expansion of metals, and those in which the action of a volatile liquid is utilized for this purpose. Thermostatic traps of large capacity for draining blast coils or very large radiators are called *blast* traps.

Float and thermostatic traps have both a thermostatic element to release air and a float element to release the water.

Impulse traps operate with a moving valve actuated by a control cylinder. When the trap is handling condensate, the pressure required to lift the valve is greater than the reduced pressure in the control cylinder and consequently the valve opens allowing a free discharge of condensate. As the remaining condensate approaches steam temperature, flashing results, flow through the valve orifice is choked and the pressure builds up in the control chamber closing the valve.

Automatic Return Traps

In the general heating plant, where thermostatic traps are installed on the heating units, it becomes necessary to provide a means for returning the water of condensation to the boiler, if a condensation or vacuum pump is not used. When the return main can be kept sufficiently high above the boiler water line for all operating conditions, the water of condensation will flow back by gravity, and no mechanical device is required. But actually this does not work out in practice. It follows, therefore, that a direct-return trap is needed for the handling of the condensation even though it may not be called into action except under some operating condition where the pressure differential exceeds the static head provided. The installation of a direct-return trap assures safety for such systems, and guarantees the operation of the plant under varying conditions.

Automatic return traps, sometimes called alternating receivers, may be of the counter-balanced, tilting type, or spring actuated. These consist of a small receiver with an internal float, and when the condensate will not flow into the boiler under pressure, it will feed into the receiver of the trap, and in so doing, raise or tilt the float or mechanism which actuates a steam valve automatically. This admits steam to the receiver, at boiler pressure, and the equalizing of the pressures which follows allows the water to flow into the boiler.

Tilting Traps. With this type of trap, water enters a bowl and rises until its weight overbalances that of a counter-weight, and the bowl sinks to the bottom. As the bowl sinks, a valve is opened thus admitting live steam pressure on the surface of the water and the trap then discharges. After the water is discharged, the counter-weight sinks and raises the bowl, which in turn closes the valve and the cycle begins again. Tilting traps are necessarily intermittent in operation. They are not ordinarily equipped with glass water gages, as the action of the trap shows when it is filling or emptying The air relief of tilting traps is taken care of by the valves of the trap.

PROBLEMS IN PRACTICE

1 ● What is meant by water line difference in a gravity steam heating system?

The water line difference is the distance between the level of the water in the dry or wet return and the boiler water line. This difference is equivalent to the pressure required to overcome the maximum drop in the system and the operating pressure of the boiler

2 ● How many types of common mechanical returns are there and what are they?

Three: (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum pump.

3 ● In the ordinary vacuum system of steam heating, where does the vacuum usually exist?

On the return side of the system only, between the radiator trap and the vacuum pump. If the radiator supply valve is closed off, the vacuum may extend back through the radiator as far as the supply valve; if an inadequate supply of steam is furnished to the system, some vacuum may be developed in the steam main; but neither of these can be termed *normal operation.*

4 ● What is the distinction between the open and the closed vapor systems?

The open vapor system has the return line always open to the atmosphere, while the closed vapor system has an automatic device on the air vent so that air once expelled from the system through the vent cannot re-enter via this route.

5 ● On a vacuum system, what device must be placed on all drips before they enter the vacuum return line?

A thermostatic drip trap or occasionally, where large volumes of condensation are to be handled, a float trap, or combination float and thermostatic trap

6 ● How does the sub-atmospheric system differ in operation from the ordinary vacuum system?

The ordinary vacuum system has pressure in the steam line, and a vacuum produced by the vacuum pump in the return line, usually varying between 5 and 10 in. of mercury. The sub-atmospheric system may have either a vacuum or pressure on the steam and return lines according to the weather conditions, but a constant difference in pressure is maintained between the lines regardless of what vacuum may be carried. The vacuum, which is generally produced jointly by condensation and the exhausting action of the pump, in the system under conditions of throttled steam supply, will run much higher than in the ordinary vacuum system, and as high as 25 in. of mercury in the radiators

7 ● What is generally understood by zoning in building steam heating systems?

Zoning is a term applied to the placing of certain sections of a building on a single temperature control instead of having either individual room control or a single temperature control governing the whole building. Zones may be horizontal, such as a single story, a basement, or an attic, or vertical such as the north side, or the west side.

8 ● Why does the water line in the far end of a wet return in a gravity steam system rise higher than the water line in the boiler?

The friction of the steam flowing through the steam main from the boiler to the far end of the system and the pressure reduction resulting from the condensing action of the radiators causes a drop in steam pressure at the point where the wet return is connected, consequently, the steam pressure on top of the water in the wet return is less than the steam pressure on top of the water in the boiler, so the water in the end of the wet return rises until a balanced condition is set up.

9 ● On gravity one-pipe systems as indicated in Fig. 1 and Fig. 3, why is the drip on the steam runout connected to wet return?

Because if it were connected to dry return, the pressure drops to two different points would not necessarily be the same and the system would short circuit.

10 ● What is the function of the automatic return trap?

To insure the return of condensate to the boiler when the operating condition is such that the boiler pressure exceeds the static head on the returns.

11 ● What advantage is there to an air valve with a check to prevent the reentrance of expelled air?

A system equipped with such valves builds up a vacuum and holds the heat longer. With proper controls on the boiler, lower radiator temperatures can be maintained in mild weather, giving better plant efficiency.

12 ● What are the essentials of a two-pipe closed vapor system?

Packless graduated valves on radiators, thermostatic return traps on return and drips; an automatic return trap to prevent water from backing out of the boiler.

13 ● Why must the automatic return trap on two-pipe vapor systems be about 18 in. above the boiler water line?

That height is necessary to overcome water line difference owing to pressure drop and friction in pipe and fittings

Chapter 16

PIPING FOR STEAM HEATING SYSTEMS

Operating Characteristics, Steam Flow, Pipe Sizes, Tables for Pipe Sizing, One-Pipe Gravity Air-Vent Systems, Two-Pipe Gravity Air-Vent Systems, Two-Pipe Vapor Systems, Vacuum, Orifice, Atmospheric and Sub-Atmospheric Systems, Steam Supply Sources, Piping Connections and Details, Boiler and Radiator Connections, Piping for Indirect Heating Units, Dripping

IT is important that steam piping systems not only distribute steam at full or *design* load but during excess and partial loads. Usually the average winter steam demand is less than half of the demand at the *design* outside temperature. Moreover, in rapidly warming up a system, the load on the steam main and returns may exceed the maximum operating load for severe weather and even in moderate weather due to the necessity of raising the temperature of the metal in the system to the steam temperature and the building to the design indoor temperature. Investigations of the return of condensation have revealed that as high as 143 per cent of the design condensation rate may exist under conditions of actual operation.

The functions of piping are to supply the heating units with steam and to remove the condensation as well as the air in the case of those systems where both air and condensation are returned. To accomplish this effectively, the distribution of the steam should be rapid, equitable and without noise. To attain this result the release of air from the system should be facilitated as much as possible, as an air bound system will not heat readily nor properly. In designing the piping arrangement for a steam system it is desirable to maintain equivalent resistances in the supply and return piping to and from a radiator. Arranging the piping so the total distance from the boiler to the radiation is the same as the return piping distance from the heating unit back to the boiler, tends to obtain such a result. The condensation which occurs in steam piping as well as in radiators must be drained to prevent impeding the ready flow of the steam and air. The effect of back pressure in the returns, and excessive revaporization such as occurs where condensation is released from pressures considerably higher than the vacuum or pressure in the return must be avoided.

The piping design of a heating system is greatly influenced by its operating characteristics. Heating systems do not operate under a fixed condition as they are continually changing due to variation in load. As the system is being filled with steam the pressure existing in various

TABLE 1. FLOW OF STEAM IN PIPES

P = loss in pressure in pounds.
d = inside diameter of pipe in inches
L = length of pipe in feet.
D = weight of 1 cu ft of steam.
W = pounds of steam per hour.

$$W = 5220 \sqrt{\frac{PDd^5}{\left(1 + \frac{3.6}{d}\right)L}}$$

$$P = 0\ 0000000367 \left(1 + \frac{3.6}{d}\right) \frac{W^2 L}{Dd^5}$$

Pressure Loss in Ounces	Col 1 $5220\sqrt{\frac{P}{100}}$	Pipe Size: Nominal	Pipe Size: Actual Internal Diameter	Internal Area of Pipe Sq Inches	Col 2 $\sqrt{\frac{d^5}{1+\frac{3.6}{d}}}$	Steam Press by Gage	Col 3 $\sqrt{D}$	Length of Pipe in Feet	Col 4 $\sqrt{\frac{100}{L}}$
0.25	65 28	1	1.049	0.864	0.536	−1.0[a]	0.187	20	2 240
0.50	92.28	1¼	1.380	1 496	1 178	−0.5[a]	0.190	40	1.580
1.00	130.5	1½	1 610	2.036	1 828	0 0	0 193	60	1 290
2	184.6	2	2 067	3.356	3 710	0 3	0 195	80	1.120
3	226.0	2½	2 469	4 788	6 109	1 3	0 201	100	1 000
4	261.0	3	3.068	7.393	11.183	2 3	0 207	120	0 912
5	291.8	3½	3 548	9 887	16 705	5 3	0 223	140	0 841
6	319.7	4	4.026	12.730	23.631	10 3	0 248	160	0.793
7	345.3	4½	4 506	15 947	32 134	15 3	0.270	180	0.741
8	369.1	5	5 047	20 006	43 719	20 3	0 290	200	0.710
10	412.7	6	6 065	28 886	71 762	30.3	0.326	250	0 632
12	452 0	7	7 023	38 743	106 278	40 3	0 358	300	0 578
14	488.3	8	7 981	50 027	149 382	50.3	0 388	350	0 538
16	522.0	9	8 941	62 786	201 833	60 3	0 415	400	0 500
20	583.6	10	10 020	78 854	272.592	75 3	0 452	450	0 477
24	639.3	12	12 000	113 098	437 503	100 3	0 507	500	0 447
28	690.5	14	13 250	137 880	566 693	125 3	0 557	600	0 407
32	738.2	16	15.250	182.655	816.872	150 3	0 603	700	0.378
40	825 4					175 3	0 645	800	0 354
48	904.1					200 3	0 685	900	0 333
80	1167 2							1000	0 316
160	1650 7							1200	0 289
320	2334.5							1500	0.258
480	2859 1							2000	0 224

Column 1 × 2 × 3 × 4 = lb of steam per hour that will flow through a straight pipe for a given condition

Example 1: 1 oz drop — 2 in pipe — 1 3 lb press — 100 ft equivalent length.

130 5 × 3 710 × 0 201 × 1 = 97 2 lb per hour
97 2 × 4[b] = 388 8 sq ft equivalent radiation

Table 1 does not allow for entrained water in low-pressure steam, condensation in covered pipe and roughness in commercial pipe as found in practice

[a]Pounds per square inch gage = 2 04 in Vacuum, Mercury Column
[b]The factor 4 is the approximate equivalent in square feet of steam radiation of 1 lb of steam per hour

locations may be different than those which exist for appreciable periods of time at other locations and which under constant pressure may have conditions that are approximately the same. In designing piping it is of especial importance to arrange the system to preclude trouble caused by such pressure differences. The systems which readily release the air permit uniform pressures to be attained in much shorter time intervals than those which are sluggish. Results are given in Fig. 1 from investigations[1] to determine the rate of condensate and air return from a two-pipe gravity heating system. Variations in the steam pressure during the warming up period when the rate of air elimination and condensation is high are clearly indicated in these curves.

It is evident that the condensation flow during the initial warming up

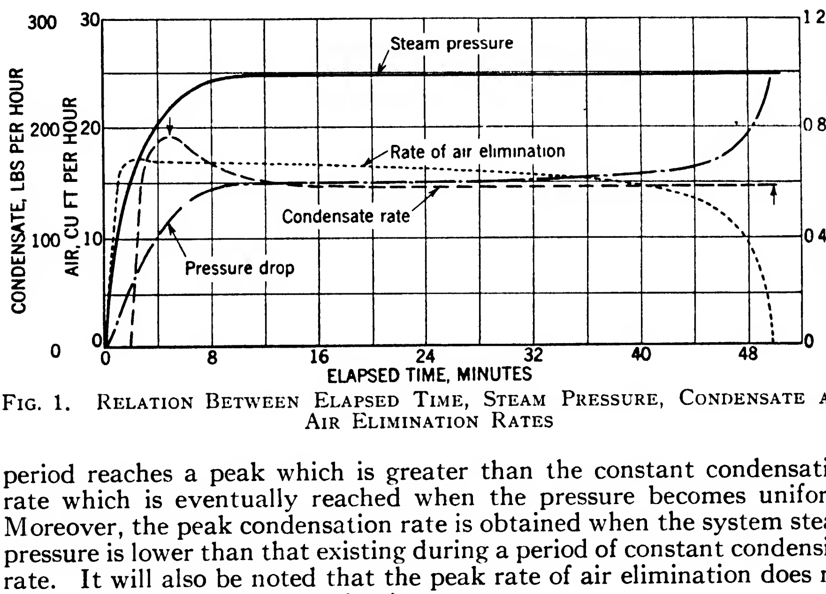


FIG. 1. RELATION BETWEEN ELAPSED TIME, STEAM PRESSURE, CONDENSATE AND AIR ELIMINATION RATES

period reaches a peak which is greater than the constant condensation rate which is eventually reached when the pressure becomes uniform. Moreover, the peak condensation rate is obtained when the system steam pressure is lower than that existing during a period of constant condensing rate. It will also be noted that the peak rate of air elimination does not coincide with the higher condensing rate.

STEAM FLOW

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship has been established by Babcock in the formula given at the top of Table 1. In Columns 1, 2, 3, and 4 of this table, the numerical values of the factors for different pressure losses, pipe diameters, steam densities and lengths of pipe have been worked out in convenient form so that the steam flowing in any pipe may be calculated by multiplying together the proper factors in each column as shown in the example at the bottom of the table.

[1]A.S H V.E. RESEARCH REPORT No 954—Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J L Blackshaw (A S H V E TRANSACTIONS, Vol 39, 1933, p. 199).

PIPE SIZES

The determination of pipe sizes for a given load in steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system, taking into consideration the direction of condensate flow.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. Unusual conditions in the building to be heated.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that (1) the total pressure drop does not exceed the initial pressure of the system; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmospheric which normally operate under controlled partial vacua, the orifice, and the vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; and (4) the equivalent head due to pressure drop does not exceed the difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, or the dry return, and the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. Laboratory experiments limit this to the capacities given in Tables 2 and 3 for vertical risers and in Table 4 for horizontal pipes at varying grades.

Maximum Velocity and Reaming

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of

TABLE 2. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR ONE-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

Pipe Size Inches	Velocity Feet Per Second	Pressure Drop Ounces per 100 Ft	Capacity		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
1	14.1	0.68	45	10,961	11.3
1¼	17 6	0 66	98	23,765	24.5
1½	20 0	0 66	152	36,860	38 0
2	23 0	0.57	288	69,840	72 0
2½	26 0	0 54	464	112,520	116 0
3	29 0	0 48	799	193,600	199 8
3½	31 0	0 44	1144	277,000	286 0
4	32 0	0.39	1520	368,000	380.0

INSTRUCTIONS FOR USING TABLE 2

1 Capacities given in Table 2 should never be exceeded on one-pipe risers

2 Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe

3 All pipe should be well reamed and free from constrictions Fittings should be up to size (See Tables 5 and 6)

TABLE 3 MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR TWO-PIPE LOW PRESSURE STEAM

Based on A. S H. V. E. Research Laboratory Tests

Pipe Size Inches	Velocity Feet per Second	Pressure Drop Ounces per 100 Ft	Capacity		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
¾	20	....	40	9550	10 0
1	23	1 78	74	17,900	18 45
1¼	27	1 57	151	36,500	37.65
1½	30	1 48	228	55,200	57.0
2	35	1 33	438	106,100	109.5
2½	38	1 16	678	164,100	169 4
3	41	0.95	1129	273,500	282.2
3½	42	0 81	1548	375,500	387 0
4	43	0.71	2042	495,000	510 5

INSTRUCTIONS FOR USING TABLE 3

1. The capacities given in this table should never be exceeded on two-pipe risers.

2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.

3 All pipe should be well reamed and free from constrictions Fittings should be up to size. (See Tables 5 and 6)

ater in certain parts of the system until the steam flow is reduced fficiently to permit the water to pass. The velocity at which such sturbances take place is a function of (1) the pipe size, whether the pipe ns horizontally or vertically, (2) the pitch of the pipe if it runs horintally, (3) the quantity of condensate flowing against the steam, and) freedom of the piping from water pockets which under certain contions act as a restriction in pipe size.

Three factors of uncertainty always exist in determining the capacity any steam pipe. The first is variation in manufacture, which appartly cannot be avoided and which caused an actual difference of 20 per nt in the capacity of a 1-in. pipe in experiments carried on at the S.H.V.E. Research Laboratory (Table 5). The second is the reaming the ends of the pipe after cutting, which, experiments indicate, might duce the capacity of a 1-in. pipe as much as 28.7 per cent (Table 6). ıe third is the uniformity in grading the pipe line. All of the capacity bles given in this chapter include a factor of safety. However, the pipe

BLE 4. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES FOR STEAM AND CONDENSATE FLOWING IN OPPOSITE DIRECTIONS[a]

Pitch of Pipe in Inches per 10 Ft

CH OF PIPE	¼ IN		½ IN		1 IN		1½ IN		2 IN		3 IN		4 IN		5 IN	
Pipe Size ıches	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel
¾	25 0	12	30 3	14	37 3	18	40 4	19	42 5	20	46 1	21	47 5	22	49 3	23
	45 8	12	52 6	15	63 0	17	70 0	20	75 2	22	83 0	23	87 9	25	90 2	26
¼	104 9	18	117 2	20	133 0	23	144 5	25	154 0	27	165 0	28	172 6	29	178 2	31
½	142 6	18	159 0	21	181 0	23	196 5	25	209 3	27	224 0	28	234 8	30	242 6	31
	236.0	19	263 5	20	299 5	23	325 5	25	346 5	27	371 5	28	388 4	29	401 1	30

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory

which Table 5 is based showed no particular defects or constrictions the inside, and the factor of safety referred to does not cover abnormal fects or constrictions *nor does it cover pipe not properly reamed.*

uivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must ow for the friction offered by the pipe as well as for the additional sistance of the fittings and valves. These resistances generally are ıted in terms of straight pipe; in other words, a certain fitting will oduce a drop in pressure equivalent to so many feet of straight run of e same size of pipe. Table 7 gives the number of feet of straight pipe ually allowed for the more common types of fittings and valves. In all e sizing tables in this chapter the *length of run* refers to the *equivalent gth of run* as distinguished from the *actual length* of pipe in feet. The gth of run is not usually known at the outset; hence it is necessary to sume some pipe size at the start. Such an assumption frequently is nsiderably in error and a more common and practical method is to sume the length of run and to check this assumption after the pipes are ed. For this purpose the length of run usually is taken as double the ual length of pipe.

TABLE 5. PER CENT DIFFERENCE IN CAPACITY FOR CARRYING STEAM AND CONDENSATE DUE TO VARIATION OF PIPE SIZE AND SMOOTHNESS[a]

	MAXIMUM CONDENSATION, LB PER HOUR			
Size of pipe	¾ In.	1 In.	1¼ In.	1½ In.
Minimum	14.00	24.89	45.42	70.50
Maximum	15 20	30.08	52.08	82.00
Per cent variation	8.6	20 8	14 7	16 3

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

TABLE 6. EFFECT OF REAMING ENTRANCE TO ONE-INCH ONE-PIPE RISERS[a]

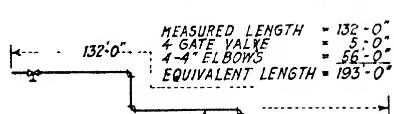
	MAXIMUM CAPACITY OF RISER	PER CENT DECREASE
Reamed entrances	24.7 lb per hour	0 0
Rounded entrances	23 9 lb per hour	3 2
Squared entrances	22.2 lb per hour	10 1
Three wheel cutter	19 2 lb per hour	22 2
Single wheel cutter	17.6 lb per hour	28 7

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory

TABLE 7. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

SIZE OF PIPE INCHES	ST'D ELBOW	SIDE OUTLET TEE	GATE VALVE	GLOBE VALVE	ANGLE VALVE
	Length in Feet to be Added to Run				
2	5	16	2	18	9
2½	7	20	3	25	12
3	10	26	3	33	16
3½	12	31	4	39	19
4	14	35	5	45	22
5	18	44	7	57	28
6	22	50	9	70	32
7	26	55	10	82	37
8	31	63	12	94	42
9	35	69	13	105	47
10	39	76	15	118	52
12	47	90	18	140	63
14	53	105	20	160	72

Example of length in feet of pipe to be added to actual length of run.



TABLES FOR PIPE SIZING[2]

Factors determining the size of a steam pipe and its allowable limit of capacity are as follows:

1. Pipe condensate flowing with steam.
2. Pipe condensate flowing against steam.
3. Pipe and radiator condensate flowing with steam.
4. Pipe and radiator condensate flowing against steam

It is apparent that (3) and (4) are practically limited to one-pipe systems while (1) and (2) cover all other systems.

Tables 8 and 9, worked out for determining pipe sizes, have their columns lettered continuously, Columns *A* through *L* being in Table 8, and *M* through *EE* in Table 9. In the following text, reference made to columns will be by letter. The tables are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation[3] (*abbreviated EDR*) per hour. The drops indicated are drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed without unusual or noticeable defects.

Table 8 may be used for sizing piping for steam heating systems by determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensation flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensation flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve, and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensation without noise.

Sizing of return piping may be done with the aid of Table 9 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 8. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2-lb gage?

Solution. It will be assumed, if the measured lengtn of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one half of the initial pressure, the drop could be 1 lb or less. With a pressure drop of 1 lb and a length of run of 1,000 ft, the drop per 100 ft would be 1/10 lb, while if the total drop were ½ lb, the drop per 100 ft

[2]Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound

[3]As steam system design has materially changed in recent years so that 240 Btu no longer expresses the heat of condensation from a square foot of radiator surface per hour, and as present day heating units have different characteristics from older forms of radiation, it is the purpose of THE GUIDE to gradually eliminate the empirical expression *square foot of equivalent direct radiation EDR*, and to substitute a logical unit based on the Btu The new terms to express the equivalent of 1000 Btu (Mb), and 1000 Btu per hour (Mbh), have been approved by the A S.H V E

would be $\frac{1}{20}$ lb. In the first instance the pipe could be sized according to Column *D* for $\frac{1}{16}$ lb per 100 ft, and in the second case, the pipe could be sized according to Column *C* for $\frac{1}{24}$ lb. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized. Ordinarily resizing will be unnecessary.

TABLE 8. STEAM PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pipe Size In	CAPACITIES OF STEAM MAINS AND RISERS — Direction of Condensation Flow in Pipe Line: With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only		Special Capacities for One-Pipe Systems Only		
	1/32 lb or ½ Oz Drop	1/24 lb or ⅔ Oz Drop	1/16 lb or 1 Oz Drop	⅛ lb or 2 Oz Drop	¼ lb or 4 Oz Drop	½ lb or 8 Oz Drop	Vertical	Horizontal	Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
A	*B*	*C*	*D*	*E*	*F*	*G*	*H* [a]	*I* [c]	*J* [b]	*K*	*L* [c]
¾	....	....	30	....	....	....	30	....	25	....	....
1	39	46	56	79	111	157	56	26	45	20	20
1¼	87	100	122	173	245	346	122	58	98	55	55
1½	134	155	190	269	380	538	190	95	152	81	81
2	273	315	386	546	771	1,091	386	195	288	165	165
2½	449	518	635	898	1,270	1,797	635	395	464	....	260
3	822	948	1,163	1,645	2,326	3,289	1,129	700	799	....	475
3½	1,228	1,419	1,737	2,457	3,474	4,913	1,548	1,150	1,144	....	745
4	1,738	2,011	2,457	3,475	4,914	6,950	2,042	1,700	1,520	....	1,110
5	3,214	3,712	4,546	6,429	9,092	12,858	....	3,150	....	....	2,180
6	5,276	6,094	7,462	10,553	14,924	21,105	....	....	....	....	....
8	10,983	12,682	15,533	21,967	31,066	43,934	....	....	....	....	....
10	20,043	23,144	28,345	40,085	56,689	80,171	....	....	....	....	....
12	32,168	37,145	45,492	64,336	90,985	128,672	....	....	....	....	....
16	60,506	69,671	84,849	121,012	169,698	242,024	....	....	....	....	....
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Undripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

Note.—All drops shown are in pounds per 100 ft of equivalent run—based on pipe properly reamed.

[a] Do not use Column *H* for drops of 1/24 or 1/32 lb; substitute Column *C* or Column *B* as required

[b] Do not use Column *J* for drop of 1/32 lb except on sizes 3 in. and over; below 3 in substitute Column *B*

[c] On radiator runouts over 8 ft long increase one pipe size over that shown in Table 8

ONE-PIPE GRAVITY AIR-VENT SYSTEMS

One-pipe gravity air-vent systems in which the equivalent length of run does not exceed 200 ft should be sized as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use $\frac{1}{16}$-lb drop (Column *D*).

2. *Where the riser runouts are not dripped* and the steam and condensation flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.

TABLE 9. RETURN PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

CAPACITY OF RETURN MAINS AND RISERS

Pipe Size Inches	1/32 Lb or ½ Oz Drop per 100 Ft			1/24 Lb or ⅔ Oz Drop per 100 Ft			1/16 Lb or 1 Oz Drop per 100 Ft			⅛ Lb or 2 Oz Drop per 100 Ft			¼ Lb or 4 Oz Drop per 100 Ft			½ Lb or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac	Wet	Dry	Vac	Wet	Dry	Vac	Wet	Dry	Vac.	Wet	Dry	Vac	Wet	Dry	Vac
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
Mains																		
¾	......	......	......	......	......	326	......	......	400	......	......	568	......	......	800	......	......	1,130
1	500	248	......	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	......	......	1,977
1¼	850	520	......	990	595	976	1,200	670	1,200	1,700	868	1,704	2,400	962	2,400	......	......	3,390
1½	1,350	822	......	1,570	943	1,547	1,900	1,058	1,900	2,700	1,362	2,696	3,800	1,512	3,800	......	......	5,370
2	2,800	1,880	......	3,240	2,140	3,256	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	......	......	11,300
2½	4,700	3,040	......	5,300	3,470	5,453	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	......	......	18,925
3	7,500	5,840	......	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,190	21,400	10,000	21,400	......	......	30,230
3½	11,000	7,880	......	13,200	8,800	13,020	16,000	10,000	16,000	22,000	12,900	22,710	32,000	14,300	32,000	......	......	45,200
4	15,500	11,700	......	18,300	13,400	17,910	22,000	15,000	22,000	31,000	19,300	31,220	44,000	21,500	44,000	......	......	62,180
5	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
6	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100
Risers																		
¾	......	190	......	......	190	570	......	190	700	......	190	994	......	190	1,400	......	......	1,977
1	......	450	......	......	450	976	......	450	1,200	......	450	1,704	......	450	2,400	......	......	3,390
1¼	......	990	......	......	990	1,547	......	990	1,900	......	990	2,696	......	990	3,800	......	......	5,370
1½	......	1,500	......	......	1,500	3,256	......	1,500	4,000	......	1,500	5,680	......	1,500	8,000	......	......	11,300
2	......	3,000	......	......	3,000	5,453	......	3,000	6,700	......	3,000	9,510	......	3,000	13,400	......	......	18,925
2½	......	......	......	......	......	8,710	......	......	10,700	......	......	15,910	......	......	21,400	......	......	30,230
3	......	......	......	......	......	13,020	......	......	16,000	......	......	22,710	......	......	32,000	......	......	45,200
3½	......	......	......	......	......	17,910	......	......	22,000	......	......	31,220	......	......	44,000	......	......	62,180
4	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
5	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100

3. *For up-feed steam risers* carrying condensation back from the radiators, use Column *J*.

4. *For down-feed systems the main risers* of which do not carry any radiator condensation, use Column *H*.

5. *For the radiator valve size and the stub connection*, use Column *K*.

6. For the *dry return main*, use Column *U*.

7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over ¼ lb. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where 1/64-lb drop is being used, the steam main and dripped runouts would be sized from Column *C*; radiator runouts and undripped riser runouts from Column *L*; up-feed risers from Column *J*; the main riser on a down-feed system from

TABLE 10. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 2

Part of System	Section of Pipe	Radiation Supplied (Sq Ft)	Theoretical Pipe Size (Inches)	Practical Pipe Size (Inches)
Branches to radiators	......	100	2	2
Branches to radiators.	....	50	1¼	1¼
Riser	*a* to *b*	200	2	2
Riser	*b* to *c*	300	2½	2½
Riser	*c* to *d*	400	2½	2½
Riser	*d* to *e*	500	3	3
Riser	*e* to *f*	600	3	3
Branch to riser	*f* to *g*	600	3½	3½
Supply main	*g* to *h*	600	3	3
Branch to supply main	*h* to *j*	600	2½	3
Dry return main	*f* to *k*	600	1¼	2
Wet return main	*k* to *m*	600	1	2
Wet return main	*m* to *n*	600	1	2
Wet return main	*n* to *p*	600	1	2

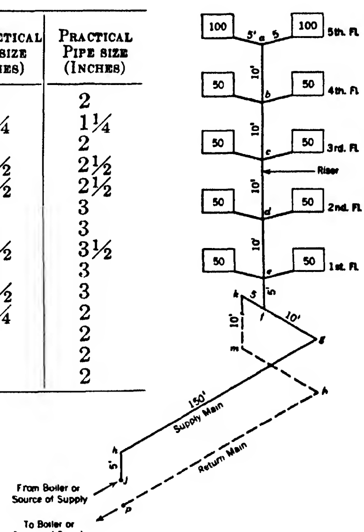


FIG. 2. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of 1/64 lb); the dry return from Column *R*; and the wet return from Column *Q*.

With a 1/32-lb drop the sizing would be the same as for 1/64 lb except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Example 3. Size the one-pipe gravity steam system shown in Fig. 2 assuming that this is all there is to the system or that the riser and run shown involve the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of ¼ lb the drop per 100 ft will be slightly less than 1/16 lb. It would be well in this case to use 1/64 lb, and this would result in the theoretical sizes indicated in Table 10. These theo-

retical sizes, however, should be modified by not using a wet return less than 2 in. while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet return is made 2 in.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should not be less than half its diameter at its largest part.

4. Supply mains, branches to risers, or risers, should be dripped where necessary.

5. Where supply mains are decreased in size they should be dripped.

TWO-PIPE GRAVITY AIR-VENT SYSTEMS

The method employed in determining pipe sizes for two-pipe gravity air-vent systems is similar to that described for one-pipe systems except that the steam mains never carry radiator condensation. The drop allowable per 100 ft of equivalent run is obtained by taking the equivalent length to the farthest radiator as double the actual distance, and then dividing the allowable or desired total drop by the number of hundreds of feet in the equivalent length. Thus in a system measuring 400 ft from the boiler to the farthest radiator, the approximate equivalent length of run would be 800 ft. With a total drop of ½ lb the drop per 100 ft would be $\frac{1/2}{8}$ or $\frac{1}{16}$ lb; therefore, Column *D* would be used for all steam mains where the condensation and steam flow in the same direction. If a total drop of ¼ lb is desired, the drop per 100 ft would be $\frac{1}{32}$ lb and Column *B* would be used. If the total drop were to be 1 lb, the drop per 100 ft would be ⅛ lb and Column *E* would be used.

For mains and riser runouts that are not dripped, and for radiator runouts where in all three cases the condensation and steam flow in opposite directions, Column *I* should be used, while for the steam risers Column *H* should be used unless the drop per 100 ft is $\frac{1}{24}$ lb or $\frac{1}{32}$ lb, when Columns *B* or *C* should be substituted so as not to exceed the drop permitted.

On an overhead down-feed system the main steam riser should be sized by reference to Column *H*, but the down-feed steam risers supplying the radiators should be sized by the appropriate Columns *B* through *G*, since the condensation flows downward with the steam through them. The riser runouts, if pitched down toward the riser as they should be, are sized the same as the steam mains, and the radiator runouts are made the same as in an up-feed system.

In either up-feed or down-feed systems the returns are sized in the same manner and on the same pressure drop basis as the steam main; the return mains are taken from Columns *O*, *R*, *U*, *X*, or *AA* according to the drop used for the steam main; and the risers are sized by reading the lower part of Table 9 under the column used for the mains. The horizontal runouts from the riser to the radiator are not usually increased on the return lines although there is nothing incorrect in this practice. The same notes apply that are given for one-pipe gravity systems.

TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensation return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, especially when it is desirable to carry a moderate or low fire, and (3) because with large variation in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Column *U* (lower portion) and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed ⅛ lb to ¼ lb, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over ⅛ lb divided by 4, or 1/32 lb. In this case the steam mains would be sized from Column *B*; the radiator and undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of 1/32 lb. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over 1/32 lb. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.

2 Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3 In general it is not desirable to have a supply main smaller than 2 in., and when the supply main is 3 in. or over at the boiler or pressure reducing valve it should not be less than 2½ in. at the far end

4. When necessary, supply main, supply risers, or branches to supply risers should be dripped separately into a wet return. The drip for a vapor system may be connected into the dry return through a thermostatic drip trap.

VACUUM, ORIFICE, SUB-ATMOSPHERIC SYSTEMS

Vacuum, atmospheric, sub-atmospheric and orifice systems are usually employed in large installations and have total drops varying from ¼ to ½ lb. Systems where the maximum equivalent length does not exceed

200 ft preferably employ the smaller pressure drop while systems over 200 ft equivalent length of run more frequently go to the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of ½ lb divided by 12, or $\frac{1}{24}$ lb. In this case the steam main would be sized from Column *C*, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of $\frac{1}{24}$ lb). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed ⅛-lb drop per 100 ft of equivalent run nor to exceed 1 lb total pressure drop in any system.
2. Pitch of mains should not be less than ¼ in. in 10 ft.
3. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.
4. In general it is not considered desirable to have a supply main smaller than 2 in. When the supply main is 3 in. or over at the boiler or pressure reducing valve, it should not be less than 2½ in. at the far end.
5. When necessary, the supply main, supply riser, or branch to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.
6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in Chapter 15.
7. No lifts can be used in orifice and atmospheric systems. In sub-atmospheric systems the lift must be at the vacuum pump.

STEAM SUPPLY SOURCES

Steam heating systems may be supplied from boilers used exclusively for heating or which provide steam both for heating and other purposes. Low pressure boilers are widely used because in modern practice steam heating systems usually operate at low pressures, usually not exceeding 15 lb gage. Frequently the heating boiler serves the heating system and the domestic hot water supply (See Chapter 43). If the boiler plant supplies steam for other purposes requiring pressures exceeding 15 lb gage, a pressure reducing valve is the means used to lower the steam to the pressure required for use in heating. In some installations there may be separate zones used to supply steam for (*a*) direct steam heating, (*b*) process work, (*c*) air conditioning or ventilation, and (*d*) domestic water heating. Each of these may require different steam pressures and consequently individual pressure reducing valves are needed.

PIPING CONNECTIONS AND DETAILS

Piping connections may be classified into two groups: *first*, those suitable for any system of steam heating; *second*, those devised for certain systems which cannot be satisfactorily applied to any other type. There are also various details that apply to piping on the steam side which cannot be used on the returns. An installation that is designed and sized

correctly and installed with care may be rendered defective by the use of improper connections, such as runouts that do not allow for expansion, thermostatic traps unprotected from scale, pressure-reducing valves without strainers, and lack of drips at required points.

BOILER CONNECTIONS

Supply

Boiler headers and connections have the largest sizes of pipe used in a system. Cast-iron, horizontal-type, low pressure heating boilers usually have several tapped outlets in the top, the manufacturers recommending their use in order to reduce the velocity of the steam in the vertical uptakes from the boiler and to permit entrained water to return to the boiler instead of being carried over into the steam main where it must be cared for by dripping. Steel heating boilers usually are equipped with

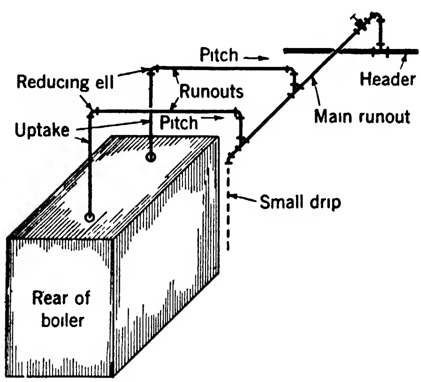


FIG. 3. OLD STYLE STANDARD BOILER CONNECTIONS

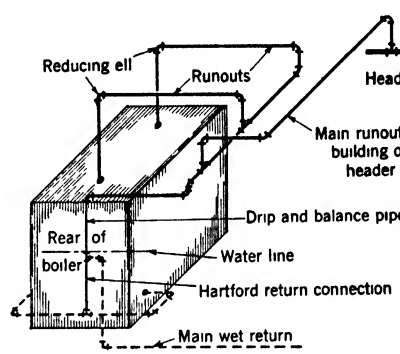


FIG. 4. APPROVED METHOD OF BOILER CONNECTIONS

only one steam outlet but many engineers believe that better results are obtained by specifying that such boilers have two. The second outlet, usually located 3 or 4 ft back of the regular one, reduces the velocity 50 per cent in the steam uptake.

Fig. 3 shows a type of boiler connection that was used for many years and one with which some boilers are now piped. The uptakes are carried as high as possible, turned horizontally and run out to the side of the boiler and then are connected together into the main boiler runout which drops into the top of the boiler header through a boiler stop valve. No drips are provided on this type of runout except a very small one which is sometimes installed on the boiler side of the stop valve. Fig. 4 shows a type of boiler connection which is regarded as superior to that shown in Fig. 3 and which is the type illustrated in the system diagrams in Chapter 15. This type is similar to that shown in Fig. 3 except that the horizontal branches from the uptakes are connected into the main boiler runout, and the steam is carried toward the rear of the boiler. The branch to the building or boiler header is taken off *behind* the last horizontal boiler connection. At the rear end of this main runout, a large size drip, or balance

pipe, is dropped down into the boiler return, or into the top of the Hartford Loop, which is described in a following paragraph. As a result, any water carried over from the boiler follows the direction of steam flow toward the rear and is discharged into the rear drip, or balance pipe, without being carried over into the system.

Return

Cast-iron boilers are generally provided with return tappings on both sides, but steel boilers often are equipped with only one return tapping. A boiler with side return tappings will usually have a more effective circulation if both tappings are used. Check valves generally should not be used on the return connection to steam heating boilers from one and two pipe gravity systems because they are not always dependable inasmuch as a small piece of scale or dirt lodged on the seat will hold the tongue open and make the check useless. These valves also offer a certain amount of resistance to the returns coming back to the boiler, and in gravity systems will raise the water line in the far end of the wet return several inches[4]. However, if check valves are omitted and the steam pressure is raised with the boiler steam valve closed, the water in the boiler will be blown out into the return system with the accompanying danger of boiler damage. These objections are largely overcome with the Hartford return connection.

Hartford Return Connection

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford Connection, or the Underwriters Loop, is recommended.

Fig. 5 shows this connection for a two-boiler installation. For a single boiler installation the connection is made as is indicated for one boiler. The essential features of construction of a Hartford Loop connection are: (1) a direct connection (made without valves) between the steam side of the boiler and the return side of the boiler, and (2) a close nipple connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return balance connection.

Sizing Boiler Connections

Little authentic information is available on the sizing of boiler runouts and steam headers. Although many engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runout on the same basis as the building

[4]See method of calculating height above water line for gravity one-pipe systems in Chapter 15

mains. The difference in size between the vertical uptakes from the boiler and the horizontal main or runout is compensated for by the use of reducing ells (Figs. 3 and 4).

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the

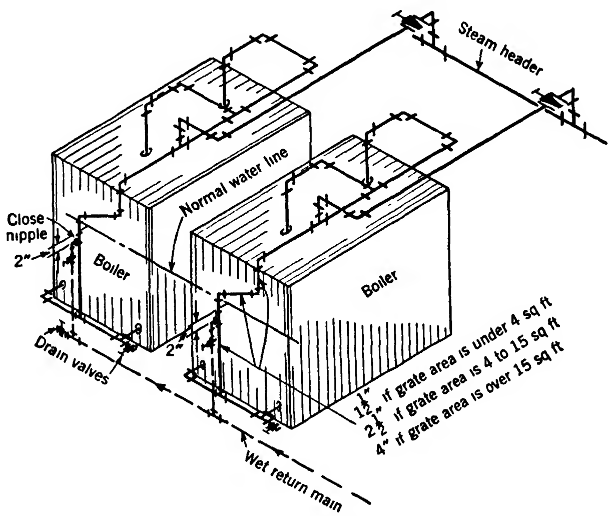


FIG. 5. THE HARTFORD RETURN CONNECTION

return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operating, and the line sized for a very small pressure drop, the size being obtained from the Chart for Pressure Drop for Various Rates of Flow of Water, Fig. 3, Chapter 43. The relative boiler loads should be considered, as in the case of gravity return connections. Boiler header and piping sizes should be based on the total load.

HIGH PRESSURE STEAM

When steam heating systems are supplied with steam from a high pressure plant, one or more pressure-reducing valves are used to bring the

pressure down to that required by the heating system. It has been considered good practice to make the pressure reductions in steps not to exceed 50 lb in each case. For example, in reducing from 100-lb gage to 2-lb gage, two pressure reducing valves would be used, the first reducing the pressure from 100-lb gage to 50 lb and the second reducing the pressure from 50-lb gage to 2-lb gage. Valves are available that will reduce 100 lb in one step, and it is questionable whether two valves are now required for initial pressures of 150 lb or less.

The pressure-reducing valve, or pressure-regulator as it is sometimes termed, has ratings which vary 200 to 400 per cent. Some of these ratings are based on arbitrary steam velocities through the valve of 5,000 to 10,000 fpm and it is assumed that the valve when wide open has the same capacity as the pipe on the inlet opening of the valve. At times

TABLE 11. CAPACITIES OF PRESSURE-REDUCING VALVES

(100-LB GAGE DOWN TO ANY PRESSURE—52 LB OR LESS)

Inlet Nominal Pipe Diameter (Inches)	Pounds Steam per Hour at 100-Lb Gage	Equivalent Direct Radiation Sq Ft at ¼ Lb	Equivalent Direct Radiation Sq Ft at ⅛ Lb
½	866	3,464	2,598
¾	1,576	6,304	4,728
1	2,459	9,836	7,377
1¼	4,263	17,052	12,689
1½	5,808	23,232	17,424
2	9,564	38,256	28,692
2½	13.623	54,492	40,869
3	21 041	84,104	63,123
3½	28,213	112.852	84,039
4	36,285	145,140	108,855
5	56,971	227,884	170,913
6	82,336	329,344	247,008

Formula:

$$\frac{A \times V \times 3600 \times .50}{144 \times 3.88} = \text{pounds per hour passed by orifice}$$

where

A = area of inlet pipe, square inches
V = velocity of steam through orifice (approximately 870 fps)
50 = 70 per cent efficiency of orifice less 20 per cent for factor of safety
144 = square inches in 1 sq ft
3600 = seconds in one hour
3 88 = cubic feet per pound at 100-lb gage

it is considered desirable to keep the steam velocity in the high pressure section of the piping and the low pressure section constant. The velocity through the valve port is obviously a function of the pressure drop across the valve. It is well known that steam flowing through an orifice increases its velocity until the pressure on the outlet side is reduced to 58 per cent of the absolute pressure on the inlet side, and that with further reduction of pressure on the outlet side little change in velocity will be obtained. As practically all pressure-reducing valves used for steam heating work lower the steam pressure to less than 58 per cent of the inlet pressures, only the maximum velocity through such valves need be considered. If it is assumed that the valve, when fully open, has an area equal to that of the inlet pipe size, that the steam is flowing into a pressure less

than 58 per cent of the initial pressure, that the orifice efficiency is approximately 70 per cent, and that 20 per cent more is allowed for a factor of safety, then the pressure reducing valves will have the working capacities shown in Table 11. If the valve, when fully open, does not give an orifice area equal to that of the pipe on the inlet side, then the capacities will be proportional to the percentage of opening secured, taking the pipe area as 100 per cent. More frequently, difficulty is encountered from the use of pressure reducing valves which are too large in size instead of being too small. Where valves are large in size, the valve tends to work close to the seat, causing it to cut out in a relatively short time, as well as being noisy in operation.

Most exact regulation of pressure on steam heating systems is secured from diaphragm-operated valves controlled by a pilot line from the low pressure pipe, taken off the low pressure main at least 15 ft from the reducing valve. The reducing valves operating on the proportional-reduction principle will give a variation of steam pressure on the low pressure side if the initial pressure varies between considerable limits. The so-called dead-end valve is used for reduced pressures where the line

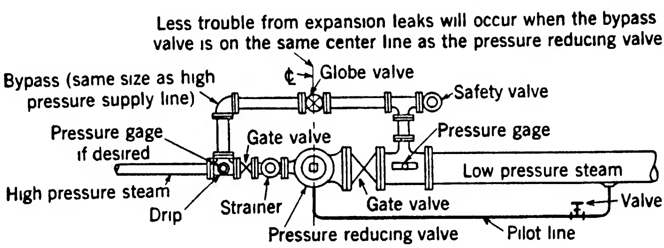


FIG. 6. TYPICAL PRESSURE-REDUCING VALVE INSTALLATION

has not sufficient condensing capacity at all times to condense the leakage that might occur with the ordinary valve. Single-disc valves do not give as close regulation as double-disc valves, but the single disc is preferable where dead-end valves are necessary, such as on short runs to thermostatically controlled hot water heaters, central fan heating units and unit heaters.

The correct installation (Fig. 6) of a pressure-reducing valve includes a pressure-reducing valve with a gate valve on each side, a by-pass controlled by a globe valve, a pressure gage on the low pressure side, and a safety valve on the low pressure main at some point, usually within a reasonable distance of the pressure-reducing valve. Pressure-reducing valves should have expanded outlets for sizes greater than 2 in. Where the steam main is of still larger diameter than the expanded outlet, and in cases where straight valves are used, an increaser is placed close against the outlet of the valve to reduce the velocity immediately after passing through the valve. Strainers are recommended on the inlets of all pressure-reducing valves. A pressure gage may be located on the high-pressure line near the valve if desired.

Owing to the large variation in steam demand on the average heating system, it is generally advisable to use two pressure-reducing valves con-

nected in parallel. One valve should be large enough for the maximum load and the other should have a diameter approximately half that of the first. The smaller valve can be used most of the time, for it will give much better regulation than the larger one on light or normal loads. It is advisable to install a safety valve on the low pressure side of a pressure reducing valve set at 15 lb gage where the initial pressure exceeds 50 lb.

Control Valves

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never be used for throttling. Angle globe valves and straight globe valves should be used for throttling, as done on by-passes around pressure reducing valves or on by-passes around traps.

Radiator Connections

Radiator connections must not only be properly pitched at the time they are installed but must be arranged so that the pitch will be maintained under the strains of expansion and contraction, the variation of the riser or drop length with expansion and contraction, and the changes resulting from shrinkage, as in the case of frame buildings, and settling. In a three story building the change in the building height may sometimes amount to 1 in. or more. The riser run outs and the spring pieces from mains to risers are made by swing joints which permit the expansion or contraction to occur under heating and cooling without bending of pipes. In longer risers expansion joints of commercial construction or comprised of a series of swing joints alternating with properly spaced anchors are placed in intermediate lengths of the risers. These are basic requirements for radiator connections in all types of steam heating systems. The simplest radiator connection is for a one-pipe system in which only one radiator connection is necessary. Where the runouts must come above the floor, the runouts can project out into the room only a short distance and with standard length of radiator legs the vertical space is small. If the runouts are located under the floors there is frequently more space available to permit making a swing joint with ample pitch. Fig. 7 illustrates two satisfactory methods of making runouts for one-pipe gravity air vent systems for either the up-feed or the down-feed type. Where the vertical distance is limited and the runouts must run above the floor the radiator may be set on pedestals or high leg radiators.

A method of connecting a unit heater to a one-pipe air-vent steam heating system is illustrated in Fig. 8.

Where radiators are located below the level of the steam main the drop to the radiator is dripped into the wet return. An air vent is used to vent the end of the radiator, or the radiator may have a return connection for the condensate at the end opposite the supply end. If connected in this latter manner a check valve should be placed in the condensation return connection.

The method of making two-pipe connections for radiators of the old steam type is shown by Fig. 9. If the hot water type is used, which is the more recent practice, the supply tapping is at the top instead of at the bottom, as shown in Fig. 10, the runouts remaining as shown in Fig. 9.

This type of radiator connection is satisfactory for radiator connections of atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of both the up-feed and down-feed types. Short radiators, not exceeding 8 to 10 sections, may be supplied and drained from the same end as indicated in Fig. 11. On down-feed systems the bottom of the supply riser must be dripped into the return somewhat as illustrated in Fig. 12.

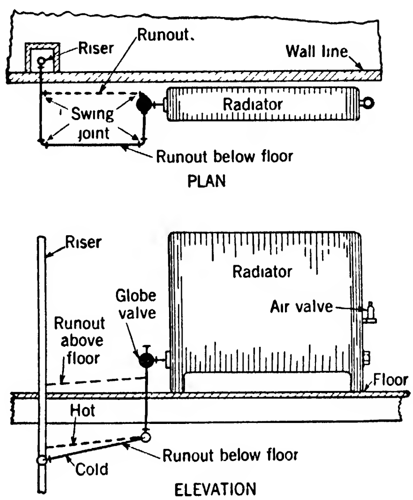


Fig 7. One-Pipe Radiator Connections

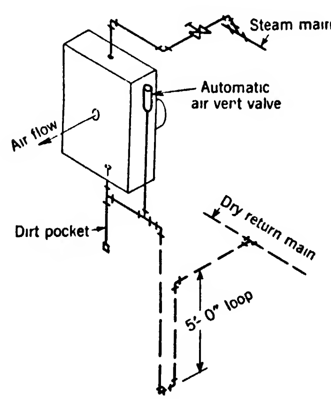


Fig. 8. Unit Heater Connected to One-Pipe Air-Vent System

Radiators which are located below the level of the steam main have the drop from the supply pipe dripped into the wet return, the return radiator connection drips to the wet return and is vented through an air line to the overhead return line through a radiator trap or air line valve, as illustrated in Fig. 13. The same method is used for connecting concealed heating units such as indirect radiators and the steam coils of air conditioning or

ventilating units where they are located at an elevation lower than the dry return pipe. Radiators located below the supply main but above the dry return main may have the drip on the steam drop to the radiator omitted if an overhead valve is used, as shown in Fig. 14. If they are located above the dry return pipe, they are connected in the manner shown by Fig. 15, which follows the same method as direct radiators similarly located.

Convector Connections

Convectors are often installed without control valves, a damper being used to shut off the flow of air to retard the heat transfer from the convector even though it is still supplied with steam. The piping connections for a convector with the inlet and outlet at the same end are shown in

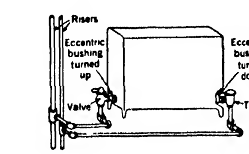


FIG. 9. CONNECTIONS TO STEAM-TYPE RADIATOR FOR TWO-PIPE SYSTEM

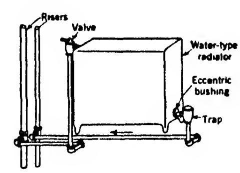


FIG. 10. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

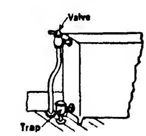


FIG. 11. TOP AND BOTTOM RADIATOR CONNECTIONS

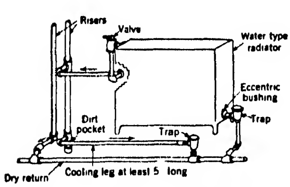


FIG. 12. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

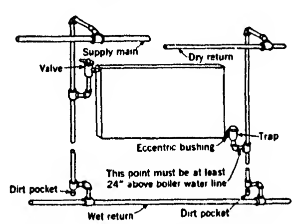


FIG. 13. CONNECTIONS TO RADIATOR HUNG ON WALL

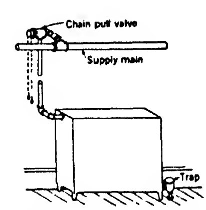


FIG. 14. CONNECTING DROP RISER DIRECT TO RADIATOR

Fig. 16. There is no valve on the steam side but there is a thermostatic trap on the return. The damper for control is shown immediately above the convector. This piping is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of the up-feed type. A similar unit with connections on opposite ends and suitable for the same systems is shown in Fig. 17. This unit has no damper but requires a valve on the steam connection for control. When valves must be located so as to be accessible from the supply air grille, the arrangement usually takes the form indicated in Fig. 18. A convector located in the basement and supplying air to a room on the floor above may be piped as pictured in Fig. 19 for all systems except gravity one-pipe or two-pipe systems. Convectors with damper control, installed in cabinets or under window-sills, usually are connected as shown in Fig. 20.

Pipe Coil Connections

Pipe coils, unless coupled in a correct manner, often give trouble from short circuiting and poor circulation. The method of connecting shown

in Fig. 21 is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

Pipe Connections for Indirect Heating Units

The coils or heat exchangers used to temper the air for ventilation and air conditioning are supplied with steam using the same piping principles as are used for connecting radiators for the given type of system. If the tempering coil is located at an elevation lower than the dry return piping, the connection would be as shown by Fig. 22 for a two-pipe vapor system. For a one-pipe system the air line through the thermostatic trap would be omitted and the usual air vent substituted.

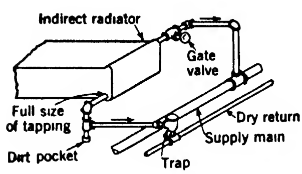


FIG. 15. PIPING CONNECTIONS TO INDIRECT RADIATORS

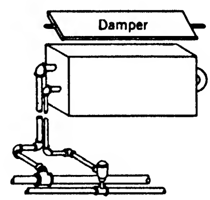


FIG. 16. CONVECTOR CONNECTIONS SAME END

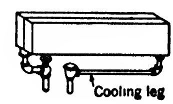


FIG. 17. HORIZONTAL FIN-TYPE HEATING UNIT

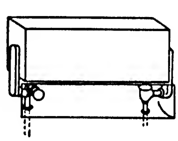


FIG. 18. HEATING UNIT VALVES BEHIND GRILLE

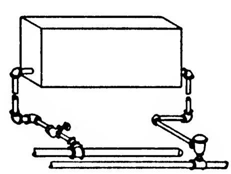


FIG. 19. HEATING UNIT WITH VALVES IN BASEMENT

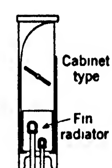


FIG. 20 FIN-TYPE HEATING UNIT IN CABINET

Where a building is served by a vacuum system or a sub-atmospheric system the air heaters should be piped in the usual manner and traps of large capacity, preferably of the combination float and thermostatic type, should be used. In an orifice system, traps should be used on the returns so that a pressure above that of the atmosphere may be secured on the heaters. A similar connection may be used for a *closed* two-pipe vapor system using a condensation pump with receiver.

In connecting air heaters a branch steam main is directed to the fan room and a single branch is connected to each row of heating units. Each of these branches is split into as many connections as are needed for each row, governed by the number of stacks and width of stacks. Each stack must have at least one steam connection. Wide stacks are more evenly heated with two steam connections, one at each end, the stacks being divided and a return connection provided for each steam connection. The return connection should use a nipple full size of the outlet tapping and reduce the pipe size to the normal return size as required by use of a reducing elbow as shown in Fig. 23.

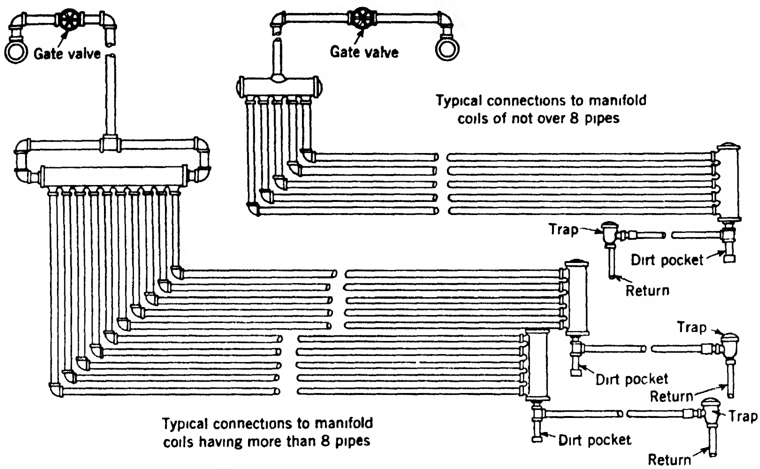


FIG 21 TYPICAL PIPE COIL CONNECTIONS

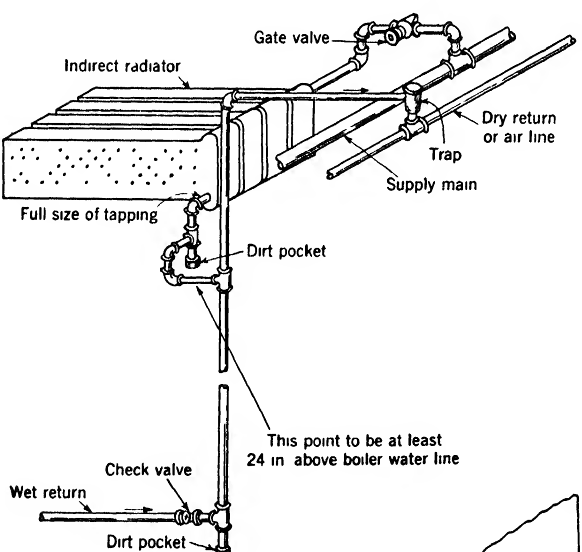


FIG. 22. TYPICAL PIPING CONNECTIONS TO CONCEALED HEATING UNITS WITH WET RETURNS

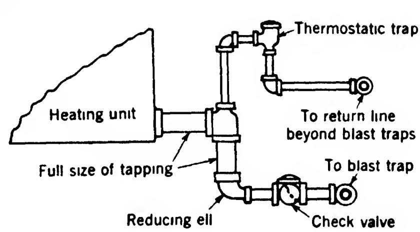


FIG. 23. HEATING UNIT RETURN CONNECTION WITH SEPARATE AIR LINE

The piping shown in Fig. 24 is for small stacks and has the steam connected at only one end. On the return side all of the returns are collected together through check valves and are passed through blast traps which are connected to the vacuum return or to an atmospheric return. The air from the stacks, in the case illustrated, passes up into a small air line and through a thermostatic trap into a line connecting into the return beyond the blast trap.

Where the stacks contain some thirteen or more sections, an auxiliary air tapping is made to the lower portion of one of the middle sections, in the manner illustrated in Fig. 25, to prevent air collecting at this point.

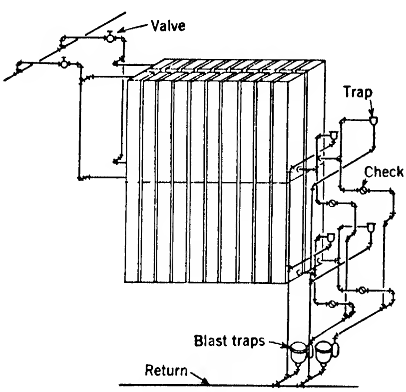


FIG. 24. SUPPLY AND RETURN CONNECTIONS FOR HEATING UNITS OF CENTRAL FAN SYSTEMS

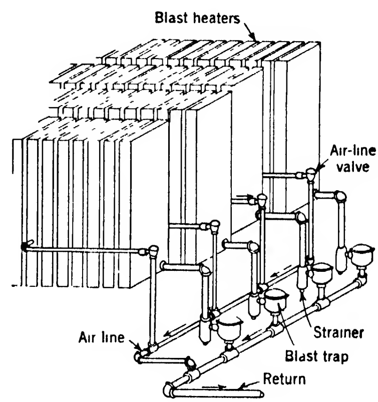


FIG. 25 TYPICAL CONNECTIONS TO CENTRAL FAN SYSTEM HEATING UNITS EXCEEDING 12 SECTIONS

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_l - t_e)}{55.2 \times 240} = \frac{Q \times (t_l - t_e)}{220.8} \qquad (1)$$

where

EDR = equivalent direct radiation, square feet.
Q = volume of air, cubic feet per minute.
t_e = the temperature of the air entering the row of heating units under consideration, degrees Fahrenheit.
t_l = the temperature of the air leaving the row of heating units under consideration, degrees Fahrenheit.
60 = the number of minutes in one hour.
55.2 = the number of cubic feet of air heated 1 F by 1 Btu.
240 = the number of Btu in 1 sq ft of EDR.

Example 4. Assume that the heating units shown in Fig. 26 are handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to

65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the in-

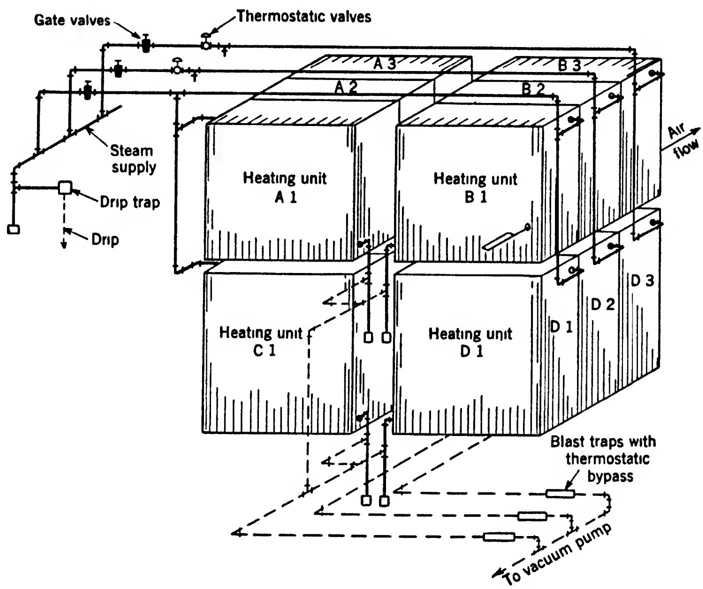


FIG. 26. TYPICAL PIPING FOR ATMOSPHERIC AND VACUUM SYSTEMS WITH THERMOSTATIC CONTROL (CENTRAL FAN SYSTEM)

Solution. For row 1,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For row 2,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For row 3,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	TOTAL LOAD (EDR)	STACK LOAD[a] (EDR)	CONNECTION LOAD[b] (EDR)
1	9058	2265	2265 or 1132
2	5661	1415	1415 or 708
3	3397	849	849 or 425

[a]One quarter of total row load.

[b]One half of stack load if two steam connections are made; otherwise, same as stack load

direct heating units on a separate system and not on supply or return lines connected to the general heating system.

DRIPPING

A steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward, and with the direction of steam flow. Any steam main in any heating system can be elevated if dripped. Fig. 27 shows a connection where the steam main is raised and the drain is to a wet return. If the elevation of the low point is above a dry return it may be drained through a trap to the dry return in two-pipe vapor, return, vacuum return line and sub-atmospheric systems. Horizontal steam pipes may also be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for the condensation (Fig. 28). Horizontal return pipes may be carried past doorways and other obstructions by using the scheme illustrated in Fig. 29. It will be noted that the large pipe, in this case, runs below the obstruction and the smaller one over it; in vacuum systems it is well to have a gate valve in the air line.

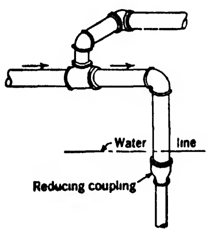


FIG. 27. DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

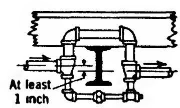


FIG. 28. LOOPING MAIN AROUND BEAM

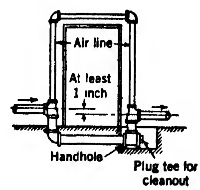


FIG. 29. LOOPING DRY RETURN MAIN AROUND OPENING

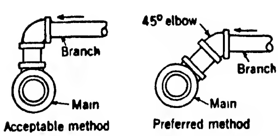
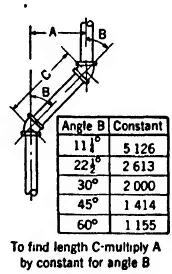


FIG. 30 METHODS OF TAKING BRANCH FROM MAIN



Angle B	Constant
11¼°	5 126
22½°	2 613
30°	2 000
45°	1 414
60°	1 155

FIG. 31. CONSTANTS FOR DETERMINING LENGTH OFFSET PIPE

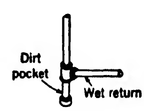


FIG. 32. DIRT POCKET CONNECTION

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 30, but where radiator condensation does not flow back into the main the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing without distorting the angle of the pipe. Runouts from the steam main

are usually made about 5 ft long to provide flexibility for movement in the main.

Offsets in steam and return piping should preferably be made with 90-deg ells but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 31.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck which will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 32.

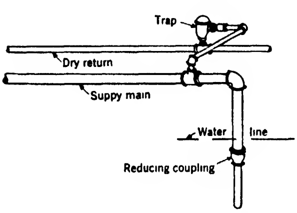


FIG. 33 DRIPPING END OF MAIN INTO WET RETURN

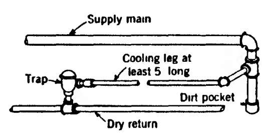


FIG. 34. DRIPPING END OF MAIN INTO DRY RETURN

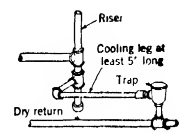


FIG 35. DRIPPING HEEL OF RISER INTO DRY RETURN

On vapor systems where the end of the steam main is dripped down into the wet return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry return line as shown in Fig. 33. On vacuum systems the ends of the steam mains are dripped and vented into the return through drip traps opening into the return line. The same method may be used in atmospheric systems. A float type trap is preferable to a thermostatic trap for dripping steam mains and large risers. If thermostatic traps are used a cooling leg (Fig. 34) should always be provided. The cooling leg is for cooling the condensation sufficiently before it reaches the trap so the trap will not be held shut by too high a temperature. On down-feed systems of atmospheric, vapor, and vacuum types, the bottom of the steam risers are dripped in the manner shown in Fig. 35. On large systems it is desirable to install a gate valve in the cooling leg ahead of the trap.

PROBLEMS IN PRACTICE

1 ● What factors determine the size of steam piping and the allowable limit of capacity?

Factors which determine the size of steam piping are the desired initial pressure and the allowable drop in pressure which is permissible to maintain a pressure in the farthest radiator. The length of run in sizing piping is important and it is generally considered as the distance along the piping from the source of steam supply to the farthest radiator, with allowances for resistance of elbows and valves expressed in terms of equivalent length.

2 ● When the size of pipe is still undetermined, what arbitrary percentage is usually added to the actual length to obtain the equivalent length?

Usually 100 per cent; in other words, the actual length is doubled to allow for the added drop produced by the valves, tees, elbows, and other fittings.

3 ● What are the major factors to be considered in determining the flow of steam in pipes?

a. The initial steam pressure available and the total pressure drop allowable between the source of steam supply and the end of the return system. The pressure drop should never exceed one half of the initial pressure.

b The maximum steam velocity allowable When condensate is flowing against the steam, the velocity must not be so great as to produce water hammer, or hold up water in parts of the system until the steam flow is reduced sufficiently to permit the water to pass The velocity at which disturbances take place depends upon.

1. Size of pipe.
2. Whether pipe is vertical or horizontal
3. Pitch or grade of pipe.
4. Quantity of water flowing against steam.

c. The equivalent length of run from the source of steam supply to the farthest heating unit, with allowance for friction in pipe fittings and valves

4 ● Name three fundamental considerations in designing the piping system for steam heating.

a. Provision for the distribution of suitable quantities of steam to the various heating units

b. Provision for the return of condensate from the radiators and piping to the boiler.

c. Provision of means for expelling air from the radiators and piping

5 ● Why is the proper reaming of the ends of pipe necessary?

The capacities of pipes depend upon the free area available for flow. In cutting the pipe this area may be restricted by a burr, which may decrease the capacity of a pipe more than 25 per cent in the smaller pipe sizes

6 ● a. What are the major factors to be considered when selecting a pressure reducing valve?

b. How should such valve be installed?

a. The initial pressure of the steam must be considered along with the desired reduced pressure. The connected load to be supplied must be known in square feet of equivalent direct radiation or in pounds of steam per hour. For operation with a continuous load, a semi-balanced or double-seated valve operated by a diaphragm gives good results Where the load is intermittent, as in process work or with thermostatically controlled blast heaters, a so-called dead end or single-seated valve should be used.

b The pressure reducing valve should be installed in a horizontal line with a gate valve on each side, and with a by-pass operated by a valve. The pressure balancing pipe from the diaphragm chamber should be connected into the top or side of the low pressure main not less than 15 ft from the reducing valve.

7 ● What is the usual expansion allowance and how it is compensated for in heating system supply risers?

The expansion of low pressure steam piping is normally taken as 1¼ to 1½ in. per 100 ft of pipe With a five story building a double swing connection between the riser and the main will suffice. In buildings between 5 and 10 stories high the riser should be anchored near its center and have double swing connections to the main. For taller buildings expansion loops or riser offsets are used which are capable of handling a length of riser reaching 5 stories in either direction from the joint. The risers are anchored at each alternate 5 stories. All radiators must have double swing connections, and those connected above where the riser is anchored must be given greater pitch to insure their having proper grade when the riser is heated.

8 ● Why should all boiler steam supply tappings be used full size?

In order to operate at low steam velocities so the water in suspension can separate from the steam and remain in the boiler.

9 ● What is the Underwriters Loop or the Hartford Connection?

An arrangement of piping on the returns to low pressure boilers wherein the return line is raised up nearly to the water line of the boiler and is then dropped back and connected to the boiler return inlet; the high point is connected by a balanced pipe to the steam runout from the boiler on the boiler side of all stop valves. With this loop no check valve is required on gravity systems, and water cannot be backed out of the boiler and into the return at a point lower than the invert of the pipe at the top of the loop.

10 ● What are the important factors in making radiator connections?

Connections to radiators should be made as direct as possible, of proper size, with ample pitch of piping and allowance for expansion.

11 ● Why should careful attention be given to proper dripping and drainage of steam piping?

The steam mains and risers must be quickly drained of condensate and where necessary vented of air in order to obtain a sufficient supply of steam to the radiators. Proper drainage is also necessary to insure a noiseless heating system.

12 ● What is the limit of pressure drop usually recommended in a vacuum system?

Not over ⅛ lb (2 oz) per 100 ft of equivalent run, and not over 1 lb total drop.

13 ● When steam and condensation are flowing in the same direction, what is the maximum total pressure drop which should be used?

The maximum total pressure drop should not exceed one half of the initial steam pressure.

14 ● What does a proper installation of a pressure reducing valve include?

A strainer in front of the pressure reducing valve; a gate valve in front of the strainer; a gate valve after the reducing valve; a by-pass around the two gate valves, strainer, and pressure reducing valve; and a globe valve in the by-pass Sometimes a safety valve on the low pressure side and pressure gages on both sides are installed. The high pressure line should be dripped just before the high pressure steam enters the pressure reducing valve assembly.

15 ● Will a pressure reducing valve which is reducing the steam pressure from 100 lb gage to 50 lb gage pass more or less steam than the same valve when reducing the steam pressure from 100 lb gage to 5 lb gage?

The valve will pass practically the same volume of steam in each case as the velocity of steam flowing through an orifice shows no material increase after the reduced absolute pressure has fallen to 58 per cent of the initial absolute pressure. Because of its greater density, the weight of steam passed will be greater in the case of the reduction to 50 lb gage.

Chapter 17

HOT WATER HEATING SYSTEMS AND PIPING

One- and Two-Pipe Systems, Mechanical Circulation, Circulators, Iron Pipe and Copper Tube Sizes, Gravity Circulation, Expansion Tanks, Relief Valves, Installation Details

THE various forms of hot water heating may be fundamentally classified according to motive force, namely, forced circulation or gravity flow. Forced circulation is accomplished by the use of centrifugal or propeller type pumps which are especially designed for this particular type of application. Gravity flow is maintained by the difference in weight of the water in the flow and return mains.

These systems may be further classified as to high or low operating water temperatures. Higher water temperatures permit a reduction in radiator size. A large temperature differential between the flow and return results in smaller pipe sizes as also does the use of forced circulation. Light wall copper tubing has recently been introduced to supplement the customary black iron piping which has been used for these systems in the past.

Low temperature water(150 to 180 F) is generally that which provides a heat emission per square foot of radiation of from 150 to 165 Btu while a high temperature water (200 to 220 F) will deliver from 200 to 240 Btu.

The use of high temperature water in a heating system is desirable as the maximum outside temperature for which the system is designed will occur for a relatively short time during the average season. The increased use of automatic heating equipment with more accurate controls, makes it possible to use higher temperatures and smaller heating units without sacrificing good design.

The unit, a square foot of equivalent direct radiation, EDR, has been used for many years for rating purposes in both steam and hot water systems, but its use, especially in hot water systems, has always resulted in complications and confusion. It is the plan of THE GUIDE *to eventually eliminate this empirical expression and to substitute a logical unit based on the Btu. The* Mb, *the equivalent of 1000 Btu and the* Mbh, *the equivalent of 1000 Btu per hour, which have been approved by the* A.S.H.V.E. *are used in this chapter on hot water systems to replace the square foot of radiation formerly used.*

In designing a piping arrangement for a hot water heating system, it is necessary to observe the fundamental rule that the total friction head in any circuit must not exceed the pressure head available for circulating the water. It is necessary to size the pipe in any circuit, so that the friction loss produced by the movement of a sufficient volume of water to handle the heating load will not be greater than the available head.

In designing a hot water heating system, it is necessary to determine:

1. The heat losses of the rooms or spaces to be heated (See Chapter 7)
2. The size and type of boiler (See Chapter 13)
3. The location, type, and size of heating units (See Chapter 14).
4. The method of piping.
5. The type and size of circulating pump (if forced circulation)
6. Suitable pipe sizes.
7. The type and size of expansion tank

ONE- AND TWO-PIPE SYSTEMS

Piping systems may be divided into two general types, namely, one-pipe and two-pipe systems. These fundamental piping layouts may differentiate between up-flow, down-flow and zoned systems. Also the type of riser and radiator connection may vary considerably. Zoning is important in modern design and it is accomplished by dividing the system into a number of circuits and controlling each circuit individually. In a two-pipe system the piping is arranged so that the water flows through only one radiator during a circuit through the system, so that all radiators are supplied with water at practically the same temperature as that in the boiler. In some one-pipe systems, the water flows through more than one radiator during its circuit. In that case, the first radiator receives the hottest water; the second radiator, somewhat cooler water; the third one, still cooler; and so on. As the temperature of the water supplied to a radiator is lowered, the size of the radiator must be increased and, consequently, the total heating surface for a one-pipe system must be greater than for a two-pipe system for the same requirements. As the velocity is *increased* in a one-pipe system, the drop in temperature is *decreased*, so that water at a higher average temperature is delivered to the radiators. This means that the radiators at the end of the main can be sized on the same basis as the radiators at the beginning of the main. If the system is correctly designed, the resulting error is less than the variation in calculating the heating load for the enclosure.

By making use of improved devices now available, one-pipe forced circulation systems may be calculated by the same procedure described later for two-pipe systems. Operation may be obtained as satisfactory as with a two-pipe system.

Two-pipe systems may be divided into two classes, *direct return* systems (Fig. 1), and *reversed return* systems (Fig. 2). In a direct return system the water returns to the heater by a direct route after it has passed through its radiator and, as a result, the paths through the three radiators shown in Fig. 1 are of unequal lengths, the path through the first radiator being the shortest and that through the third radiator, the

longest. In a reversed return system, the water returns to the heater by an indirect route after it has passed through the radiators, so that the paths leading through the three radiators shown in Fig. 2 are practically of equal length.

The reversed return system has an advantage over the direct return system in that it is more likely to function satisfactorily even though the pipe system is not accurately designed For example, if in Fig. 2 all pipes are of one size, each of the three radiators will receive approximately the same quantity of hot water because the three paths are practically of equal length, whereas in Fig. 1, if all pipes are of the same size, Radiator 1 will receive more water than the others because the path through it is shorter than those through the other radiators. As a result, Radiator 1 will be filled with water at a higher average temperature than the remaining two radiators, and will therefore dissipate more heat. To prevent this unequal distribution of heat it is necessary to throttle the paths through Radiators 1 and 2 so that the friction heads of the three paths are equal when each radiator receives its proper quantity of water.

The two-pipe direct return system, with its inherent lack of balance, is the least satisfactory type of piping possible, yet is the most widely used.

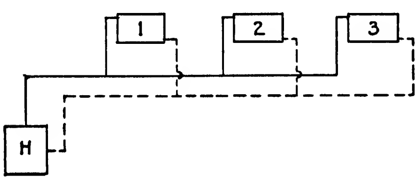


FIG. 1. A DIRECT RETURN SYSTEM

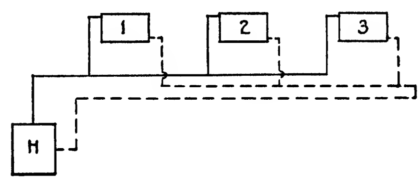


FIG. 2. A REVERSED RETURN SYSTEM

The modern applications of automatic heating require a system to be very nearly in balance so that uniform distribution of heat will be obtained.

Two-pipe systems must be balanced first by calculation and then by test after the plant is in operation. Unbalanced conditions in a forced circulation system are more detrimental to satisfactory operation than in the system circulated by gravity. The selection of orifices for correcting the unbalance must be more accurate. Due to the variations in water delivery from pipes, the accuracy of calculations is decreased, so that more reliance must be placed on actual test work. This is always costly and seldom completely satisfactory.

A comparison of Fig. 1 and Fig. 2 may suggest that a reversed return system requires considerably longer mains than a direct return system. This is not always the case, as will be noted from the reversed return system of Fig. 3.

MECHANICAL CIRCULATION AND CIRCULATORS

The designer of a forced circulation system generally makes use of the pumps commercially available. Pumps of this type will have characteristics which govern the water velocity selected for the heating system. However, available pumps generally have a sufficient range of capacities

to promote the selection of an economical velocity. If a system is designed to handle a load of 96 Mbh with a 20 F drop allowable in the system, a circulating pump will be required, handling about 10 gpm and at a head pressure high enough to allow a satisfactory friction drop in the system.

Frequently water velocities are selected which produce objectionable noises in the system. A velocity of over 4 fps is apt to cause noise in the smaller pipes and tubes. Velocities higher than this value will cause no objectionable trouble in industrial applications.

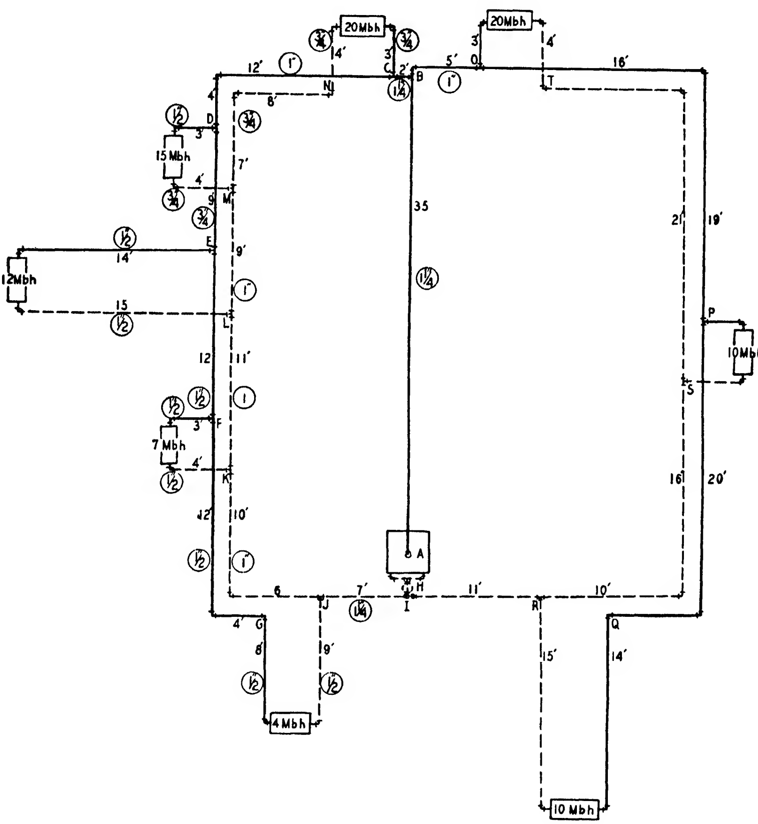


FIG. 3. A FORCED CIRCULATION REVERSED RETURN SYSTEM[a]

[a]Note that the numbers on the radiators indicate thousands of Btu per hour (Mbh) and not square feet

Low head centrifugal pumps especially designed for hot water systems are used to provide the necessary head pressure for forced circulation and to improve the operation of an improperly designed or installed gravity system. These pumps are designated by the nominal pipe size of their connection, but the selection of the pump should be governed by the capacity curves and not by the nominal pipe size. These pumps operate with little noise and low power consumption, both of which are features of prime importance to the satisfactory operation of a forced

circulation system. They are designed for installation directly into the heating main and require no other support. The common practice is to install them in the return line but where desirable there is no objection to their location in the supply line. Gate valves should be installed in either side of the pump so that it can be removed without draining a system. A by-pass is not necessary as the friction drop through the pump is not sufficient to prevent gravity recirculation if the pump should become inoperative.

Propeller type pumps are also available for hot water service, generally being built into a fitting and are made in all of the commercial pipe sizes commonly used in heating. They are installed in the same manner as a centrifugal pump.

Forced circulation lends itself to automatic control and the arrangement of the circuit depends entirely on the design of the system. The control may consist of a thermostat controlling both the automatic firing device and the circulator with the same type of limit control, as a safety switch. This type of control can be satisfactory, provided the radiation is properly selected and accurately located in the building. A circuit using flow control valves to regulate the gravity flow of the water when the pump is not running allows the temperature to be maintained closer to the desired setting. Under these circumstances, the circulator motor is controlled by a room thermostat while the automatic firing device is controlled by a limit switch with a safety device in series.

For exceptionally large installations, such as central heating plants circulating pumps of the centrifugal single stage type having an average operating efficiency of 70 per cent against heads up to 125 ft are sometimes used. In some cases it is advisable to install pumps in duplicate to provide for contingencies and to insure continuous operation. In such cases, each pump should be made equal to the maximum capacity required.

PIPE SIZES

The pressure heads available in forced circulation systems are much greater than those in gravity circulation systems, consequently, higher velocities may be used in designing the system, with the result that smaller pipes may be selected and the first cost of the installation reduced. As the pipes of a heating system are reduced in size, the necessary increase in the velocity of the water increases both the cost of operation and the initial cost of the circulating equipment. The increased velocity of a forced circulation system offers a number of advantages, such as a much shorter heating-up period and a more flexible control of hot water circulation. This improved performance merits the small increase in operating cost necessary to mechanically circulate the system. The velocity required should be determined by calculation for the particular system under consideration.

Since the velocities in forced circulation systems are higher than those in gravity circulation systems, and since the friction heads in a heating system vary almost as the squares of the velocities, a given error in the calculation or assumption of a velocity is less important in a forced circulation system than in a gravity circulation system and, consequently, it

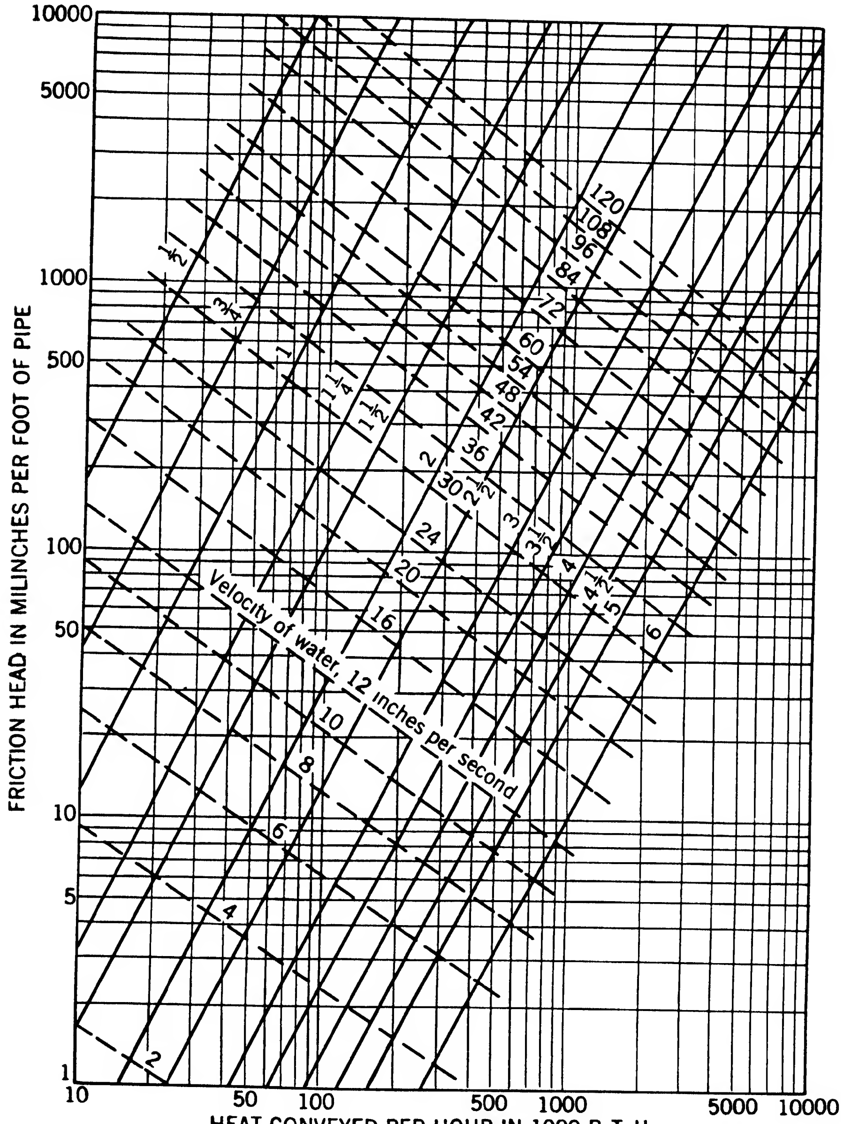


FIG. 4. FRICTION HEADS IN BLACK IRON PIPES FOR A 20 F TEMPERATURE DIFFERENCE OF THE WATER IN THE FLOW AND RETURN LINES

For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1 5

is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

FORCED CIRCULATION

In designing a forced circulation system, black iron pipe sizes may be selected from either Fig. 4 or Table 1, both of which are based on a 20 F temperature difference between the flow and return lines. For other temperature drops, the pipe capacities may be changed to correspond to the desired differentials. Research data are lacking for determining the capacities of copper tube sizes. In the absence of complete test data at the present time, capacities are given in Table 2 for type *L* copper tube sizes which are based on a recently developed hydraulic formula[1]. The friction heads of boiler, radiator valve and tee may be expressed in terms of friction head in one elbow according to the values given in Table 3 for iron pipe, and Table 4 for copper tubing.

The following examples will illustrate the procedure to be followed in designing forced circulation systems.

Example 1. From the plan of Fig. 3 note that the longest circuit consists of 151 ft of iron pipe; 1 boiler; 1 radiator; 1 radiator valve; 1 stop cock; 10 ells and 3 tees; and the shortest circuit consists of 127 ft of pipe; 4 tees; 1 boiler; 1 radiator; 1 radiator valve; 1 stop cock; and 6 ells. Design the piping for this system.

Solution. The friction in the various fittings can be expressed in terms of the friction in a 90-deg elbow from the values given in Table 3. The longest circuit consists of 151 ft of pipe and 44 elbow equivalents. The short circuit consists of 127 ft of pipe and 39 elbow equivalents.

The friction head in one elbow is approximately equal to the friction produced by the same sized pipe 25 diameters in length Assume that the average pipe size for this system is 1 in The equivalent length of the longest circuit will be 151 ft plus 100 ft or 251 ft of pipe. The equivalent length of the short circuit will be 217 ft.

Having determined the equivalent length of the circuits, the next step is to assume the rate at which the water is to be circulated in the system The water may flow through the system so that it will cool any reasonable number of degrees. For the most economical average system a 20 F drop seems to be a satisfactory rate This entails a slower water flow from the pumping equipment with a reasonable relationship between pipe size and flow. Assume 20 F drop for this system. One gallon of water per minute with a density of 7.99 at 215 F will deliver approximately 9600 Btu per hour with a 20 F drop. The total radiation load is 98 Mbh, therefore the pump must deliver 10.2 gpm or 4900 lb of water per hour.

Knowing that the rate of flow is 10.2 gpm, the next step is to determine from the characteristics of available pumps, which one will produce a satisfactory velocity in the system. Assume that 4 pumps are available for this load which will produce 10 2 gpm at pressure heads of 2, 5, 10 and 18 ft At these heads the pumps would produce a velocity high enough to make available a friction head per foot of pipe of 96, 240, 480 and 860 milinches per foot respectively. If 95 milinches per foot were used, the gravity head at 215 F average temperature in the mains would be 26 per cent of the total head and should be considered in sizing the system At 240 milinches per foot the gravity effect is 10 per cent and as this is lower than the delivery variation from the pipe used, it can be neglected At 480 and 860 milinches the gravity effect is still a smaller percentage of the total, but at these losses in the average system the cost of pumping will more than offset the advantage gained in pipe sizes. Therefore, pipe size this system at 240 milinches per foot which is equivalent to a total loss of 60,000 milinches for the 250 ft equivalent length of pipe.

[1]Hydraulic Service Characteristics of Small Metallic Pipes, by G M Fair, M. C Whipple and C. Y. Hsiao (*Journal of the New England Water Works Association*, Vol. XLIV, No 4, 1930)

TABLE 1. CAPACITIES FOR BLACK IRON PIPE

A = *Carrying capacities in Mbh*
B = *Velocity in inches per second*

HEAD LOSS, FT	MILINCH FRICTION LOSS PER FOOT OF PIPE													
	720	480	360	300	240	180	160	144	120	96	90	80	70	60
	EQUIVALENT LENGTH OF PIPE IN FEET (LONGEST CIRCUIT)													
2	33	50	66	80	100	133	150	167	200	250	270	300	340	400
2½	42	62	84	100	125	167	188	208	250	312	333	375	428	500
3	50	75	100	120	150	200	225	250	300	375	400	450	510	600
3½	59	87	117	140	175	233	263	291	350	437	463	525	593	700
4	67	100	133	160	200	266	300	333	400	500	533	600	685	800
4½	75	112	149	180	225	300	338	374	450	562	593	675	758	900
5	83	125	167	200	250	333	375	416	500	625	666	750	860	1000
5½	92	137	183	220	275	366	413	457	550	687	713	825	923	1100
6	100	150	200	240	300	400	450	500	600	750	800	900	1030	1200
6½	108	162	217	260	325	433	488	540	650	812	843	975	1088	1300
7	116	175	233	280	350	465	525	580	700	875	933	1050	1200	1400
7½	124	187	249	300	375	500	563	623	750	937	973	1125	1252	1500
8	133	200	266	320	400	533	600	666	800	1000	1070	1200	1370	1600
8½	142	212	283	340	425	566	638	706	850	1062	1103	1275	1417	1700
9	150	225	300	360	450	600	675	750	900	1125	1200	1350	1540	1800
9½	159	237	317	380	475	633	713	789	950	1187	1233	1425	1577	1900
10	167	250	333	400	500	666	750	833	1000	1250	1333	1500	1715	2000
10½	175	262	349	420	525	700	788	872	1050	1312	1363	1575	1737	2100
11	183	275	366	440	550	733	825	916	1100	1375	1466	1650	1885	2200
11½	192	287	383	460	575	766	863	955	1150	1437	1533	1725	1897	2300
12	200	300	400	480	600	800	900	1000	1200	1500	1600	1800	2030	2400
NOMINAL PIPE SIZE, IN.	CAPACITY OF PIPES *Mbh* WITH A 20 F[a] DROP													
½ A	20	16	14	13	11	10	9	9	8	7	7	6	6	5
½ B	*27*	*22*	*19*	*17*	*15*	*13*	*12*	*11*	*10*	*9*	*9*	*8*	*8*	*7*
¾ A	43	35	30	27	24	21	19	18	17	15	14	13	12	11
¾ B	*33*	*25*	*23*	*21*	*18*	*16*	*15*	*14*	*13*	*11*	*11*	*10*	*9*	*9*
1 A	85	70	60	54	48	41	39	36	33	30	28	27	25	23
1 B	*39*	*32*	*27*	*25*	*22*	*19*	*18*	*17*	*15*	*13*	*13*	*12*	*11*	*10*
1¼ A	180	145	125	115	98	85	80	75	68	60	58	55	51	47
1¼ B	*48*	*39*	*33*	*30*	*27*	*23*	*21*	*20*	*19*	*16*	*15*	*15*	*14*	*12*
1½ A	285	230	195	180	160	135	125	120	110	96	92	88	82	75
1½ B	*54*	*44*	*38*	*34*	*30*	*26*	*24*	*23*	*21*	*19*	*18*	*17*	*15*	*14*
2 A	540	435	370	340	300	255	240	230	205	180	175	165	150	140
2 B	*64*	*52*	*45*	*40*	*36*	*30*	*29*	*27*	*24*	*22*	*21*	*20*	*19*	*17*
2½ A	890	720	610	550	480	420	390	370	330	300	280	270	250	230
2½ B	*74*	*60*	*50*	*46*	*41*	*35*	*33*	*31*	*28*	*24*	*24*	*22*	*21*	*19*
3 A	1650	1340	1130	1000	900	760	720	670	600	540	520	480	450	410
3 B	*88*	*70*	*60*	*54*	*48*	*41*	*38*	*36*	*33*	*29*	*28*	*26*	*24*	*22*
3½ A	2500	2000	1700	1500	1350	1150	1080	1000	900	800	760	720	670	620
3½ B	*99*	*78*	*66*	*60*	*54*	*46*	*43*	*40*	*36*	*32*	*31*	*29*	*27*	*25*
4 A	3500	2800	2400	2200	1900	1600	1520	1440	1300	1150	1100	1050	960	880
4 B	*110*	*87*	*74*	*66*	*58*	*50*	*47*	*45*	*40*	*35*	*34*	*32*	*30*	*27*
5 A	7000	5600	4700	4300	3700	3200	3000	2750	2500	2200	2100	2000	1800	1700
5 B	*132*	*106*	*90*	*80*	*70*	*60*	*56*	*53*	*48*	*42*	*41*	*38*	*35*	*32*
6 A	12 000	9200	7800	7000	6200	5200	4800	4600	4100	3600	3500	3300	3000	2800
6 B	*156*	*124*	*104*	*94*	*82*	*69*	*64*	*61*	*55*	*48*	*46*	*44*	*41*	*37*

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F the capacities shown in this table are to be multiplied by 1.5

TABLE 2. CAPACITIES FOR TYPE *L* COPPER TUBE

A = *Carrying capacity in Mbh*
B = *Velocity in inches per second*

Head Loss Ft	Milinch Friction Loss per Foot of Tube											
	720	600	480	360	300	240	180	150	120	90	75	60
	Equivalent Length of Tube in Feet (Longest Circuit)											
2	33	40	50	67	80	100	133	160	200	267	320	400
2½	42	50	63	83	100	125	167	200	250	333	400	500
3	50	60	75	100	120	150	200	240	300	400	480	600
3½	58	70	88	117	140	175	233	280	350	467	560	700
4	67	80	100	133	160	200	267	320	400	533	640	800
4½	75	90	113	150	180	225	300	360	450	600	720	900
5	83	100	125	167	200	250	333	400	500	667	800	1000
5½	92	110	138	183	220	275	367	440	550	733	880	1100
6	100	120	150	200	240	300	400	480	600	800	960	1200
6½	108	130	163	217	260	325	433	520	650	867	1040	1300
7	117	140	175	233	280	350	467	560	700	933	1120	1400
7½	125	150	188	250	300	375	500	600	750	1000	1200	1500
8	133	160	200	267	320	400	533	640	800	1067	1280	1600
8½	142	170	213	283	340	425	567	680	850	1133	1360	1700
9	150	180	225	300	360	450	600	720	900	1200	1440	1800
9½	159	190	238	317	380	475	633	760	950	1267	1520	1900
10	167	200	250	333	400	500	667	800	1000	1333	1600	2000
10½	175	210	263	350	420	525	700	840	1050	1400	1680	2100
11	183	220	275	367	440	550	733	880	1100	1467	1760	2200
11½	192	230	288	383	460	575	767	920	1150	1533	1840	2300
12	200	240	300	400	480	600	800	960	1200	1600	1920	2400

Nominal Tube Size, In	Capacity of Tubes *Mbh* With a 20 F[a] Drop												
⅜	A	10	9	8	6.8	6.2	5.4	4.6	4	3.6	3	2.8	2.4
	B	*27*	*24*	*21*	*18*	*16.5*	14	*13*	11	*10*	*8.5*	*8*	*7*
½	A	20	18	16	13.5	12	10.8	9	8	7	6	5.4	4.7
	B	*33*	*30*	*25*	*21*	*19*	*17*	*15*	*13*	*12*	*10*	*9*	*8*
⅝	A	36	30	26	22.1	20	17.8	15	13.1	11.8	9.9	9	7.9
	B	*37*	*34*	*30*	*24*	*21*	*19*	*17*	*15*	*13*	*11*	*10*	*9*
¾	A	51	46	40	34	31	28	23.2	20.5	18.1	15.3	13.9	12.1
	B	*42*	*38*	*33*	*27*	*24*	*21*	*19*	*17*	*14*	*12*	*11.5*	*10*
1	A	104	94	82	70	63	56	47	42	37	32	28	25
	B	*48*	*45*	*39*	*34*	*30*	*25*	*22*	*19*	*17*	*14.5*	*13*	*12*
1¼	A	185	169	149	125	112	100	84	75	66	56	50	44
	B	*55*	*51*	*45*	*39*	*35*	*30*	*25*	*22*	*19*	*17*	*15*	*13*
1½	A	300	270	235	200	180	160	134	120	105	90	81	71
	B	*62*	*57*	*51*	*43*	*39*	*35*	*30*	*25*	*22*	*19*	*17*	*15*
2	A	625	560	495	420	375	335	280	250	200	188	170	150
	B	*76*	*68*	*59*	*51*	*47*	*42*	*36*	*32*	*27*	*22*	*20*	*18*
2½	A	1130	1010	890	750	680	600	500	450	395	335	305	270
	B	*90*	*80*	*69*	*58*	*49*	*47*	*42*	*37*	*33*	*26*	*23*	*21*
3	A	1840	1650	1450	1210	1100	980	820	740	650	550	490	420
	B	*98*	*90*	*80*	*66*	*59*	*52*	*47*	*42*	*36*	*30*	*27*	*23*
3½	A	2750	2480	2170	1840	1650	1450	1210	1100	980	820	740	650
	B	*110*	*100*	*89*	*75*	*66*	*57*	*51*	*45*	*40*	*35*	*30*	*26*
4	A	3900	3505	3100	2600	2350	2090	1760	1580	1390	1180	1080	950
	B	*120*	*108*	*96*	*83*	*73*	*63*	*55*	*49*	*44*	*37*	*34*	*29*

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5

TABLE 3. IRON ELBOW EQUIVALENTS[a]

1 90-deg elbow	1 0
1 45-deg elbow	0.7
1 90-deg long turn elbow	0 5
1 open return bend	1.0
1 open gate valve	0.5
1 open globe valve	12 0
1 angle radiator valve	2 0
1 radiator	3 0
1 boiler or heater	3 0
1 tee	(Note[b])

[a]The loss of head in one elbow can be expressed in terms of the velocity head by the formula

$$h = \frac{v^2}{2g} \tag{1}$$

where

h = the loss of head in feet, v = the velocity of approach in feet per second, and $2g$ = 64 4 ft per second per second

[b]The loss of head in tees when water is diverted at right angles through a branch of the tee varies with the per cent diverted When the water diverted is less than 60 per cent of that approaching the tee, the loss of head, in elbow equivalents may be expressed as follows

$$h_e = \frac{v_1^2}{v_2^2} \tag{2}$$

where

h_e = the loss of head in elbow equivalents, v_1 = the velocity of approach, v_2 = the velocity of water diverted at right angles

Values in elbow equivalents for the most common percentages of water diverted in a 1x1x1 in tee are as follows

25 per cent	16 0
33 per cent	9 0
50 per cent	4 0
100 per cent	1 8

TABLE 4. COPPER ELBOW EQUIVALENTS[a]

1 90-deg elbow	1 0
1 45-deg elbow	0.7
1 90-deg long turn elbow	0 5
1 open return bend	1 0
1 open gate valve	0 7
1 open globe valve	17 0
1 radiator valve	3 0
1 radiator	4 0
1 boiler or heater	4 0
1 tee	(Note[b])

[a]The loss of head in an elbow can be expressed in terms of the velocity head by the formula

$$h = \frac{0\ 7\ v^2}{2g} \tag{3}$$

where

h = loss of head in milinches, v = velocity in inches per second, and g = acceleration of gravity (386 in per second per second)

[b]The loss of head in copper tees

$$N = 0\ 7\ \frac{(v_1^2 + v_3^2)}{v_3^2} \tag{4}$$

where

N = number of elbows that would cause the same loss as the tee when the velocity of water in the connecting pipe is v_3,
v_1 = velocity of the water in the pipe entering the tee, and
v_3 = velocity of the water in the pipe discharging from the tee at right angles to v_1

Values in elbow equivalents for most common percentages of water diverted in a 1 in x 1 in tee

100 per cent	1 2
50 per cent	4 0
30 per cent	16 0
25 per cent	20 0

The pipe sizes may be selected from Fig. 4 or from Table 1 which has been derived from Fig. 4.

Size the supply main of the longest circuit first. Section AB carries 98 Mbh. From Fig. 4 it will be noted that at 240 milinches per foot, a 1¼ in. pipe carries 98 Mbh. Therefore, use 1¼ in. pipe in Section AB. Section BO carries 40 Mbh. A 1 in. pipe carries 48 Mbh at 240 milinches per foot. Use a 1 in pipe. Section OP carries 20 Mbh and this will require ¾ in. pipe Section PQ carries 10 Mbh and requires ½ in pipe. To size the return start from the boiler and proceed backwards. Section IR carries 40 Mbh and from Fig. 4 a 1 in. pipe is required Section RS carries 30 Mbh which is only slightly over the capacity of a ¾ in. pipe, so use ¾ in. Section ST carries 20 Mbh and requires a ¾ in pipe. The radiator branches are determined in the same manner. It is evident from the chart that it is impossible to maintain a constant friction loss per foot and therefore as the delivery varies there will be a change in the desired friction loss per foot of pipe

It is desirable to check the various circuits so that if the variation from the calculated resistance is too great, it may be compensated by adding additional resistance at the proper point. This may be accomplished by sizing the short circuits by the procedure previously outlined. Prepare a chart such as Table 5 to be used in calculating the resistance of each circuit.

Section AB carries 98 Mbh with a unit head of 240 milinches per foot In section AB there are 37 ft of pipe and 1¼ in elbow. At 240 milinches per foot this is equivalent to 9600 milinches total loss in this section. Section BC carries 58 Mbh with a length of 2 ft and 4 elbows The unit loss in this section is 90 milinches per foot. Loss in this section is then 1080 milinches Section CD carries 38 Mbh and has 16 ft of pipe and 1 elbow The unit loss in 1 in pipe is 155 milinches The loss in this section is 2790 milinches The balance of the supply main and the return main are handled in a similar manner.

TABLE 5. PIPING CHECK CHART

LOAD, *Mbh*		PIPE LENGTH FT	ELBOWS	PIPE SIZE IN.	UNIT HEAD MILINCHES PER FT	FRICTION MILINCHES	TOTAL LOSS MILINCHES
Supply Main							
AB 98		37	1	1¼	240	9600	9,600
BC 58		2	4	1¼	90	1080	10,680
CD 38		16	1	1	155	2790	13,470
DE 23		9	0	¾	220	1980	15,450
EF 11		12	0	½	240	2880	18,330
FG 4		16	1	½	50	850	19,180
Return Main							
HI 98		5	5	1¼	240	4320	4,320
IJ 58		11	1	1¼	90	1260	5,580
JK 54		16	1	1	300	5400	10 880
KL 47		11	0	1	230	2530	13,410
LM 35		9	0	1	140	1260	14,670
MN 20		15	1	¾	170	2890	17,560
Radiator Circuits							
CN 20	*Supply*	3	13	¾	170	3910	
	Return	4	2	¾	170	1190	5,100
DM 15	*Supply*	3	19	½	420	9250	
	Return	4	17	¾	96	2880	12,130
EL 12	*Supply*	14	20	½	270	9180	
	Return	15	20	½	270	9450	18,630
FK 7	*Supply*	3	19	½	100	2200	
	Return	4	17	½	100	2100	4,300
GJ 4	*Supply*	8	5	½	50	650	
	Return	9	17	½	50	1300	1,950

The radiator circuits are then checked. The 20 Mbh radiator on this circuit has 3 ft of supply pipe and 13 elbow equivalents while the return is composed of 4 ft and 2 elbows The unit loss in ¾ in. pipe at this delivery is 170 milinches per foot. The total loss in the supply is 3910 milinches. The loss in the return is 1190. Total loss in the radiator circuit is 5100 milinches. Check each radiator circuit in a similar manner.

The total calculated loss for the longest circuit was determined as 60,000 milinches. The maximum loss in the short circuit is 18,630 plus 13,410 plus 15,450 or a total of 47,490 milinches. This difference is caused by the variation in length of the two circuits and may be corrected by using a flow control in the return main to supply the additional resistance or by introducing resistance into each separate circuit to compensate for the difference. A 10 per cent variation will cause no complication as the flow from the various pipes will not exactly follow the curves of Fig. 4 any closer than this value.

Example 2. Design a two-pipe direct return forced circulation system with copper tubing and fittings for the piping layout as detailed in Fig. 5, based on a 20 F temperature drop through the radiation.

The piping circuit from the boiler to the highest radiator on the farthest riser and back to the boiler is 250 ft of pipe There are about 16 elbow equivalents having an equivalent pipe length of about 50 ft, so that the total equivalent pipe length is 300 ft.

Assume that a circulator is available which will provide a pressure head of 6 ft

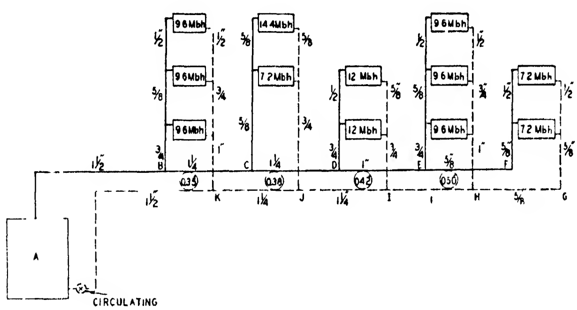


FIG. 5. A FORCED CIRCULATION DIRECT RETURN SYSTEM

Solution. Refer to Table 2, which indicates the total equivalent lengths for pressure heads from 2 to 12 ft. With a circulator having a 6 ft pressure head and a system with a total equivalent length of 300 ft, the piping system will be designed on a basis of 240 milinch.

Checking the piping diagram it will be noted that sections AB and KA, both supply 117.6 Mbh. Referring to the 240 milinch column of Table 2, 1½ in is shown to be the necessary pipe size. Sections BC and JK carry 88.8 Mbh and require 1¼ in. tubing Sections CD and IJ supply 67.2 Mbh and require 1¼ in tubing Sections DE and HI supply 43 2 Mbh, which requires 1 in. tubing. Sections EF and GH with a load of 14.4 Mbh require ⅝ in. tubing.

The risers are pipe sized in a similar manner. To secure proper distribution of hot water in the direct return system among the several risers, it is necessary to introduce resistances to balance the circuit.

The first riser is 80 ft nearer the boiler than the fifth riser. In order that the two may be balanced, that is, operated under equal pressure heads, resistance must be added to the first riser equal to the friction head in the 80 ft of supply main B to F plus the 80 ft of return main G to K for a total of 160 ft of pipe.

Having designed the piping system on a 240 milinch basis, the total friction head in the supply and return mains between the first and fifth risers is therefore 160 × 240 = 38,400 milinches, or 3 2 ft which must be supplied by additional resistance in the first riser.

Table 6. Friction Heads (in Milinches) of Central Circular Diaphragm Orifices in Unions

Diameter of Orifices (Inches)	Velocity of Water in Pipe in Inches per Second									
	2	3	4	6	8	10	12	18	24	36
					¾-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45.000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1¼-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					1½-in. Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30 000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois.(*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

This resistance can be supplied by a calibrated and adjusted modulating valve or by an orifice resistor in a union. If the orifice resistor is to be used, its size may be selected from Table 6.

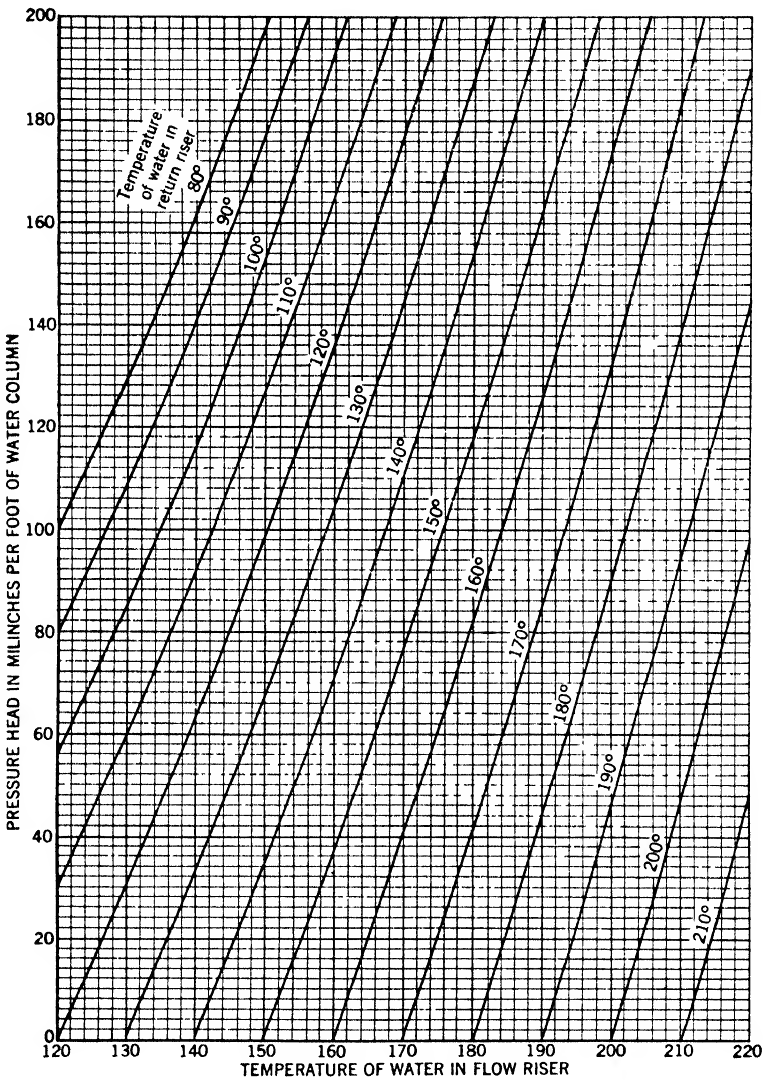


FIG. 6. GRAVITY PRESSURE HEADS FOR VARIOUS TEMPERATURE DIFFERENCES

Since the first section of riser No. 1 is ¾ in. pipe and supplies 28 8 Mbh, it may be noted from Table 2 that a corresponding velocity is approximately 22 in. per second.

From Table 6 a ¾ in. pipe with a velocity of 24 in. per second, used with a 0 35 orifice will produce a loss of 47,000 milinches. For a velocity of 22 in. per second the loss of

head will be less, probably about 41,700 milinches, which is approximately 10 per cent more than the required resistance This is permissible and the 0.35 in. orifice is selected.

The sizes of the orifice resistors for the second, third and fourth risers are selected in a similar manner and found to be 0.38, 0.42 and 0 50 in respectively.

GRAVITY CIRCULATION

In a gravity system the motive force to supply circulation is the difference in the weight of the water in the supply and the return and is proportional to the height of the risers. In this system, two distinct heads are available, the head provided in the mains by their elevation above the boiler and the head produced by the elevation of the risers above the mains. From Fig. 6 it is possible to determine the head produced per foot of height by the temperature difference to be used in designing the system A chart such as Table 1 can be arranged using Fig. 4 for black iron.

To affect a balanced circulation in a gravity hot water heating system careful consideration must be given in sizing the pipes against the amounts of water to be carried, and the head available. The larger the temperature drop, the greater the motive force available.

It is generally customary to use a heat emission of 150 Btu per square foot of radiation, which normally requires an average water temperature of 170 F in the radiator. This can be accomplished by using a 35 F drop with the water entering the radiation at 187 F and leaving at 153 F. Raising the water temperature leaving the boiler will increase the average radiator temperature and alter the heat emission of the radiator.

Assuming that the height of mains above the boiler is 4 ft and that a 35 F drop is desirable, it will be noted that from Fig. 6, a maximum temperature of 200 F and return temperature of 165 F with a pressure head of 150 milinches per foot of height will be produced. A total head of 600 milinches or 0.6 in. is thus produced in the mains. Assuming that the average height of first floor radiators to be 3 ft above the main and second floor radiators to be 12 ft, third floor radiators 21 ft and fourth floor radiators 30 ft, the circulating head will be respectively, 450, 1800, 3150 and 4500 milinches.

The data given in Fig. 4 are based on a 20 F temperature drop which may be converted for capacities of 35 F drop by multiplying the capacity by 1.75. From these data, Tables 7 and 8 may be constructed.

The most common piping layouts used in gravity design are the one-pipe system of Fig. 7 and the two-pipe system of Fig. 8. The same objections are to be found with direct return design in gravity as in forced circulation and the reverse return system of Fig. 2 is to be preferred.

Example 3. Design a one-pipe gravity circulation system for the layout shown in Fig. 7. Assume that the main circuit consists of 150 ft of pipe, 7 elbows, and one boiler.

Solution Replace the boiler by 3 elbow equivalents and assume that the size of the main will be about 2 in. According to Table 7 Column 2, a 2 in. elbow is equivalent to 4 ft of pipe, and the total equivalent length of the main will be about 150 plus 40, or 190 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main and that the temperature drop in the system is to be 35 F, Table 7 may be used to determine the size of the mains. Note from Column 8, for a 200 ft length, that a 2 in. main will supply 48 Mbh and a 2½ in. main, 75.4 Mbh. Since the system to be designed is to supply 66 Mbh, a 2 in. pipe is too small and a 2½ in. pipe

too large. The solution is to use some 2 in. and some 2½ in. pipe. Since the 2½ in. is nearer the correct size than the 2 in., select 2 in. pipe for the first 50 or 60 ft from the boiler and 2½ in. for the remaining pipe back to the boiler.

Tables 8 and 9 may be used to design the radiator risers and connections. According to Table 8, for 12 Mbh the flow riser should be ¾ in and the return riser 1 in., and the riser branches should be 1 in. and 1¼ in, respectively. Note that according to Table 9, both radiator tappings should be 1 in. To simplify the construction, select 1 in. flow risers with 1 in riser branches and 1 in radiator tappings. Also select 1¼ in. return risers with 1¼ in riser branches, and 1¼ in. radiator tappings. Similarly, for 18 Mbh, select 1¼ in. flow and return risers and riser branches, and 1¼ in. radiator tappings.

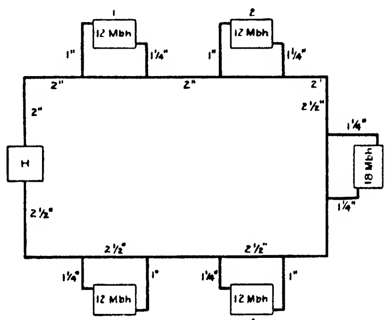


FIG. 7. A ONE-PIPE GRAVITY CIRCULATION SYSTEM

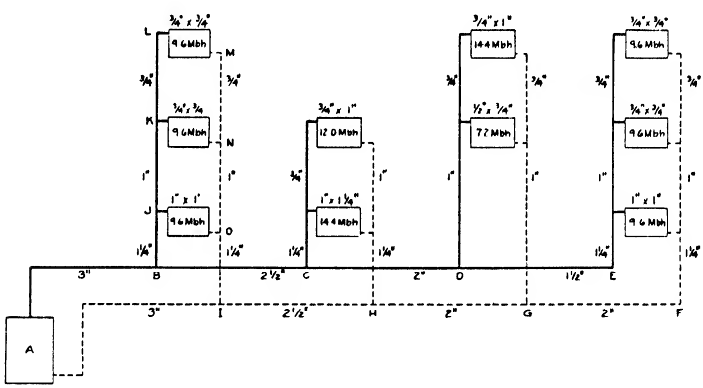


FIG 8. A TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEM

To develop a rule for determining radiator sizes, assume a system similar to that of Fig. 7, in which the total temperature drop is to be 35 F and which is equipped with 7 radiators, all radiators dissipating equal quantities of heat. The mean temperature of the water in the radiators will be reduced 5 F for each successive radiator. If the mean temperature of the water in the first radiator is 200 F, the mean temperature of the water in the seventh radiator will be 170 F, and, according to Table 4, Chapter 14, the heat dissipation of these two radiators will be to each other as 1.62 is to 1.15, or as 140 is to 100, and therefore if the last radiator is to dissipate as much heat as the first, its size must be 40 per cent larger.

TABLE 7. CAPACITIES OF MAINS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 0.6 IN., A TEMPERATURE DROP OF 35 F. WHEN THE MAINS ARE 4 FT ABOVE THE CENTER OF THE BOILER

1	2	3	4	5	6	7	8	9	10	11
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[a])	Equivalent Total Length of Pipe in Feet in Longest Circuit								
		75	100	125	150	175	200	250	300	350
		Unit Friction Head, in Milinches								
		8.0	6.0	4.8	4.0	3.4	3.0	2.4	2.0	1.7
1½	3.0	*43.0*	*37.5*	*33.0*	*30.0*	*27.0*	*25.0*	*22.2*	*20.2*	*18.7*
2	4.0	*83.0*	*72.0*	*63.0*	*57.0*	*51.0*	*48.0*	*42.0*	*38.0*	*35.0*
2½	4.5	*140.0*	*115.0*	*100.0*	*90.0*	*81.5*	*75.4*	*67.2*	*61.0*	*56.0*
3	5.0	*234.0*	*204.0*	*175.5*	*160.0*	*143.0*	*133.0*	*110.0*	*107.5*	*100.0*
3½	5.5	*347.0*	*300.0*	*260.0*	*236.0*	*214.0*	*200.0*	*177.0*	*160.0*	*146.0*
4	6.0	*490.0*	*422.0*	*370.0*	*334.0*	*297.0*	*278.0*	*248.0*	*223.0*	*205.0*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity

Example 4. Design a two-pipe, direct return, gravity circulation system for the layout shown in Fig 8 Assume that the main circuit from the boiler to the farthest flow riser and from the farthest return riser back to the boiler consists of 160 ft of pipe, 6 elbows, and 1 boiler.

Solution. Replacing the boiler by 3 elbow equivalents and assuming that the largest size of the main will be about 3 in , the total equivalent length of the main will be 160 plus 45, or 205 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main, and that the temperature drop will be 35 F for the system, the pressure head caused by the difference in weight between the water in the flow and return risers joining the mains to the boiler will be about 0.6 in of water.

Table 7 may be used to determine the size of the main as follows: Refer to Column 8 and note that for Sections *AB* and *IA*, which supply 105.6 Mbh, a 3 in pipe is too large and a 2½ in. pipe is too small, hence, select 2½ in. rather than 3 in as noted in Fig. 8 for Section *AB* and 3 in for Section *IA* For Sections *BC* and *HI*, which supply 76.8 Mbh, a 2½ in. pipe is almost exactly the correct size and is selected for both sections.

Tables 7 and 8 are based on the assumption that the boiler pressure head must be equal to the friction head in the mains, and that the several radiator pressure heads must be equal to the respective radiator and riser friction heads

To design the radiator risers, use Table 8 and begin with the set nearest the boiler. The first floor risers must supply 28.8 Mbh According to the table, 1¼ in. flow and return risers will supply 26.0 Mbh, if the return riser is increased to 1½ in., the capacity will be increased to 34.0 Mbh. This is considerably larger than necessary, and 1¼ in. flow and return risers are selected. However, it must be remembered that the riser branches, which are the connections from the flow and return mains to the flow and return risers, are to be one size larger than the risers.

The second floor risers must supply 19.2 Mbh. According to the table, the capacity of 1 in. flow and return risers is 20.0 Mbh, and that size is selected.

The third floor risers must supply 9.6 Mbh. If a ½ in. flow and a ¾ in. return riser is used, the capacity will be 8.0 Mbh, if both risers are ¾ in., the capacity will be 14.0 Mbh. The ¾ in. pipe is selected for both risers.

To design the radiator connections, use Table 9 and note that for the first floor radiator connections the capacity of a ¾ in. flow and 1 in. return is 9.1 Mbh, and that of

TABLE 8. MAXIMUM CAPACITIES OF RISERS[a] IN *Mbh*, AND **Velocities of Water in Pipes in Inches Per Second** FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A DROP OF 35 F THROUGH EACH RADIATOR

PIPE SIZE (INCHES)		EQUIVALENT LENGTH OF PIPE (FEET[c])	1ST FLOOR[b]			2ND FLOOR	3RD AND 4TH FLOORS
				Vel (In per Sec[d])			
Flow	Return		*Mbh*	Flow	Return	*Mbh*	*Mbh*
½	½	1.0				*5*	*6.2*
½	¾					*6.4*	*8.0*
¾	¾	1.5	*9*	2.3	2.3	*10.1*	*14.0*
¾	1		*12*	3.2	2.0	*12.8*	*17.1*
1	1	2.0	*18*	2.5	2.5	*20*	*26.0*
1	1¼		*21*	3.0	2.0	*25.2*	*34*
1¼	1¼	3.0	*26*	3.0	3.0	*43*	*55*
1¼	1½		*34*	4.0	2.5		
1½	1½	3.5	*48*	3.0	3.0		

[a]This table is based on pressure heads of 450, 1800, 3150, and 4500, respectively, for the first, second, third, and fourth floor radiators, and on friction heads of 200 milinches for the first floor radiators and connections, and 700 milinches for all other radiators and their connections

[b]The riser branches, the piping which connects the risers to the mains, are to be one size larger than the risers

[c]Approximate length of pipes in feet equivalent to one elbow in friction head This value varies with the velocity

[d]Velocities apply to the riser branches

a 1 in. flow and a 1 in. return is 12.5 Mbh. The former is more nearly the correct size, but since it is difficult to secure a good flow through first floor radiators, the 1 in. flow and return connection is selected. For the two upper floors, the capacity of a ¾ in flow and return connection is 10.5 Mbh, and that size is used.

As explained in the design of the forced circulation system of Fig. 5, the two-pipe direct return system of Fig. 8 will not function correctly unless its four sets of risers are balanced among themselves. This necessary balancing is accomplished by adding resistances to all risers, except the one farthest from the boiler, equal to the excess boiler pressure heads available for those risers above the boiler pressure head available for the farthest riser. For example, the first set of risers is 60 ft nearer the boiler than the last set. Since the flow and return mains are designed for a friction head of 3 milinches per foot (see Table 7, Column 8), the boiler pressure head available for the first set of risers is 360 milinches in excess

TABLE 9. MAXIMUM CAPACITIES OF RADIATOR CONNECTIONS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TEMPERATURE DROP OF 35 F THROUGH EACH RADIATOR

PIPE SIZE		EQUIVALENT LENGTH OF PIPE (FEET[a])	1ST FLOOR	2ND, 3RD, AND 4TH FLOORS
Flow	Return		*Mbh*	*Mbh*
½	½	1.0	*4.1*	*5.9*
½	¾		*5.2*	*7.5*
¾	¾	1.5	*7.0*	*10.5*
¾	1		*9.1*	*13.0*
1	1	2.0	*12.5*	*17.8*
1	1¼		*17.5*	*23.2*
1¼	1¼	3.0	*23.3*	*33.2*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

of that available for the fourth set. The velocity in the riser branch is 3 in. per second (see Table 8) and, therefore, according to Table 6, an 0.65 in. orifice in a 1¼ in union should be used. This will provide a resistance of about 420 milinches. In the same manner it is found that for the second set of risers a resistance of 240 milinches is required and that an 0.70 in. orifice in a 1¼ in. union will provide a resistance of 285 milinches. For the third set of risers, a resistance of 120 milinches is required and an 0.60 in. orifice in a 1 in. union will provide sufficient resistance.

EXPANSION TANKS

When water at ordinary temperatures is heated or cooled, its volume is increased or decreased. This variation in the volume of the water in a heating system is generally provided for by means of an expansion tank

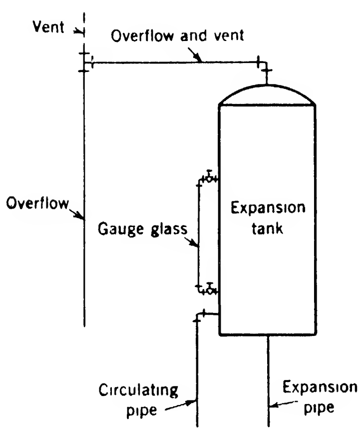


FIG. 9. AN OPEN EXPANSION TANK

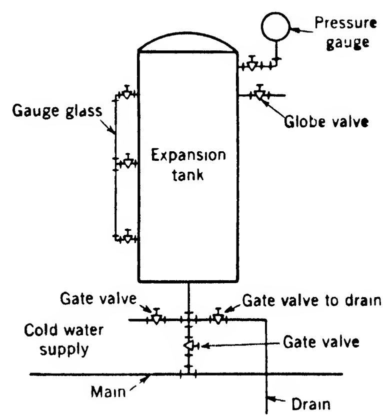


FIG. 10. A CLOSED EXPANSION TANK

into which the water can flow from the system during the heating-up periods and from which it can flow back into the system during the cooling-down periods.

The expansion tank may be open or closed. In an open expansion tank (Fig. 9), the water is subjected to atmospheric pressure and can expand freely without a material increase in pressure. In a closed expansion tank (Fig. 10), the water is subjected to the pressure of the compressed air within the tank, and as the water expands, the volume of the air in the tank is decreased and its pressure increased.

The open expansion tank must be placed at a sufficient elevation above the highest radiator to prevent boiling when the water in that radiator is at the highest temperature to which it is to be heated. For example, if the water is to be heated to 225 F on extremely cold days, the absolute pressure on the water in the highest radiator must be at least 19 lb per square inch. This pressure will be secured if the open expansion tank is located 15 ft above the highest radiator. If a closed expansion tank is used and is located 30 ft below the highest radiator, an absolute pressure

of about 32 lb per square inch must be maintained in the expansion tank if the water in the highest radiator is to be heated to 225 F without danger of boiling.

The type of expansion tank used in a heating system, whether open or closed, has no influence on the operation of the system. The only function performed by the expansion tank is to provide for the variation in the volume of the water in the system, and at the same time to maintain a sufficient pressure in the system to prevent boiling when the water is at the highest temperature for which the system is designed. The capacity of the cushion or expansion tank should not be less than the tank sizes indicated in Table 10 and in addition provisions must be made for draining it without emptying the system.

The capacity of the expansion tank should be at least twice the increase in volume produced when the water in the system is heated from its normal to its maximum temperature. When 25 gal of water are heated

TABLE 10. EXPANSION TANK SIZES FOR HOT WATER HEATING SYSTEMS

Tank Size Gallons	Equivalent Direct Radiation Installed in Sq Ft	Capacity Direct Radiation Installed in Mbh
18	Up to 350	Up to 52 5
21	Up to 450	Up to 67 5
24	Up to 650	Up to 97 5
30	Up to 900	Up to 135 0
35	Up to 1100	Up to 165 0
40	Up to 1400	Up to 210 0
2–30	Up to 1600	Up to 240 0
2–30	Up to 1800	Up to 270 0
2–35	Up to 2000	Up to 300 0
2–40	Up to 2400	Up to 360 0

from 40 F to 200 F, the volume of water increases to 26 gal. A safe rule, therefore, is to make the water capacity of the expansion tank equal to 10 per cent of the capacity of the heating system.

In a forced circulation system, the expansion tank can either be connected to the flow or return main. In a gravity circulation system, the expansion tank should be connected to the flow riser so that air liberated from the water in the boiler may escape through the expansion tank.

The expansion tank should be protected so that the water in the tank or in the connecting pipe lines cannot freeze. If the water should freeze and the water in the system is heated causing further expansion, the resulting force will burst the boiler or some other portion of the system.

RELIEF VALVES

A relief valve should be installed on any hot water system using a closed circuit. The valve should be of ample capacity to provide for relief of expansion of the system without allowing an excessive pressure rise above the valve setting.

A relief valve should be of the diaphragm-operated or gravity-weighted type without guide wings below the seat. Provision should be made for manual operation to assure that the valve is in the proper operating condition at all times, and valves should be checked periodically.

A relief valve installed in conjunction with a compression tank will not operate often provided the tank is of adequate size. It is essential that the relief valve be kept in good condition to eliminate any possible failure when operation is necessary.

INSTALLATION DETAILS

The detailed installation of the pipe system should be governed by four fundamental rules:

1. All piping must be pitched either up or down so that all gases which are liberated from the water can move freely to a vented section of the system Whenever practicable, the pipe line should be pitched so that gases flowing to a vent will flow in the same direction as the water. When a pipe system cannot be installed without creating *air pockets*, that is, sections in the system from which liberated gases cannot escape, such sections must be provided with automatic air relief valves or with air valves which may be operated manually when necessary, or trapped into a pressure tank

2. All piping must be arranged so that the entire system can be drained, either to permit alterations or repairs, or to prevent freezing if the system is not to be operated during a cold period

It is well to install a gate valve and union in every riser near the main to permit the draining of individual risers without draining the entire system It is also well, in large installations, to divide the system into branches and to provide each branch with unions and valves so that any one branch can be drained without disturbing the remaining ones

The dividing of large heating systems into branches or zones and providing each zone with individual valves has the further advantage of permitting a varying temperature control For example if a building is equipped with a forced circulating system and if the south rooms are on one branch of the main and the north rooms are on a separate branch, the valves may be set so that the water will circulate through the north branch with a temperature drop of, say, 10 F, and through the south branch with a temperature drop of, say, 20 F, thus delivering less heat to the south rooms than to the north rooms. This arrangement is especially valuable when the regulating valves are controlled thermostatically by the temperatures in the two zones, because no matter how accurately the heating system may have been designed, the heat demand of any group of rooms varies with sunshine and with wind velocity, and these intermittent variations can be provided for only by the individual control made possible by changing the valve settings controlling the heat supplied to particular groups of rooms.

3. All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections For this purpose it is generally sufficient to allow for a variation in length of 1 in for 100 ft of pipe.

4. The pipe system must be installed so that each circuit has its correct friction head. To bring this about, it is necessary in some cases to minimize the friction, *i e.*, to make the pipe line as short as possible and to provide as few fittings as possible; and in other cases it is necessary to increase the length of the pipe and the number of fittings so that, for every circuit the friction head will be equal to the available pressure head.

The connections from the boiler to the mains should be short and direct, to reduce the friction head. It is frequently possible to avoid an elbow and to reduce the length of the pipe by running the pipe in a diagonal direction, either in a horizontal or in a vertical plane.

The mains and branches should pitch up and away from the heater, generally not less than 1 in in 10 ft. The flow main should always be covered, the return main should be covered except where it is to provide the heating surface for the basement.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow

connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater

In some cases it may be advisable to take the flow connection off the top of the main and the return connection from the side, but in most cases both connections should be at an angle of 45 deg. This method shortens the lines and substitutes 45-deg ells for 90-deg ells

Preferably, connection of the flow riser to a radiator should be to the upper tapping, and connection of the return riser to a radiator should be to the lower tapping When hot water enters at the top of a radiator it will distribute itself along the entire length of the radiator, and as it cools it will settle gradually to the bottom, the cool water may then be taken out of the radiator at either end.

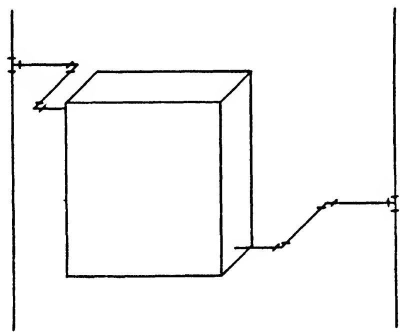


FIG. 11. METHOD OF CONNECTING RADIATOR TO ALLOW FOR EXPANSION OF PIPE

With forced circulation and high velocities, it is advisable to let the water enter at the top of the radiator and leave at the bottom of the opposite end With gravity circulation and low velocities it makes little difference whether the water leaves at the end at which it enters or at the opposite end.

The connections of the risers to the radiators should be such that provision is made for the vertical expansion of the risers. This can be accomplished as indicated in Fig 11 by using one tee and two ells for each connection. These connections should be pitched upward or downward, whichever may be necessary to prevent the formation of air pockets and to permit draining

PROBLEMS IN PRACTICE

1 ● Will altering a hot water heating system from an open to closed type system (*a*) increase the circulation and (*b*) give more heat?

a. No Tests conducted by the A.S.H.V.E. indicate that there is little, if any difference in the circulation when the system is under pressure. The difference in temperature between the supply and return, and the friction are the governing factors.

b. With a closed system the water may be carried at a higher temperature without boiling which permits warmer radiators.

2 ● What tends to prevent or to retard the circulation of water in hot water heating systems?

In both gravity flow and forced circulation systems, the friction which must be overcome when the water is flowing through pipes, fittings, valves, heaters, and radiators tends to

prevent or retard circulation. For a given pipe the friction varies approximately as the 1.7 power of the velocity, and for given fittings, valves, heaters, and radiators, the friction varies approximately as the square of the velocity. It is therefore sufficiently accurate to express the friction in fittings, valves, heaters, and radiators in terms of the friction in one standard elbow, as shown in Table 3.

3 ● If a single radiator located 10 ft above a boiler is connected with a flow and return black iron pipe, what is the pressure head maintaining the circulation if the water in the return riser is at 180 F and that in the flow riser is at 200 F?

It is found, from Table 7, Chapter 1, that 180 F water weighs 60.61 lb per cubic foot and 200 F water weighs 60.13 lb per cubic foot. The pressure head is independent of the size of the pipe. If the two risers were each 1 ft square, the water in the flow riser would weigh 601.3 lb and that in the return riser would weigh 606.1 lb. Thus the water in the return riser would weigh 4 8 lb more than that in the flow riser. Consequently, the resulting pressure head is 4.8 lb per square foot.

Pressure heads are generally expressed in feet, or inches, or milinches of water of a given temperature. In this case water is at both 180 F and 200 F, so the pressure head is expressed in terms of 190 F water. Such water weighs 60 39 lb per cubic foot, and to secure a pressure of 4 8 lb per square foot, it is necessary to have a column of water having a weight of 4.8 divided by 60.39 = 0 0795 ft, or 0.9540 in., or 954 milinches This is the pressure head which maintains the circulation.

4 ● In the elementary system of Question 3, if the radiator dissipates 14,000 Btu per hour, what is the velocity of the water in the pipe line, if the pipes are 1 in. in diameter? What, if they are ¾ in. in diameter?

Since the temperature drop through the radiator is from 200 F to 180 F or 20 F, every pound of water flowing through the radiators delivers 20 Btu, consequently, 14,000 divided by 20 = 700 lb of water, or for 190 F water, 700 divided by 60.39 = 11.59 cu ft of water must flow through the radiator and through the pipe lines every hour.

The interior area of a 1 in. pipe is 0 864 sq in. The velocity in the 1 in pipe is 11.59 divided by 0 864 and multiplied by 144 = 1932 ft per hour or 6.44 in per second.

For ¾ in pipe, the interior area is 0.533, and the velocity is 6.44 multiplied by 0 864 and divided by 533 = 10.44 in. per second.

5 ● If, in the elementary heating system of Question 3, a 1 in. pipe line is used, what would be the friction head?

If the radiator is connected with the heater to provide for freedom of expansion, the heating circuit may be assumed to consist of a heater, 25 ft of pipe, 8 elbows, 1 radiator valve, and 1 radiator From Table 3 it appears that the heater and radiator are equivalent, in friction, to 6 elbows, hence, the circuit may be placed equal to 25 ft of pipe and 14 elbows.

From the diagram of Fig. 4 it appears that the friction head for a 1 in. pipe and a velocity of 6.44 in per second is about 25 milinches per foot. For 25 ft of pipe, the friction head will be 625 milinches.

It appears from Table 3 that the friction head in one elbow is $\frac{v^2}{2g}$, or in this case 0 54 multiplied by 0.54 and divided by 64.4 = 0.0045 ft or 54 milinches. Hence, for the 14 elbows the friction is 756 milinches. For the entire circuit, the friction head is the sum of the 625 milinches of the pipe plus the 756 milinches of the elbows, or 1381 milinches which equal 1.381 in.

6 ● If the elementary heating system of Question 3 is installed with a 1 in. pipe line, how will it function?

It is found from the answer to Question 3 that the pressure head is 954 milinches and from the answer to Question 5 that the friction head is 1381 milinches when the water is flowing with such velocity that the specified 14,000 Btu will be delivered with a 20 F temperature drop through the radiators. Since the pressure head is smaller than the friction head, the system will not function as planned for the water will flow through the

system more slowly and remain in the radiator longer. The temperature drop through the radiator will be more than 20 F, and the difference in the weight of the water in the return and flow risers will be greater than that intended. The final result will be that the pressure head will become equal to the friction head at a value somewhere between 954 and 1381 milinches. Since the average water temperature in the radiator will be less than 190 F, the radiator should be larger than the size given in Question 4.

7 ● Should a hot water heating system be designed to embody small pipes or large pipes?

As pipe sizes in gravity circulation heating are reduced, the friction head is increased and it is necessary to increase the temperature drop through radiators, this lowers the average temperature of the water in the radiators and necessitates an increase in the size of the radiators, so whereas the cost of the pipe in a system is reduced, the cost of the radiators is increased For each installation there is a definite pipe size which entails maximum economy.

As pipe sizes in forced circulation systems are reduced, friction heads are increased so a circulating pump of greater size or capacity is required. Thus, by decreasing the size of the piping, both the first cost of the circulating pump and the cost of its operation are increased There is a definite pipe size for every installation which is most economical. For each installation of both types of systems there is a definite pipe size entailing maximum economy which can be determined by a series of comparative calculations.

8 ● What should be the size of the radiators for the elementary heating system of Question 3 in which the water enters the radiator with a temperature of 200 F and leaves with a temperature of 180 F? The average temperature of the water in the radiator is, approximately, 190 F.

If test results are available for the particular radiators to be used, and for the temperatures named, the size of the radiators should be selected from them. If no such test results are to be had, but if test results are available for the type of radiator to be used when it is supplied with 215 F steam and placed in a 70 F room, the required size may be determined by the following ratio The required size is to the corresponding steam radiator size as $(215 - 70)^{1.3}$ is to $(190 - 70)^{1.3}$. This ratio works out to 1.28. Hence, the radiators should be 28 per cent larger under the conditions prescribed than are corresponding radiators under standard conditions. It is immaterial whether a radiator is filled with steam or with water, as long as the average temperature of its outer surface is the same in both cases.

Chapter 18

PIPE, FITTINGS, WELDING

Pipe Material, Types of Pipe Used, Dimensions of Pipe Commercially Available, Expansion and Flexibility of Pipe, Pipe Threads and Hangers, Types of Fittings, Welding as Applied to Erection of Piping, Valves, Corrosion of Piping

IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning work are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought-iron, copper and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating permits the use of thin-wall pipe and thereby has reduced the initial cost of such installation. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe,* the latter as *seamless pipe.*

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld or butt-weld process. While the lap-weld process produces a better weld than the butt type, lap-weld pipe is seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operation. Its advantages are its somewhat greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe is considered to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher

first cost can be justified on the basis of longer life expectancy. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color. Otherwise, there is little difference in the appearance of wrought-iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose the difference.

Cast Ferrous Pipe. There are now available several types of cast ferrous-metal pipe made of a good grade of cast iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft with external and internal diameters closely approximating those of extra-strong wrought pipe. Cast ferrous pipe may be obtained coupled, beveled for welding, or with ends plain or grooved for the several types of couplings. It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper or other alloying element and iron pipe bearing a small alloy of copper and molybdenum have been claimed to possess more resistance to corrosion than plain steel pipe and they are advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to its inherent resistance to corrosion, copper and brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The recent introduction of fittings which permit erection by soldering or sweating allows the use of pipe with thinner walls than are possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin-wall pipe, but the corrosive nature of the fluid conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

COMMERCIAL PIPE DIMENSIONS

The *IPS* dimensions of commercial pipe universally used at the present time conform to the recommendations made by a Committee of the *A.S.M.E.* in 1886. Pipe up to 12 in. in diameter is made in certain definite sizes designated by nominal internal diameter which is somewhat different from the actual internal diameter, depending on the wall thickness required. There are three weights of wrought-iron and steel pipe commonly used, known as *standard-weight*, *extra-strong*, and *double extra-strong*. Because of the necessity of maintaining the same external diameter in all three weights for the same nominal size, the added wall thickness is obtained by decreasing the internal diameter. The term *full-weight*, when applied to sizes below 8 in., means that the pipe is up to the nominal weight per foot. When applied to sizes between 8 and 12 in., inclusive, it often indicates that the pipe has the heaviest of several wall

thicknesses listed. In sizes 14 in. and upward, pipe is designated by its outside diameter (O.D.) and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry are reasonably well served by the *standard-weight* and *extra-strong* pipe, demands for pipe for higher pressures and temperatures in industry resulted in the use of a multiplicity of wall thicknesses for all sizes. Even in heating installations, the erection of piping by welding was deemed to

TABLE 1. DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE

Nominal Pipe Size	Outside Diam	Nominal Wall Thicknesses for Schedule Numbers									
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80	Schedule 100	Schedule 120	Schedule 140	Schedule 160
1/8	0.405				0.068*		0.095*				
1/4	0.540				0 088*		0.119*				
3/8	0.675				0 091*		0.126*				
1/2	0.840				0 109*		0 147*				0.187
3/4	1.050				0 113*		0 154*				0.218
1	1.315				0 133*		0.179*				0 250
1 1/4	1.660				0 140*		0 191*				0.250
1 1/2	1.900				0 145*		0 200*				0.281
2	2.375				0.154*		0 218*				0.343
2 1/2	2.875				0 203*		0 276*				0 375
3	3.500				0 216*		0 300*				0 437
3 1/2	4.000				0 226*		0.318*				
4	4.500				0 237*		0.337*		0 437		0 531
5	5.563				0 258*		0.375*		0.500		0 625
6	6.625				0 280*		0.432*		0.562		0 718
8	8.625		0 250	0.277*	0.322*	0.406	0.500*	0.593	0.718	0 812	0 906
10	10.75		0 250	0.307*	0.365*	0.500*	0.593	0.718	0 843	1 000	1 125
12	12.75		0 250	0.330*	0 406	0 562	0.687	0 843	1.000	1 125	1.312
14 O D.	14 0	0 250	0.312	0.375	0.437	0 593	0 750	0.937	1 062	1 250	1.406
16 O. D.	16.0	0 250	0.312	0 375	0 500	0 656	0 843	1.031	1 218	1.437	1.562
18 O. D.	18.0	0 250	0.312	0 437	0.562	0 718	0 937	1 156	1 343	1 562	1.750
20 O. D.	20.0	0 250	0.375	0.500	0.593	0 812	1 031	1 250	1 500	1.750	1 937
24 O D	24.0	0 250	0.375	0 562	0.687	0.937	1 218	1 500	1 750	2.062	2.312
30 O D.	30.0	0 312	0.500	0 625							

All dimensions are given in inches

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12 5 per cent under nominal thicknesses

*Thicknesses marked with asterisk in Schedules 30 and 40 are identical with thicknesses for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for *extra-strong* pipe in former lists

The Schedule Numbers indicate approximate values of the expression 1000 x P/S

warrant the use of pipe lighter than standard weight. For these reasons, a *Sectional Committee on Standardization of Wrought Iron and Wrought Steel Pipe and Tubing* functioning under the procedure of the *American Standards Association* was appointed to standardize the dimensions and materials of pipe.

The proposed pipe standard recommended by that sectional committee has set up several schedules of pipe including standard-weight and extra-strong thicknesses which are now included in Schedules 40 and 60, respectively. The schedules approved by the Sectional Committee are given in Tables 1 and 3 and the corresponding weights in Tables 2 and 4.

Standard-weight pipe is generally furnished with threaded ends in

random lengths of 16 to 22 ft, although when ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Five per cent of the total number of lengths ordered may be *jointers* which are two pieces coupled together. Extra-strong pipe is generally furnished with plain ends in random lengths of 12 to 22 ft, although 5 per cent may be in lengths of 6 to 12 ft.

In addition to *IPS* copper pipe, several varieties of copper tubing are in use with either flared or compression couplings or soldered joints. Dimensions of copper water tubing intended for plumbing, underground water service, fuel-oil lines, gas lines, etc., have been standardized by the U. S. Government and the *American Society for Testing Materials*. There are three standard wall-thickness schedules of copper water tubing classified in accordance with their principal uses as follows:

Class *K*—Designed for underground services and general plumbing service.

Class *L*—Designed for general plumbing purposes.

Class *M*—Designed for use with soldered fittings only.

In general, Type *K* is used where corrosion conditions are severe, and

TABLE 2. NOMINAL WEIGHTS OF WELDED AND SEAMLESS STEEL PIPE

Nominal Pipe Size Inches	Sched 10 Plain Ends	Sched 20 Plain Ends	Schedule 30		Schedule 40		Sched 60 Plain Ends	Sched 80 Plain Ends	Sched 100 Plain Ends	Sched 120 Plain Ends	Sched 140 Plain Ends	Sched 160 Plain Ends
			Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings						
1/8	...	...	...	...	0 25*	0.25*	...	0.32*	...	...	...	...
1/4	...	...	...	...	0.43*	0.43*	...	0 54*	...	...	...	...
3/8	...	...	...	...	0.57*	0 57*	...	0.74*	...	...	...	...
1/2	...	...	...	...	0 86*	0.86*	...	1.09*	...	...	...	1.31
3/4	...	...	...	...	1 14*	1 14*	...	1.48*	...	...	...	1 94
1	...	...	...	...	1.68*	1.69*	...	2.18*	...	...	...	2 85
1¼	...	...	...	...	2.28*	2.29*	...	3 00*	...	...	...	3 77
1½	...	...	...	...	2.72*	2 74*	...	3 64*	...	...	...	4 86
2	...	...	...	...	3.66*	3.68*	...	5.03*	...	...	...	7 45
2½	...	...	...	...	5.80*	5.82*	...	7.67*	...	...	...	10 0
3	...	...	...	...	7.58*	7.62*	...	10 3*	...	...	...	14 3
3½	...	...	...	...	9.11*	9.21*	...	12 5*	...	...	...	...
4	...	...	...	...	10.8*	10 9*	...	15 0*	...	19 0	...	22 6
5	...	...	...	...	14.7*	14.9*	...	20.8*	...	27 1	...	33.0
6	...	...	...	...	19.0*	19.2*	...	28.6*	...	36.4	...	45 3
8	...	22.4	24.7*	25.0*	28.6*	28.8*	35.7	43.4*	50 9	60.7	67.8	74 7
10	...	28.1	34.3*	35.0*	40.5*	41.2*	54.8*	64 4	77.0	89.2	105 0	116 0
12	...	33.4	43.8*	45.0*	53.6	55.0	73.2	88 6	108.0	126.0	140 0	161.0
14 O. D.	36.8	45.7	54.6	...	63.3	...	85 0	107 0	131.0	147 0	171.0	190.0
16 O. D.	42.1	52.3	62.6	...	82.8	...	108.0	137.0	165.0	193.0	224.0	241.0
18 O. D.	47.4	59.0	82.0	...	105.0	...	133 0	171 0	208.0	239.0	275 0	304 0
20 O. D.	52.8	78.6	105.0	...	123.0	...	167 0	209 0	251.0	297.0	342 0	374 0
24 O. D.	63.5	94.7	141.0	...	171.0	...	231.0	297.0	361.0	416.0	484.0	536.0
30 O. D.	99.0	158.0	197.0	...	...	...	...	...	...	...	...	...

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed

*The weights marked with asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

Types *L* and *M* where such conditions may be considered normal as, for instance, in heating work. Types *K* and *L* are available in both hard and soft tempers; Type *M* is available only in hard temper. Where flexibility is essential as in hidden replacement work or where as few joints as possible are desired as in fuel-oil lines, the soft temper is commonly used. New or exposed work generally employs copper pipe of a hard temper. All three classes are extensively used with soldered fittings.

Standard dimensions, weights, and diameter and wall thickness tolerances for these classes of copper tubing are given in Table 5. Copper pipe is also available with dimensions of steel pipe.

Refrigeration lines used in connection with air conditioning equipment also employ copper tubing extensively. For refrigeration use where tubing absolutely free from scale and dirt is required, bright annealed copper tubing that has been deoxidized is used. This tubing is available in a variety of sizes and wall thicknesses.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature 100 F or more above room tem-

TABLE 3. DIMENSIONS OF WELDED WROUGHT-IRON PIPE

Nominal Pipe Size	Outside Diameter	Nominal Wall Thicknesses for Schedule Numbers					
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80
1/8	0 405	...	...	...	0.070*	...	0 098*
1/4	0 540	...	...	...	0.090*	...	0.122*
3/8	0 675	...	...	...	0 093*	...	0 129*
1/2	0 840	...	...	...	0 111*	...	0 151*
3/4	1 050	...	...	...	0.115*	...	0 157*
1	1 315	...	...	...	0 136*	...	0 183*
1 1/4	1.660	...	...	...	0.143*	...	0 195*
1 1/2	1 900	...	...	...	0.148*	...	0 204*
2	2.375	...	...	...	0 158*	...	0 223*
2 1/2	2 875	...	...	...	0 208*	...	0 282*
3	3 5	...	...	...	0.221*	...	0 306*
3 1/2	4 0	...	...	...	0 231*	...	0 325*
4	4.5	...	...	...	0.242*	...	0 344*
5	5 563	...	...	...	0.263*	...	0 383*
6	6 625	...	...	...	0 286*	...	0 441*
8	8 625	...	...	0.283*	0 329*	...	0 510*
10	10 75	...	...	0.313*	0.372*	0 510*	0 606
12	12 75	...	...	0.336*	0 414	0.574	0 702
14 O D.	14 0	0.250	0 312	0.375	0 437	0.625	0 750
16 O. D.	16 0	0.250	0.312	0.375	0.500	0.687	...
18 O D	18.0	0 250	0.312	0.437	0.562	0.750	...
20 O. D	20 0	...	0.375	0 500	0 562	...	...

All dimensions are given in inches

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12 5 per cent under the nominal thickness

*Thicknesses marked with an asterisk in Schedules 30 and 40 are identical with thicknesses for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for *extra-strong* pipe in former lists

The Schedule Numbers indicate approximate values of the expression 1000 x P/S

perature results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the case of refrigeration lines) per unit length of material per degree change in temperature is termed the *coefficient of linear expansion* of that material, or commonly, the *coefficient of expansion*. This coefficient varies with the material.

The linear expansion of cast iron, steel, wrought iron, and copper pipe,

TABLE 4. NOMINAL WEIGHTS OF WELDED WROUGHT-IRON PIPE

Nominal Pipe Size (Inches)	Sched 10	Sched 20	Schedule 30		Schedule 40		Schedule 60	Schedule 80
	Plain Ends	Plain Ends	Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings	Plain Ends	Plain Ends
1/8	....	......	....	....	0.25*	0.25*	....	0.32*
1/4	....	.. . .	... .	....	0.43*	0.43*	....	0 54*
3/8	....	......	....	....	0.57*	0.57*	....	0 74*
1/2	....	......	....	....	0.86*	0.86*	....	1 09*
3/4	....	......	....	....	1.14*	1.14*	....	1 48*
1	....	......	....	....	1.68*	1.69*	....	2.18*
1 1/4	....	......	....	....	2.28*	2.29*	....	3 00*
1 1/2	....	......	....	....	2.72*	2.74*	.. ..	3 64*
2	....	......	....	....	3.66*	3.68*	....	5 03*
2 1/2	....	......	....	....	5.80*	5.82*	....	7.67*
3	..... .	......	....	....	7.58*	7.62*	....	10.3*
3 1/2	......	......	....	....	9 11*	9.21*	....	12.5*
4	....	......	....	....	10.8*	10.9*	....	15.0*
5	......	......	....	....	14.7*	14.9*	....	20.8*
6	......	......	....	....	19.0*	19.2*	....	28.6*
8	....	....	24.7*	25 0*	28.6*	28.8*	....	43.4*
10	....	....	34.3*	35 0*	40.5*	41.2*	54 8*	54.4
12	....	....	43.8*	45.0*	53.6	55.0	73.2	88.6
14 O D.	36 0	44.8	53.6	....	62.2	....	87.6	104.0
16 O. D.	41.3	51.4	61.4	....	81 2	....	111.0	....
18 O D.	46 5	57.9	80.5	....	103 0	....	136.0	....
20 O D.	....	77.0	103.0	....	115.0	....	....	....

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed

*Weights marked with an asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe in former lists, those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists

The Schedule Numbers indicate approximate values of the expression 1000 x P/S

the materials most frequently used in heating and ventilating work, can be determined from Table 6.

The elongation values in Table 6 were computed from the following formula:

$$L_t = L_o \left[1 + a\left(\frac{t-32}{1000}\right) + b\left(\frac{t-32}{1000}\right)^2\right] \quad (1)$$

where

L_t = length at temperature t degrees Fahrenheit, feet.

L_o = length at 32 F, feet.

t = final temperature, degrees Fahrenheit

a and b are constants as given on the next page.

Metal	a	b
Cast-Iron	0.005441	0.001747
Steel	0.006212	0.001623
Wrought-Iron	0.006503	0.001622
Copper	0.009278	0.001244

The three methods by which the elongation due to thermal expansion may be taken care of are:

1. Expansion joints.
2. Swivel joints.
3. Inherent flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line

TABLE 5. STANDARD DIMENSIONS, WEIGHTS, AND DIAMETER AND WALL THICKNESS TOLERANCES FOR COPPER WATER TUBES*

(All Tolerances Plus and Minus)

Nominal Size, In	Actual Outside Diameter, In	Permissible Variation in Mean Outside Diameter, In		Wall Thickness, In						Weight per Ft Lb		
				Class K		Class L		Class M				
		Annealed	Hard Drawn	Nominal	Permissible Variation	Nominal	Permissible Variation	Nominal	Permissible Variation	Class K	Class L	Class M
3/8	0.500	0.0025	0.001	0.049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0.144
1/2	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.203
3/4	0.875	0.003	0.001	0.065	0.0045	0.045	0.004	0.032	0.003	0.641	0.455	0.328
1	1.125	0.0035	0.0015	0.065	0.0045	0.050	0.004	0.035	0.0035	0.839	0.655	0.464
1 1/4	1.375	0.004	0.0015	0.065	0.0045	0.055	0.0045	0.042	0.0035	1.04	0.884	0.681
1 1/2	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.94
2	2.125	0.005	0.002	0.083	0.005	0.070	0.005	0.058	0.0045	2.06	1.75	1.46
2 1/2	2.625	0.005	0.002	0.095	0.005	0.080	0.005	0.065	0.0045	2.92	2.48	2.03
3	3.125	0.005	0.002	0.109	0.005	0.090	0.005	0.072	0.0045	4.00	3.33	2.68
3 1/2	3.625	0.005	0.002	0.120	0.005	0.100	0.005	0.083	0.005	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0.006	0.110	0.005	0.095	0.005	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.006	0.125	0.006	0.109	0.005	9.67	7.61	6.65
6	6.125	0.005	0.002	0.192	0.006	0.140	0.006	0.122	0.005	13.87	10.20	8.91

*From Standard Specifications for Copper Water Tube of the *American Society for Testing Materials*, *A S T M* Designation B88-33

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used extensively in low-pressure steam and hot water heating systems and in hot water supply lines. The swivel joints absorb the expansive movement of the pipe by the turning of threaded joints. In many cases the straight pipe in the offset of a swivel joint is sufficiently flexible to take up the expansion without developing enough thrust to produce swiveling in the threaded joint. This is preferable since con-

tinued turning in the threaded joint may in time result in a leak, particularly when the pressure is high. The amount of elongation which a swivel joint can take up is controlled by the length of the swing piece employed and by the lateral displacement which is permissible in the long pipe runs.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion

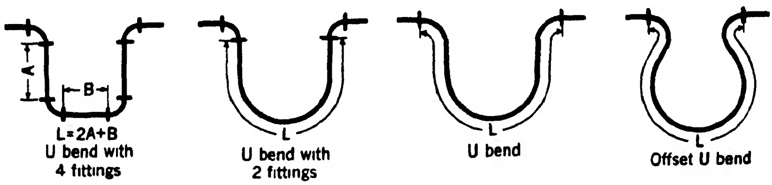


FIG. 1. MEASUREMENT OF L ON VARIOUS PIPE BENDS

is relatively complicated[1]. The following approximate method, however, has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating work.

Fig. 1 shows several types of expansion bends commonly used for taking up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from the following formula:

$$L = 6.16 \sqrt{D \Delta} \tag{2}$$

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber stress of 16,000 lb per square inch. When square type bends are used, the width of the bend should not exceed about two times the height. It is further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter.

All risers must be anchored and safeguarded so that the difference in

[1]Piping Handbook, by Walker and Crocker, and A Manual for the Design of Piping for Flexibility by the Use of Graphs, by E. A. Wert, S Smith, and E. T. Cope, published by The Detroit Edison Company

length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

It is especially necessary with light-weight radiators so to anchor the piping and so to give it freedom for expansion that no strain therefrom shall be allowed to distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters they usually emit sounds of distress which are exceedingly troublesome.

TABLE 6 THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]

(For superheated steam and other fluids refer to temperature column)

SATURATED STEAM			ELONGATION IN INCHES PER 100 FT FROM −20 F UP				SATURATED STEAM		ELONGATION IN INCHES PER 100 FT FROM −20 F UP			
Vacuum Inches of Hg.	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe
		−20	0	0	0	0	664.3	500	3.847	4.296	4.477	6.110
		0	0 127	0 145	0.152	0.204	795.3	520	4.020	4.487	4 677	6 352
		20	0.255	0.293	0.306	0.442	945.3	540	4.190	4.670	4 866	6.614
		40	0.390	0.430	0.465	0.655	1115.3	560	4.365	4.860	5 057	6.850
29.39	....	60	0 518	0.593	0.620	0 888	1308.3	580	4.541	5.051	5.268	7.123
28.89	....	80	0.649	0.725	0.780	1.100	1525.3	600	4.725	5.247	5.455	7.388
27.99	....	100	0.787	0.898	0.939	1.338	1768.3	620	4.896	5 437	5.660	7.636
26.48	....	120	0.926	1.055	1.110	1.570	2041.3	640	5.082	5.627	5.850	7.893
24.04	....	140	1.051	1.209	1.265	1.794	2346.3	660	5.260	5.831	6.067	8.153
20.27	....	160	1.200	1.368	1.427	2.008	2705	680	5.442	6 020	6.260	8.400
14.63	....	180	1.345	1.528	1.597	2.255	3080	700	5.629	6.229	6.481	8.676
6 45	....	200	1.495	1 691	1.778	2.500		720	5.808	6.425	6.673	8.912
	2 5	220	1.634	1.852	1.936	2.720		740	6.006	6 635	6.899	9 203
	10 3	240	1.780	2.020	2.110	2.960		760	6.200	6.833	7.100	9.460
	20.7	260	1.931	2 183	2.279	3 189		780	6.389	7.046	7.314	9.736
	34.5	280	2.085	2.350	2 465	3.422		800	6.587	7.250	7.508	9.992
	52.3	300	2.233	2.519	2.630	3.665		820	6.779	7.464	7.757	10.272
	74.9	320	2.395	2.690	2.800	3.900		840	6.970	7.662	7.952	10.512
	103.3	340	2.543	2.862	2.988	4.145		860	7.176	7.888	8.195	10.814
	138.3	360	2.700	3.029	3.175	4.380		880	7.375	8.098	8.400	11.175
	180.9	380	2.859	3.211	3.350	4.628		900	7.579	8.313	8.639	11.360
	232 4	400	3.008	3.375	3.521	4.870		920	7.795	8.545	8.867	11 625
	293.7	420	3.182	3.566	3.720	5.118		940	7.989	8.755	9.089	11.911
	366.1	440	3.345	3 740	3.900	5.358		960	8.200	8.975	9.300	12.180
	451.3	460	3.511	3.929	4 096	5 612		980	8.406	9.196	9.547	12.473
	550.3	480	3.683	4.100	4.280	5.855		1000	8.617	9.421	9.776	12.747

[a]From *Piping Handbook*, by Walker and Crocker. This table gives the expansion from −20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2 519 − 0 593 = 1 926 in.

PIPE THREADS

All threaded pipe for heating and ventilating installations uses the American Standard taper pipe thread which is made with a taper of 1 in 16 measured on the diameter of the pipe so as to secure a tight joint. Threads of fittings are tapped to the same taper. The number of threads per inch varies with the different pipe sizes. All threaded pipe should be made up with a thread paste suitable for the service under which the pipe is to be used.

HANGERS AND SUPPORTS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom, or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to prevent excessive load on the bottom support when expansion takes place, or the bottom support should be designed to withstand the entire load.

TYPES OF FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast iron, but in addition to this malleable iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure.

As in the case of pipe, there are several weights of fittings manufactured. Recognized American Standards for the various weights are as follows:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in and larger), 125 lb, and 250 lb maximum saturated steam pressure.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure

Steel flanged fittings for 150 and 300 lb maximum steam service pressure

The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard to 500 lb for the 300 lb steel standard.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings, usually of wrought-iron only; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close or open pattern, and may be cast with either a back or side outlet; tees; crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another,

may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Fittings for copper tubing are available in the soldered, flared, or compression types. Illustrations of each of these types is shown in Fig. 2. Fittings for copper pipe of *IPS* dimensions are available in screwed or soldered types of connection.

The compression type fitting is generally limited to smaller size tubing while the flared and soldered types are used in large and small sizes. While no effort has been made to standardize dimensions of flared tube fittings, manufacturers have quite generally used *S.A.E.* standard dimensions. Flared tube fittings are widely used in refrigeration work and the use of *S.A.E.* dimensions and a 45-deg flare renders most fittings

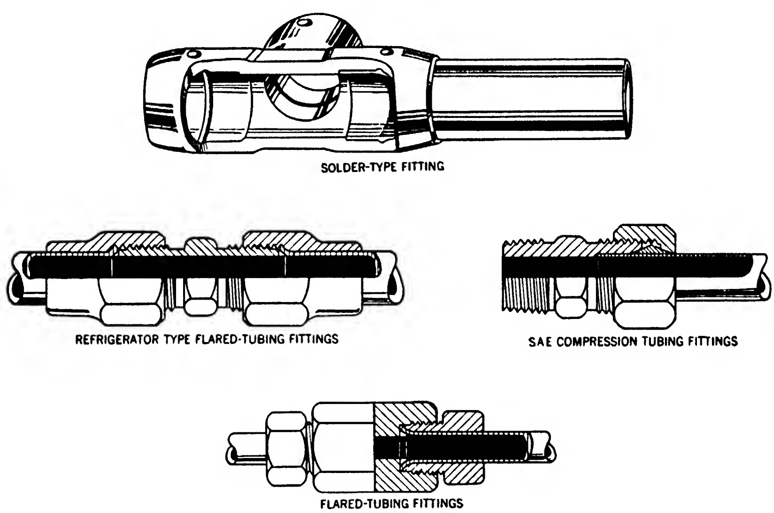


FIG. 2. COPPER OR BRASS TUBING FITTINGS

interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits.

Ammonia pipe fittings made of cast iron are extensively used in handling refrigerants in larger installations. Until recently, no standard dimensions were adhered to in the manufacture of ammonia flanged fittings or companion flanges with the result that fittings of different manufacturers were not interchangeable. A subcommittee of *A.S.A Sectional Committee* B16 has prepared proposed American Standard dimensions for ammonia flanged fittings and companion flanges for maximum service pressure of 300 lb per sq in. which will be available soon.

Thread Connections

Threads used for fittings are the same American Standard taper pipe threads as those used for pipe, and unless otherwise ordered, right-hand

threads are used. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of ¼ in. to the foot. These elbows are known to the trade as pitched elbows and are commercially available. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively.

Flanged fittings are generally used in the best practice for connecting all piping above 4 in. in diameter. While screwed fittings may be used for the larger sizes and are satisfactory under the proper working conditions, it will be found difficult either to make or to break the joints in these large sizes.

A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 lb pressure and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a 1/16-inch raised face. The standard facing for steel flanged fittings for 150 and 300 lb is a 1/16-inch raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed fittings are given in Table 7, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Tables 8 and 9.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a relatively narrow recessed facing.

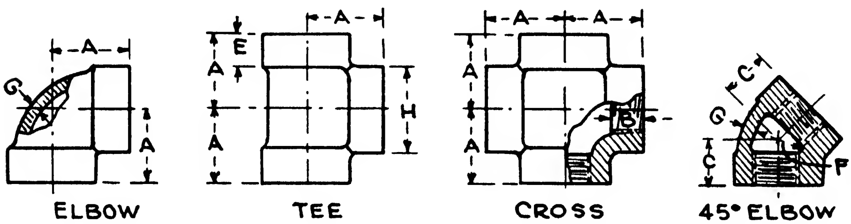
WELDING

Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as a competitive method to the screwed and flanged joint. Since the question of economy of welding as against the use of screwed and flanged fittings is dependent on the individual job, the use of welding is generally recommended on the basis of a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding.

Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds.

Welding application requires the same basic knowledge of design as do the other types of assembly, but in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. This requirement applies equally to employer and employee with the employer accepting

TABLE 7. TENTATIVE AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45 DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON SCREWED FITTINGS



	A	C	B	E	F		G	H
Nominal Pipe Size	Center to End Elbows Tees and Crosses	Center to End, 45 Deg Elbows	Length of Thread Min	Width of Band, Min	Inside Diameter of Fitting Min	Max	Metal Thickness, Min	Outside Diameter of Band, Min
1/4	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
3/8	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
1/2	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
3/4	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1 1/4	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1 1/2	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2 1/2	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3 1/2	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50	5.97	1.88	1.88	12.750	12.850	0.800	15.47
14 O.D	10.40	.	2.00	2.00	14.000	14.100	0.880	16.94
16 O.D.	11.82	.	2.20	2.20	16.000	16.100	1.000	19.30

All dimensions given in inches

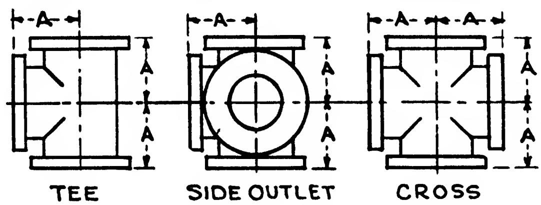
all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests. Industry will not accept the employment of mechanics of undetermined ability nor on the basis of past experience. Neither does industry accept the statement that a weld is only as good as the workman who makes it. The control Codes now in process of adoption will be the law governing

the use of the welding process. These Codes prohibit individual practices contrary to their specified procedure and rules of control, and this is predicated upon the sound requirement that the employer must assume full responsibility for the deposited weld.

It is advisable that this management responsibility be included in all welding specifications and that authoritative standards of workmanship also be specified. The standards of workmanship for this industry are as set forth in the *Standard Manual on Pipe Welding* of the *Heating, Piping and Air Conditioning Contractors National Association.*

A complete line of manufactured steel welding fittings is now available

TABLE 8. AMERICAN STANDARD DIMENSIONS OF TEES AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON FLANGED FITTINGS



NOMINAL PIPE SIZE a-b	*A* CENTER TO FACE TEES AND CROSSES b-c	*AA* FACE TO FACE TEES AND CROSSES b-c	DIAMETER OF FLANGE	THICKNESS OF FLANGE, MIN	METAL THICKNESS OF BODY, MIN
1	3½	7	4¼	7/16	7/16
1¼	3¾	7½	4⅝	½	7/16
1½	4	8	5	9/16	7/16
2	4½	9	6	⅝	7/16
2½	5	10	7	11/16	7/16
3	5½	11	7½	¾	7/16
3½	6	12	8½	13/16	7/16
4	6½	13	9	15/16	½
5	7½	15	10	15/16	½
6	8	16	11	1	9/16
8	9	18	13½	1⅛	⅝
10	11	22	16	1 3/16	¾
12	12	24	19	1¼	13/16
14 O.D.	14	28	21	1⅜	⅞
16 O.D.	15	30	23½	1 7/16	1
18 O D.	16½	33	25	1 9/16	1 1/16
20 O D.	18	36	27½	1 11/16	1⅛
24 O.D.	22	44	32	1⅞	1¼
30 O.D.	25	50	38¾	2⅛	1 7/16
36 O.D.	28	56	46	2⅜	1⅝
42 O D	31	62	53	2⅝	1 13/16
48 O D.	34	68	59½	2¾	2

All dimensions given in inches.

aSize of all fittings listed indicates nominal inside diameter of port.

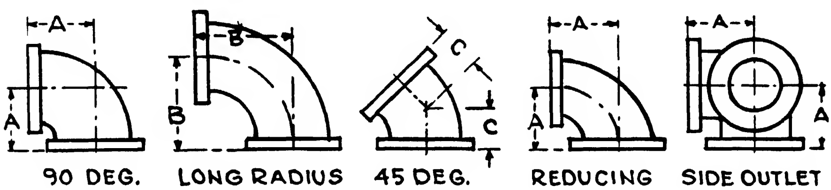
bTees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet

cTees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening

with plain ends machine beveled for welding and with radii similar to short and long radius flanged fittings. Some typical types of these fittings are shown in Fig. 3. They are made in pipe sizes ¾ to 24 in., standard and extra heavy, in steel, wrought iron, brass, copper, and special alloys.

Socket welding fittings of forged steel are also commercially available. These fittings have a machined recess into which the pipe slips. A fillet weld between the pipe and socket edge provides a pressure-tight joint. A proposed American Standard containing dimensions of steel welding-neck flanges for pressures up to 1500 lb per sq in. has been developed in *A.S.A*

TABLE 9. AMERICAN STANDARD DIMENSIONS OF ELBOWS FOR 125 LB CAST-IRON FLANGED FITTINGS



Nominal Pipe Size[a]	A Center to Face Elbow[b-c-d]	B Center to Face Long Radius Elbow[b-c-d]	C Center to Face 45 Deg Elbow[c]	Diameter of Flange	Thickness of Flange, Min	Metal Thickness of Body, Min
1	3½	5	1¾	4¼	7/16	7/16
1¼	3¾	5½	2	4⅝	½	7/16
1½	4	6	2¼	5	9/16	7/16
2	4½	6½	2½	6	⅝	7/16
2½	5	7	3	7	11/16	7/16
3	5½	7¾	3	7½	¾	7/16
3½	6	8½	3½	8½	13/16	7/16
4	6½	9	4	9	15/16	½
5	7½	10¼	4½	10	15/16	½
6	8	11½	5	11	1	9/16
8	9	14	5½	13½	1⅛	⅝
10	11	16½	6½	16	1 3/16	¾
12	12	19	7½	19	1¼	13/16
14 O.D.	14	21½	7½	21	1⅜	⅞
16 O.D.	15	24	8	23½	1 7/16	1
18 O.D.	16½	26½	8½	25	1 9/16	1 1/16
20 O.D.	18	29	9½	27½	1 11/16	1⅛
24 O.D.	22	34	11	32	1⅞	1¼
30 O.D	25	41½	15	38¾	2⅛	1 7/16
36 O.D	28	49	18	46	2⅜	1⅝
42 O D.	31	56½	21	53	2⅝	1 13/16
48 O.D.	34	64	24	59½	2¾	2

All dimensions given in inches

[a]Size of all fittings listed indicates nominal inside diameter of port

[b]Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening

[c]Special degree elbows, ranging from 1 to 45 deg, inclusive, have the same center to face dimensions as given for 45 deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90 deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces

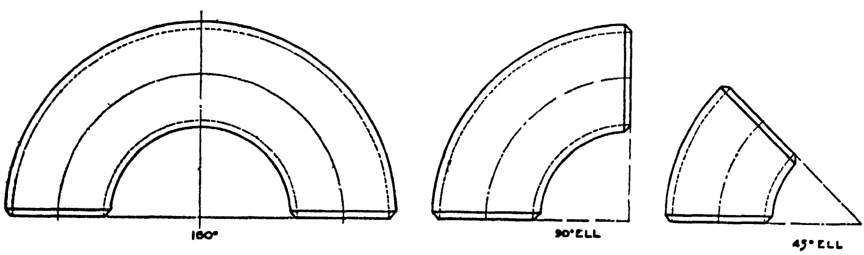
[d]Side outlet elbows shall have all openings on intersection center-lines

Sectional Committee B16. Tables 10 and 11 give these dimensions for welding-neck flanges suitable for 150 and 300 lb per sq in. gage pressure.

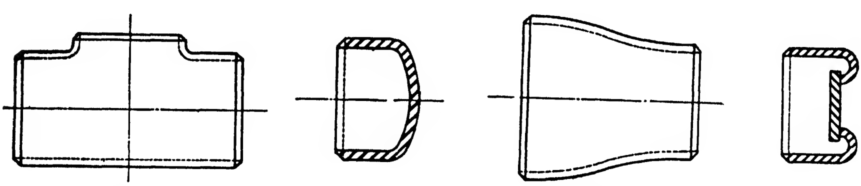
VALVES

Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger sizes either cast-iron, cast-steel or some of the steel alloys are employed.



a. TYPICAL SHORT RADIUS ELBOWS



b. TEE

c. FORGED CAP

d. CONCENTRIC REDUCER

e. END CLOSURE

FIG. 3. TYPICAL WELDING FITTINGS

Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

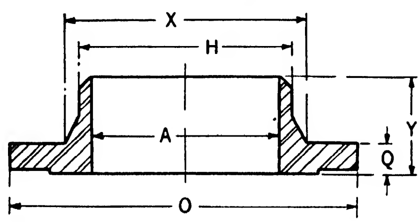
Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure. The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum. These valves may be secured with either a rising or a non-rising stem, although in the smaller sizes the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether

the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive to manufacture than the gate valve, but its peculiar construction offers a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between

TABLE 10. PROPOSED DIMENSIONS OF STEEL WELDING NECK FLANGES FOR MAXIMUM STEAM SERVICE PRESSURE OF 150 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 500 F, AND 100 LB AT 750 F



NOMINAL PIPE SIZE	DIAMETER OF FLANGE	THICKNESS OF FLG MIN	DIAMETER OF HUB	HUB DIAM. BEGINNING OF CHAMFER	LENGTH THRU HUB	DIAM FOR STANDARD PIPE	DIAM OF BOLT CIRCLE	NO OF BOLTS	SIZE OF BOLTS
	O	*Q*	*X*	*H*	*Y*	*A*			
1	4 1/4	7/16	1 15/16	1.32	2 3/16	1.05	3 1/8	4	1/2
1 1/4	4 5/8	1/2	2 5/16	1.66	2 1/4	1.38	3 1/2	4	1/2
1 1/2	5	9/16	2 9/16	1.90	2 7/16	1.61	3 7/8	4	1/2
2	6	5/8	3 1/16	2.38	2 1/2	2.07	4 3/4	4	5/8
2 1/2	7	11/16	3 9/16	2.88	2 3/4	2.47	5 1/2	4	5/8
3	7 1/2	3/4	4 1/4	3.50	2 3/4	3.07	6	4	5/8
3 1/2	8 1/2	13/16	4 13/16	4.00	2 13/16	3.55	7	8	5/8
4	9	15/16	5 5/16	4.50	3	4.03	7 1/2	8	5/8
5	10	15/16	6 7/16	5.56	3 1/2	5.05	8 1/2	8	3/4
6	11	1	7 9/16	6.63	3 1/2	6.07	9 1/2	8	3/4
8	13 1/2	1 1/8	9 11/16	8.63	4	7.98	11 3/4	8	3/4
10	16	1 3/16	12	10.75	4	10.02	14 1/4	12	7/8
12	19	1 1/4	14 3/8	12.75	4 1/2	12.00	17	12	7/8
14 O D	21	1 3/8	15 3/4	14.00	5	13.25	18 3/4	12	1
16 O. D.	23 1/2	1 7/16	18	16.00	5	15.25	21 1/4	16	1
18 O. D.	25	1 9/16	19 7/8	18.00	5 1/2	17.25	22 3/4	16	1 1/8
20 O. D	27 1/2	1 11/16	22	20.00	5 11/16	19.25	25	20	1 1/8
24 O. D.	32	1 7/8	26 1/8	24.00	6	23.25	29 1/2	20	1 1/4

All dimensions given in inches

A raised face of 1/16 in is included in *thickness of flange minimum*

It is recommended that the taper of the hub should not exceed 6 deg for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding

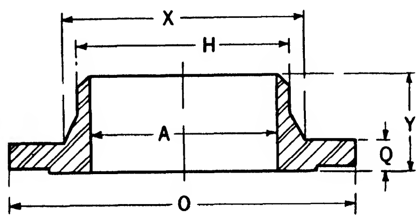
the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat.

Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever

handles are often supplied to indicate the relative opening of the valve in any position. Standard roughing-in dimensions for angle-type valves are given in Table 12.

Automatic control of steam supply to individual radiators can be

TABLE 11. PROPOSED DIMENSIONS OF STEEL WELDING NECK FLANGES FOR MAXIMUM STEAM SERVICE PRESSURE OF 300 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 750 F



NOMINAL PIPE SIZE	DIAM OF FLANGE	THICKNESS OF FLANGE MIN	DIAM OF HUB	HUB DIAM BEGINNING OF CHAMFER	LENGTH THRU HUB	DIAM FOR STANDARD PIPE	DIAM FOR EXTRA STRONG PIPE	DIAM OF BOLT CIRCLE	NO. OF BOLTS	SIZE OF BOLTS
	O	*Q*	*X*	*H*	*Y*	*A*	*A*			
*2	6½	⅞	3 5/16	2 38	2¾	2.07	1.94	5	8	⅝
2½	7½	1	3 15/16	2 88	3	2.47	2 32	5⅞	8	¾
3	8¼	1⅛	4⅝	3.50	3⅛	3.07	2 90	6⅝	8	¾
3½	9	1 3/16	5¼	4 00	3 3/16	3.55	3.36	7¼	8	¾
4	10	1¼	5¾	4.50	3⅜	4.03	3 83	7⅞	8	¾
5	11	1⅜	7	5.56	3⅞	5.05	4.81	9¼	8	¾
6	12½	1 7/16	8⅛	6 63	3⅞	6 07	5.76	10⅝	12	¾
8	15	1⅝	10¼	8.63	4⅜	7 98	7 63	13	12	⅞
10	17½	1⅞	12⅝	10.75	4⅝	10 02	9 75	15¼	16	1
12	20½	2	14¾	12.75	5⅛	12 00	11 75	17¾	16	1⅛
14 O D.	23	2⅛	16¾	14 00	5⅝	13 25	....	20¼	20	1⅛
16 O. D	25½	2¼	19	16.00	5¾	15.25	....	22½	20	1¼
18 O. D	28	2⅜	21	18.00	6¼	17.25	....	24¾	24	1¼
20 O. D.	30½	2½	23⅛	20 00	6⅜	19.25	....	27	24	1¼
24 O D	36	2¾	27⅝	24.00	6⅝	23.25	....	32	24	1½

*For sizes below 2 in use dimensions of 600 lb flanges

All dimensions given in inches

A raised face of 1/16 in is included in *thickness of flange minimum*

It is recommended that the taper of the hub should not exceed 6 deg for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding.

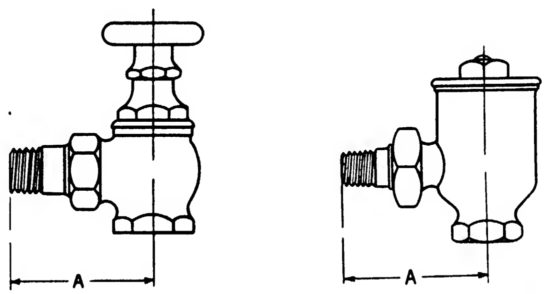
effected by use of direct-acting radiator valves having a thermostatic element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit individual taste.

Ordinary steam valves may be used for hot water service by drilling a 1/16-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made particularly

for use in hot water heating systems are of less complex design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when

TABLE 12. STANDARD ROUGHING-IN DIMENSIONS ANGLE TYPE VALVES



Size of Valve	Dimension A Steam and Hot Water Angle Valves and Union Elbows Effective January 1, 1926	Dimension A Modulating Valves Effective January 1, 1926	Dimension A Return Line Vacuum Valves Effective January 1, 1925
½	2¼	2¾	3¼
¾	2¾	2¾	......
1	3	3	......
1¼	3½	3½	......
1½	3¾	3¾	......
2	4¼	4¼	......
Tolerance	±⅛	±⅛	.....

All dimensions given in inches

Connecting ends shall be threaded and gaged as to threading according to the American (Taper) Pipe Thread Standard, *A.S.A.* No. B2—1919

The standardization of the Roughing-in Dimensions of Angle Steam and Hot Water, and Modulating Radiator Valves was made possible by the cooperation of the *Manufacturers Standardization Society of the Valves and Fittings Industry*

a vacuum is formed by the condensation of steam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when the pressure dies down and a vacuum tends to be formed the air is drawn back into the radiator.

A system operating continuously or intermittently and supplied with vacuum valves will generally heat more quickly than one provided with non-vacuum air valves; thus, it will effect considerable economy of fuel because the idle period during which no heat is delivered is shortened. In those cases, where a system is equipped with vacuum air valves and which has been cold for several days, the system will probably have an

internal pressure within the radiator closely approaching atmospheric. At such times, the vacuum valve will not vent the system any more rapidly than the ordinary type. Automatic air valves are provided with a float to close them in case the radiator becomes flooded with water because it does not drain properly.

CORROSION[2]

Corrosion is sometimes encountered in heating work on the outside of buried pipes or the inside of steam heating systems; it is seldom experienced in hot water heating systems unless the water is frequently renewed. Piping buried in the ground is quite successfully protected by coatings of the asphaltic type which are usually applied hot and often reinforced with fabric wrappings. Galvanizing by the hot-dip process and painting with specially prepared mixtures also afford some protection.

Internal corrosion in steam heating systems occurs principally in the condensate return pipes and is nearly always caused by oxygen or carbon dioxide, or both, in solution in the condensate. Oxygen may enter the heating system with the steam, owing to its presence in the boiler-feed water, or it may enter as air through small leaks, particularly in systems which operate at sub-atmospheric pressures. When a steam heating system is operated intermittently, air rushes in during each shutdown period and oxygen is absorbed by the condensate which clings to the interior surfaces of the pipes and radiators. The rate of corrosion depends upon the amounts of oxygen and carbon dioxide present in solution, upon the operating temperature, and upon the length of time that the pipe surfaces are in contact with gas-laden condensate.

Another possible cause of corrosion is a flow of electric current sometimes resulting from faulty electrical circuits which should be corrected. Electrolytic corrosion also may occur because of the presence of two dissimilar metals, such as brass and iron, but the condensate in practically all steam heating systems is such a weak electrolyte that this cause of corrosion is very infrequent.

If trouble is experienced from corrosion, oxygen should be eliminated from the feed water by proper deaeration with commercial apparatus. The elimination of the oxygen due to air leakage is more difficult because of the multitude of small leaks which exist around valve stems and in pipe joints. In vacuum systems, however, an attempt should be made to minimize such leakage.

Carbon dioxide in varying amounts is contained in steam produced from the majority of water supplies. It is formed from the breaking down of carbonates and bicarbonates which are present in nearly all natural waters. It can be partly removed by chemical treatment and deaeration, but there is no simple method whereby it can be entirely eliminated.

These gases cause corrosion only when in solution in the condensate; when they are mixed with dry steam their corrosive effect is negligible.

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[2]New Light on Heating System Corrosion, by J H Walker (*Heating and Ventilating*, May, 1933) A.S H V E RESEARCH REPORT No 983—Corrosion Studies in Steam Heating Systems, by R R Seeber, F A Rohrman and G E Smedberg, (A S H V E TRANSACTIONS, Vol 40, 1934, p 253) A S H V E RESEARCH REPORT No 1037—Corrosion Studies in Steam Heating Systems, by R R Seeber, F A Rohrman and G. E Smedberg, (A.S H V E TRANSACTIONS, Vol 42, 1936, p 263) Corrosion Studies in Steam Heating Systems, by R R Seeber and Margaret R Holley (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1937, p 387)

The amount of gas in solution depends upon the partial pressure of that gas in the atmosphere above the surface of the solution, in accordance with the well known physical law of Henry and Dalton[8]. The exact application of this law, however, assumes equilibrium conditions which do not always exist under the flow conditions prevailing in a heating system.

Distinction should be made between corrosion in heating systems proper and in the condensate discharge lines from other apparatus using steam, such as water heaters, kitchen equipment, and sterilizers. Experience has shown that in heating systems the partial pressures of the gases do not reach such magnitudes as to cause harmful amounts of gas to become dissolved in the condensate when steam supplies are of reasonable purity. In other kinds of steam-using apparatus which are not ordinarily well vented, the gases tend to accumulate in the steam space and to become dissolved in the condensate in appreciable concentrations. Consequently, corrosion is frequently observed in the condensate discharge lines from such apparatus, but this does not necessarily indicate that equally serious corrosion is taking place in the heating system supplied with steam from the same source.

When corrosive conditions are believed to exist, their seriousness should be determined by actual measurement, rather than by inference from isolated instances of pipe failures. The *National District Heating Association* has perfected a corrosion tester for measuring the inherent corrosiveness of existing conditions. This corrosion tester consists of a frame supporting three coils of wire which are carefully weighed. After the tester has been inserted in the pipe line for a definite length of time, the loss of weight of the coils, referred to an established scale, indicates the relative corrosiveness of the condensate. Accompanying such corrosion measurements, a careful chemical analysis should be made of the condensate, and the findings will serve as a basis for an intelligent study of the problem.

Corrosion, if found to exist, can be lessened or overcome by several means. If the steam supply is found to be definitely contaminated, proper chemical treatment of the water, followed by deaeration, is an obvious remedy. The leaks in the piping system, particularly in vacuum systems, should be stopped so far as is practicable.

Some success has been reported with the use of inhibitors, chief among which are oil, and sodium silicate. Oil may be fed into the main steam-supply pipe by means of a sight-feed lubricator. The type of oil known as 600-W is usually recommended. In the present state of knowledge on this point, the quantity to be fed can best be determined by trial. The use of sodium silicate, fed in a similar manner, is reported to be successful but it has not been widely used.

In view of the fact that corrosion is most frequently found in the return lines from special equipment, which constitute a relatively small part of the total piping in a building, a simple solution of the corrosion problem may be to use non-corroding materials in those certain portions of the piping system, since the higher cost will usually be an unappreciable portion of the total. Brass and copper are undoubtedly less subject to

[8]Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R E. Hall and A R Mumford (A S H V E TRANSACTIONS, Vol. 38, 1932, p 121).

this type of corrosion than the ferrous metals, and considerable attention is now being given to corrosion-resistant linings for ferrous pipe. Cast-iron pipe, sometimes alloyed with other metals, also deserves consideration.

PROBLEMS IN PRACTICE

1 ● What is the meaning of IPS brass pipe?

It means that the brass pipe has the same external diameter as steel pipe in the same nominal pipe size and that the wall thickness is sufficient to allow cutting of threads for use with standard size threaded fittings.

2 ● Why is thin-walled copper pipe made up with sweated joints?

If the pipe were threaded it would be necessary to use at least standard-weight wall thickness on account of the metal removed in threading. Flared ends with coupling nuts may be used, but this construction is expensive and hard to keep tight.

3 ● How are pipes designated in diameters of 12 in. and less?

By weight and nominal size, referring to the approximate inside diameter.

4 ● How are pipe sizes designated in diameters of 14 in. and more?

By wall thickness and outside diameter

5 ● Why are expansion joints required in steam pipes?

To care for the change in length of the line brought about by a change in temperature.

6 ● What devices are used for taking up expansion?

Expansion joints, swivel joints, and the inherent flexibility of the pipe itself.

7 ● Where are swivel joints principally used?

In branch connections to radiators, and in the risers of multi-story buildings where they are installed between the floor joists.

8 ● Name three grades of American Standard screwed pipe fittings.

125-lb cast-iron, 150-lb malleable iron, and 250-lb cast-iron.

9 ● In what sizes are American Standard cast-iron flanges and flanged fittings for 25-lb saturated steam pressure made?

In nominal sizes from 4 in. to 72 in., inclusive.

10 ● What fittings are generally used for threaded connections in low pressure heating systems?

Cast-iron

Chapter 19

GRAVITY WARM AIR FURNACE SYSTEMS

Design Procedure, Estimating Heating Requirements, Leader Pipe Sizes, Proportioning Wall Stacks, Register Selections, Recirculating Ducts and Grilles, Furnace Return Connection, Furnace Capacity, Examples, Booster Fans

WARM air heating systems of the gravity type are described in this chapter[1], and those of the mechanical type are described in Chapter 20. In the gravity type, the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head. Booster fans are often used in conjunction with gravity-designed systems to increase air circulation.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal or brick, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard.

The air supply to the furnace may be taken (1) entirely from inside the building through one or more recirculating ducts, (2) entirely from outside the building, in which case no air is recirculated, or (3) through a combination of the inside and the outside air supply systems.

DESIGN PROCEDURE

The design of a furnace heating system involves the determination of the following items:

1. Heat loss in Btu from each room in the building
2. Area and diameter in inches of warm-air pipes in basement (known as leaders).
3. Area and dimensions in inches of vertical pipes (known as wall stacks).
4. Free and gross area and dimensions in inches of warm-air registers.
5. Area and dimensions of recirculating or outside air ducts, in inches.
6. Free and gross area and dimensions in inches of recirculating registers.

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[1]All figures and much of the engineering data which follow are from University of Illinois, *Engineering Experiment Station Bulletins* Nos 141, 188, 189 and 246, Warm Air Furnaces and Heating Systems, by A C Willard, A P Kratz, V S Day, and S Konzo

7. Size of furnace necessary to supply the warm air required to overcome the heat loss from the building. This *size* should include square inches of leader pipe area which the furnace must supply. It is also desirable to call for a minimum bottom fire-pot diameter in inches, which is the nominal grate diameter.

8. Area and dimensions in inches of chimney and smoke pipe. If an unlined chimney is to be used, that fact should be made clear.

The heat loss calculations should be made in accordance with the procedure outlined in Chapter 7, taking into consideration the transmission losses as well as the infiltration losses.

LEADER PIPE SIZES

In a gravity circulating warm-air furnace system the size of the leader to a given room depends upon the temperature of the warm air entering the room at the register. A reasonable air temperature at the registers must, therefore, be chosen before the system can be designed. The *National Warm Air Heating and Air Conditioning Association* has approved an air temperature of 175 F at the registers as satisfactory for design purposes. At this temperature, the heat-carrying capacity (heat available above 70 F) per square inch of leader pipe per hour for first, second or third floors is shown by Fig. 1 at 175 F to be 105, 170 and 208 Btu, respectively. For average calculations, the values 111, 166 and 200 will simplify the work and may be satisfactorily substituted for these heat-carrying capacities. If H represents the total heat to be supplied any room, the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{111} = \text{approximately } 0.009H \qquad (1)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \qquad (2)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{200} = \text{approximately } 0.005H \qquad (3)$$

In designing for a lower warm-air register temperature, say 160 F, the factors 111, 166 and 200 become 80, 140 and 166 (Fig. 1 at 160 F), and the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{80} = \text{approximately } 0.012H \qquad (4)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{140} = \text{approximately } 0.007H \qquad (5)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \qquad (6)$$

These equations are applicable to straight leaders from 6 to 8 ft in length. Longer leaders must be thoroughly covered or the vertical stacks must be increased in area as discussed under wall stacks. If some provision is not made for these longer leaders, the air temperature may be much lower than anticipated and the room will not be properly heated.

The values shown in Fig. 1 apply only to the case where the straight, leader pipe is 8 ft in length and is connected to stacks whose cross-sectional area is approximately 75 per cent of that of the leader pipe.

Any deviation from these conditions requires a modification of the constants used in Equations 1, 2, and 3. The temperature drop in leaders of various lengths at three different register temperatures is shown in Fig. 2, and should be used to obtain new register temperatures, lower than 175 F, on which to base selections from the curves of Fig. 1, and thereby new constants for Equations 1, 2 and 3.

Leader sizes should in general be not less than those obtained by Equations 1 to 3 nor should leaders less than 8 in. in diameter be used. It is not considered good commercial practice to specify diameters except

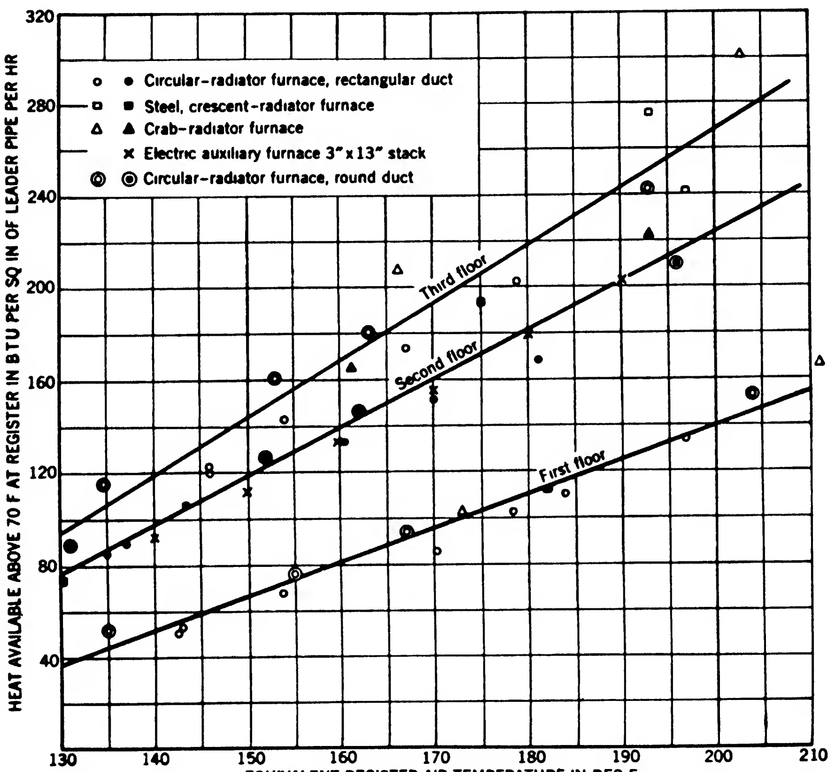

FIG. 1. VALUE OF SQUARE INCH OF LEADER PIPE AREA FOR FIRST, SECOND, AND THIRD FLOORS FOR SIMPLE SYSTEM HAVING LEADERS 8 FT IN LENGTH

in whole inches. The tops of all leaders should be at the same elevation as they leave the furnace bonnet, and from this point there should be a uniform up-grade of 1 in. per foot of run in all cases. Leaders over 12 ft in length should be avoided if possible. In cases where such leaders are required, the use of a larger size pipe, than is required by the application of the equations, smooth transition fittings, and duct insulation are recommended.

PROPORTIONING WALL STACKS

The wall stack for an upper floor should be made not less than 70 per cent of the area of the leader. In cases where the leader is short and straight as was the case for Fig. 1, such a practice is probably justified, since the loss (Fig. 3) in capacity occasioned by the smaller stack is not serious for stacks having areas in excess of 70 per cent of the leader area. For leaders over 8 ft in length or for leaders which are not straight, the ratio of stack area to leader area should be greater than 70 per cent in

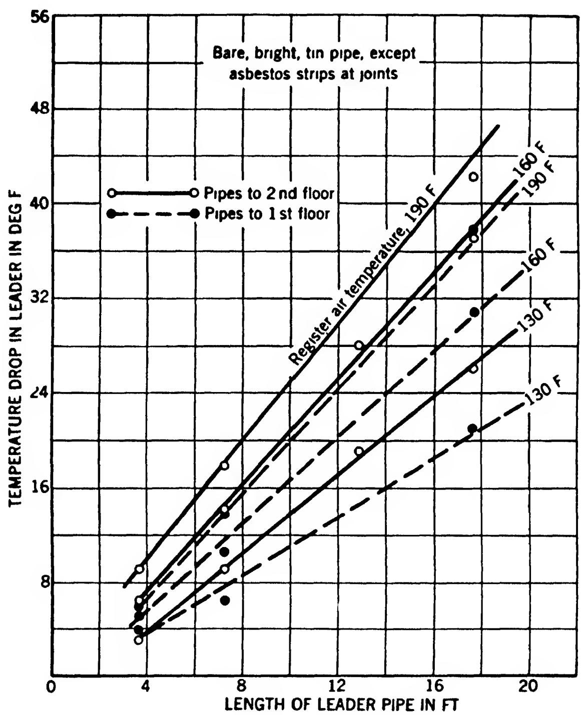

FIG. 2. INFLUENCE OF LEADER PIPE LENGTH ON TEMPERATURE LOSS IN AIR FLOWING THROUGH PIPE

order to offset the greater temperature losses (Fig. 2) in the longer leader. In gravity circulating systems, this ratio of stack to leader area is a very important matter.

The curves in Figs. 4 and 5 indicate that for rooms having a heat requirement exceeding approximately 9000 Btu per hr, exceedingly high register temperatures are required for stacks whose width is less than 3½ in. For such requirements either multiple stacks, or stacks having larger cross-sectional area (placed in 6 in. studding spaces) will be required.

REGISTER SELECTIONS

The registers used for discharging warm air into the rooms should have a free or net area not less than the area of the leader in the same run of piping. The free area should be at least 70 per cent of the gross area of the register. No upper-floor register should be wider horizontally than the wall stack, and it should be placed either in the baseboard or side wall, if this can be done without the use of offsets. First-floor registers may be of the baseboard or floor type, with the former location preferred. High

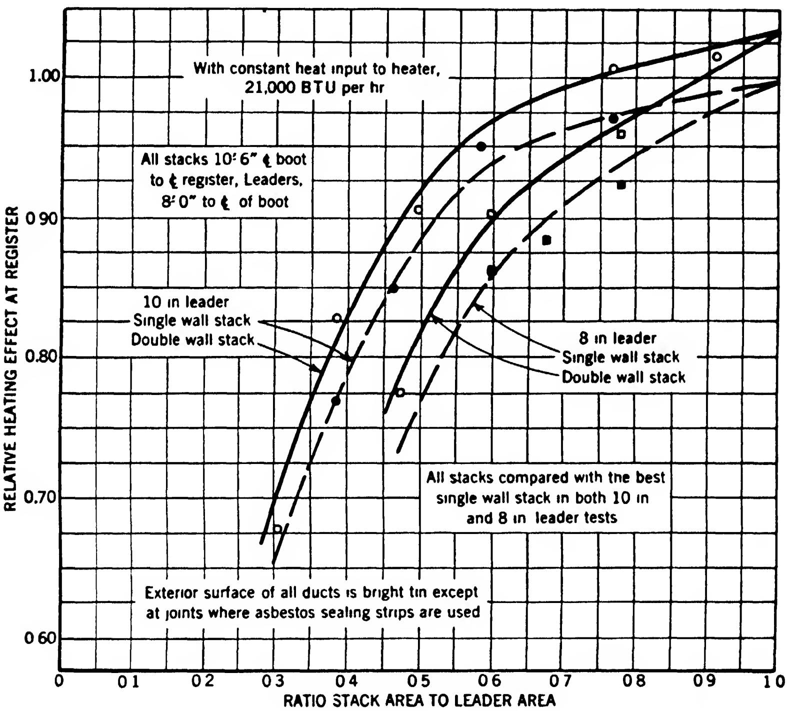

FIG. 3. RELATIVE HEATING EFFECT OF STACKS AT CONSTANT HEAT INPUT TO FURNACE

sidewall locations for warm air registers in gravity circulating systems are not recommended on account of the tendency for stratification of the air in the room, resulting in high temperatures at the ceiling.

RECIRCULATING DUCTS AND GRILLES

The ducts through which air is returned to the furnace should be designed to minimize friction and turbulence. They should be of ample area, in excess of the total area of warm-air pipes, and at all points where

the air stream must change direction or shape, streamline fittings should be employed. Horizontal ducts should pitch at least ½ in. per foot upward from the furnace.

The recirculating grilles (or registers) should have a free area at least equal to the ducts to which they connect, and their free area should never be less than 50 per cent of their gross area.

The location and number of return grilles will depend on the size, details and exposure of the house. Small compactly built houses may frequently be adequately served by a single return effectively placed in a central hall. More often it is desirable to have two or more returns, provided, however, that in two-story residences one return is placed to effectively receive the cold air returning by way of the stairs.

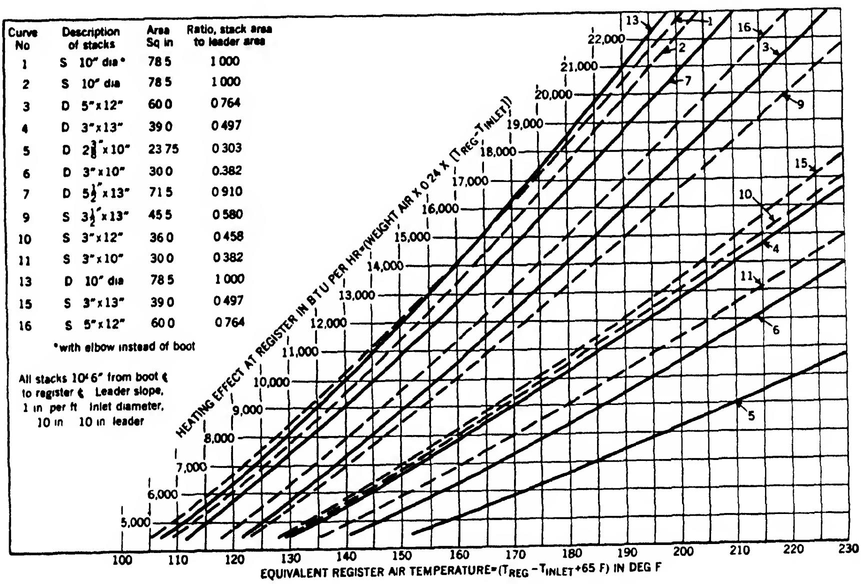

FIG. 4 HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 10-IN. LEADER

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus in rooms having only small windows the grille should be brought as close to the furnace as possible, but if the room has a bay window, French doors, or other large sources of cooling or leakage of cold air, the grille should be placed close by, so as to collect the cool air and prevent drafts. When long ducts of this type are employed they must be made oversize. This precaution is particularly important when long ducts and short ducts are used in the same system. The long ducts must be oversize, if they are to operate satisfactorily in parallel with short ducts.

Return ducts from upstairs rooms may be necessary in apartments or other spaces which are closed off or badly exposed. Metal linings are

advisable in such ducts. It is important that these ducts be free from unnecessary friction and turbulence, and that they be located to prevent preheating of the air before it reaches the furnace.

Furnace Return Connection

Circulation of the air is accelerated if the return connection to the furnace is through a round inclined pipe connected to two 45 deg elbows rather than through a vertical pipe connected to two 90 deg elbows. The top of the return shoe should enter the casing below the level of the grate in the furnace. In order to accomplish this the shoe must be wide as is indicated in Fig. 6, No. 1 arrangement.

Tests of six different systems of cold air returns, Fig. 6, made at the University of Illinois[2], resulted in the following conclusions:

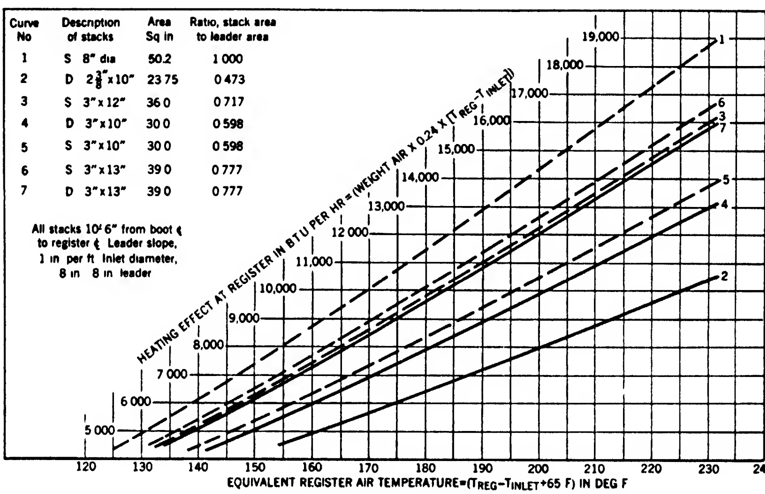

FIG. 5. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 8-IN. LEADER

1. In general, somewhat better room temperature conditions may be obtained by returning the air from positions near the cold walls.
2. Friction and turbulence in elaborate return duct systems retard the flow of air, and may seriously reduce furnace efficiency, and lessen the advantages of such a design.
3. The cross-sectional duct area is not the only measure of effectiveness. Friction and turbulence may operate to make the air flow out of all proportion to the various duct areas.

FURNACE CAPACITY

The size of furnace should, of course, be such as will provide the necessary air heating capacity, usually expressed in square inches of leader pipe area, and at the same time provide a grate of the proper area to burn the necessary fuel at a reasonable chimney draft. The total leader pipe area required is obtained by finding the sum of the leader pipe areas as already designated.

[2]Investigation of Warm Air Furnaces and Heating Systems, Part IV, by A. C. Willard, A. P. Kratz, and V. S Day (University of Illinois, *Engineering Experiment Station Bulletin* No. 189).

The grate area will depend on several factors of which four are very important. First of all, the air temperature at the register for which the plant has been designed must be determined. Usually, this temperature is taken at 175 F. Second in importance is the combustion rate, *which must always correspond with the register air temperature*, as is shown by a set of typical furnace performance curves (Fig. 7) for a cast-iron, circular radiator furnace with a 23 in. diameter grate and 50 in. diameter casing. The third factor is efficiency, which is a function of the combustion rate, and varies with it as shown by the efficiency curve of Fig. 7. The fourth factor is the heat value per pound of fuel burned, which was 12,790 Btu. This is not shown on the curves since it was constant for all combustion rates.

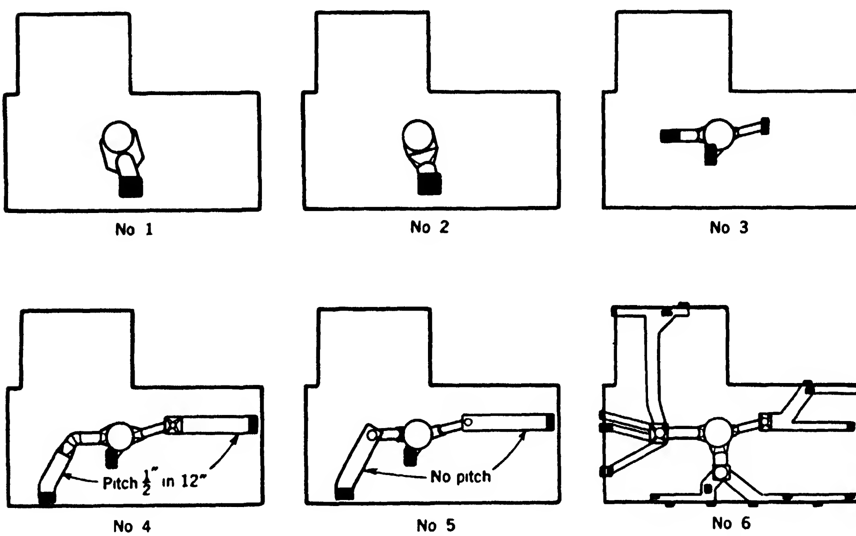

FIG. 6. ARRANGEMENT OF COLD AIR RETURNS FOR SIX INSTALLATIONS

It may be noted from Fig. 7 that for this particular furnace a register temperature of 175 F was accompanied by a combustion rate of approximately 7.5 lb per sq ft per hr, a capacity at the bonnet of 152,000 Btu per hr and a furnace efficiency of 58 per cent. Under these conditions the capacity at the bonnet per square foot of grate was equivalent to a value of 52,800 Btu per hr and per square inch of grate was equivalent to 367 Btu per hr. If it is desired to use these curves to select a furnace to deliver air at 175 F register temperature in a house where the total heat loss is H Btu per hour and the loss between the furnace and the registers is 0.25 H Btu per hour, the area of the grate in square inches will be $\frac{1.25\,H}{367} = 0.0034\,H$.

If, on the other hand, it is desired to select a furnace to deliver air at 160 F register temperature, the combustion rate is 5.5 lb and the efficiency

of the furnace is 62 per cent. Under this condition the capacity at the furnace bonnet per square foot of grate is 43,200 Btu per hr and per square inch of grate is 300 Btu per hr, the required area of the grate in square inches in this case will be $\frac{1.25\,H}{300} = 0.0042\,H$. It should be noted that a larger grate area is required if the furnace is to deliver air at a lower register temperature.

The typical performance curves shown in Fig. 7 are not applicable to

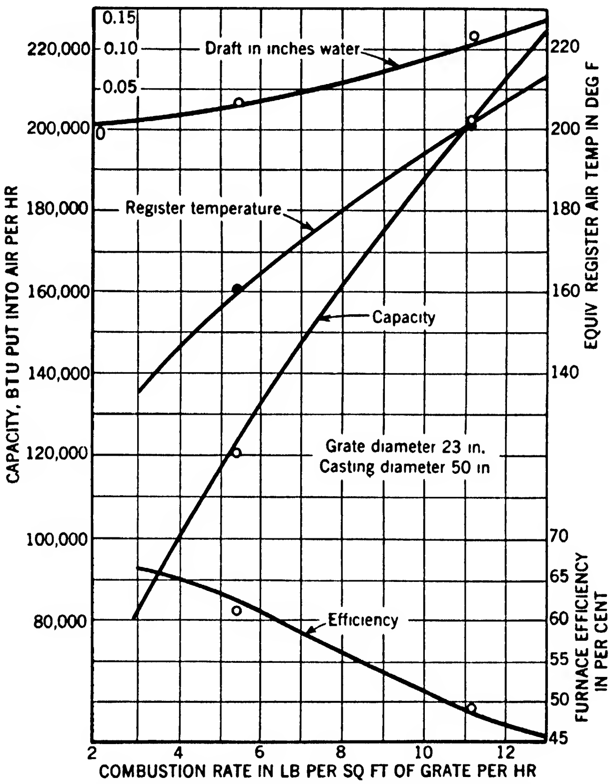

FIG. 7. TYPICAL PERFORMANCE CURVES FOR A WARM-AIR FURNACE AND INSTALLATION IN A THREE-STORY TEN LEADER PLANT, OPERATING ON RECIRCULATED AIR

all furnaces and hence for ordinary design purposes the values recommended in the Standard Code[3] should be used. The equation for a furnace having a ratio of heating surface to grate area of 20 to 1 is equal to:

$$H = \frac{G \times p \times f \times E_1 \times E_2 \times 0.866}{144} \qquad (7)$$

[3]Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences This code has been sponsored by the *National Warm Air Heating and Air Conditioning Association*, the *National Association of Sheet Metal Contractors*, and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS It is recommended that the installation of all gravity warm air heating systems in residences be governed by the provisions of this code, the ninth edition of which may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 50 W Broad St, Columbus, Ohio

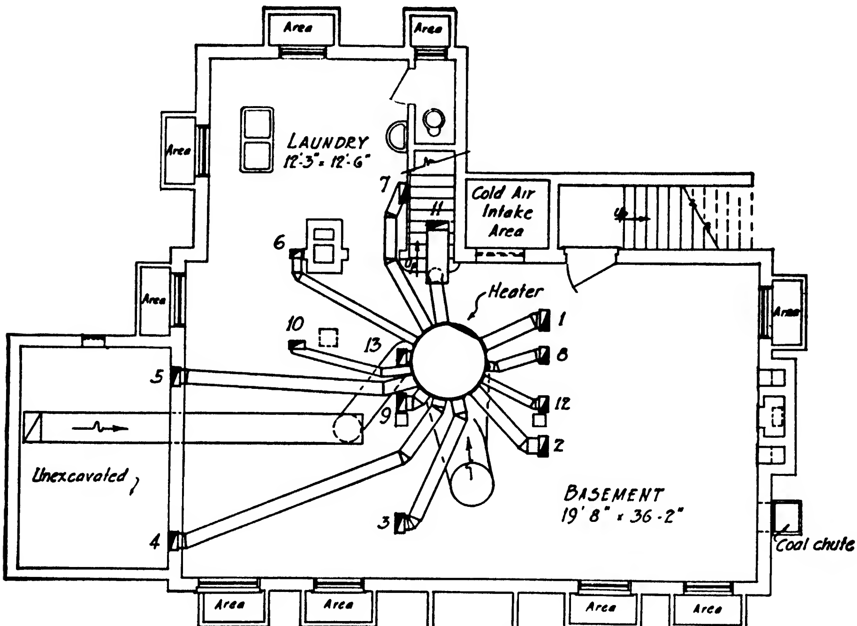

FIG. 8. BASEMENT PLAN, RESEARCH RESIDENCE

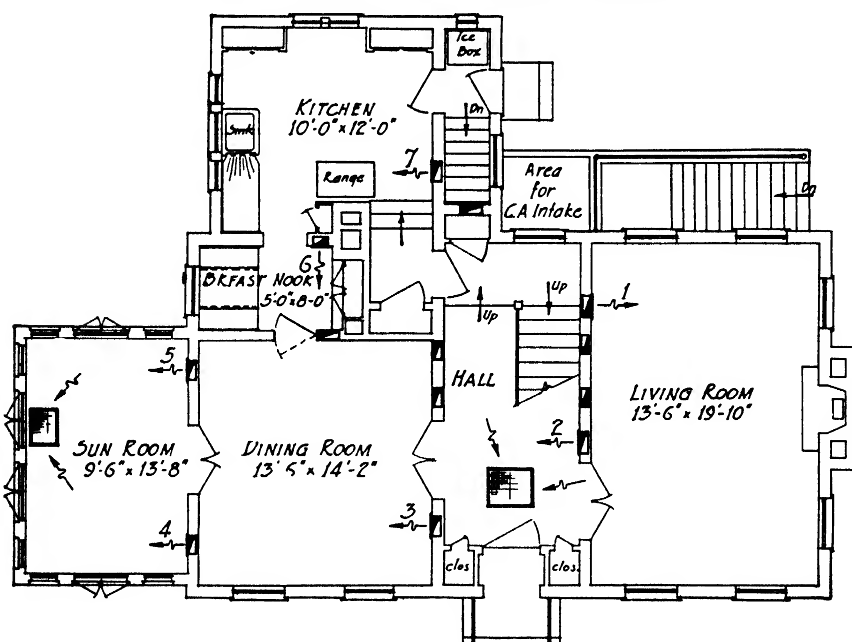

FIG. 9. FIRST-FLOOR PLAN, RESEARCH RESIDENCE

where

G = grate area, square inch.
p = combustion rate, pound coal per square foot of grate per hour.
f = heating value of the coal, Btu per pound.
E_1 = efficiency at bonnet, ratio of heat delivered at bonnet to heat developed in furnace.
E_2 = efficiency of duct transmission, ratio of heat delivered at register to heat delivered at bonnet.
0.866 = factor of safety to allow for contingencies under service conditions such as accumulations of soot and ashes, ineffective firing methods, etc.
H = total heat loss from structure.

An addition of 2 per cent of the furnace capacity is proposed for each unit that that ratio of heating surface to grate area exceeds 20. This addition is based on tests[4] conducted at the University of Illinois on seven types of furnaces having varying ratios of heating surface to grate area. This correction does not, however, apply to values of the ratio less than 15 nor greater than 30.

By transposing the terms in Equation 7 and adding the correction term for ratios of heating surface to grate area other than 20 to 1, the following equation is obtained:

$$G = \frac{144 \times H}{p \times f \times E_1 \times E_2 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \tag{8}$$

in which R = ratio of heating surface to grate area.

In the case of the Standard Code[5] the numerical values used in Equation 8 were based on those determined from the tests conducted on the different types of furnaces.

$$G = \frac{144 \times H}{7.5 \times 12{,}790 \times 0.55 \times 0.75 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \tag{9}$$

$$G = 0.004205\ \frac{H}{[1 + 0.02\ (R\text{-}20)]} \tag{10}$$

As used in these calculations, H = Btu heat loss from the entire house per hour = summation of all room losses $H_1 + H_2$ + etc. + the Btu necessary to heat the outside air, if any, at intake. This outside air loss in Btu per hour will be approximately 1.27 times the cubic feet of air admitted through the intake per hour on a zero day. For systems which recirculate all the air this value will be zero. For systems which have an outside air intake, controlled by damper, this value might well be approximated, since this loss will probably be reduced to a minimum on a zero day. Assume for such cases that the building loss is increased by 25 per cent, and that there is the usual 25 per cent loss between furnace and registers.

TYPICAL DESIGN

The application of the preceding data to an actual example may be of assistance to the designer. Figs. 8, 9, 10 and 11 represent the plans of

[4]University of Illinois, *Engineering Experiment Station Bulletin* No. 246, by A. C. Willard, A. P. Kratz, and S. Konzo, Chapter X, pp. 126-146

[5]Loc. Cit. Note 3

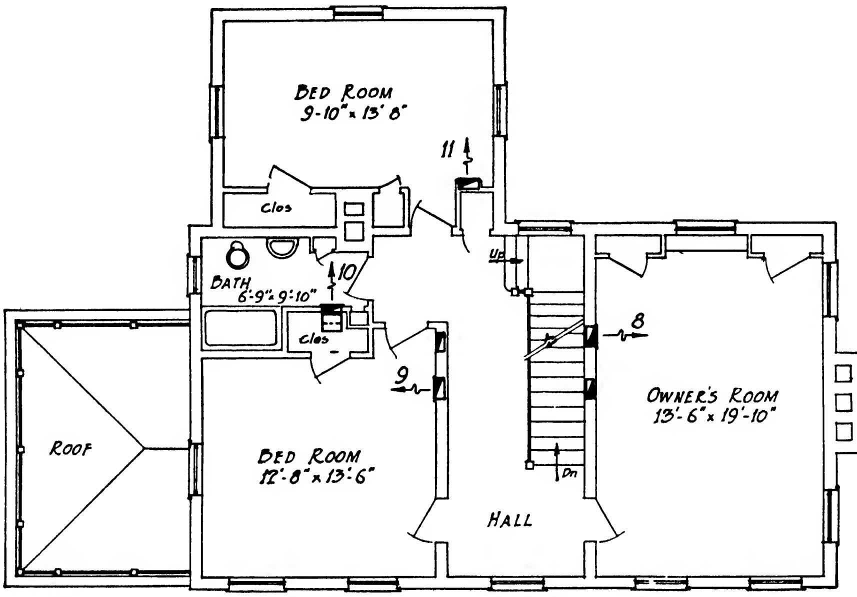

FIG. 10. SECOND-FLOOR PLAN, RESEARCH RESIDENCE

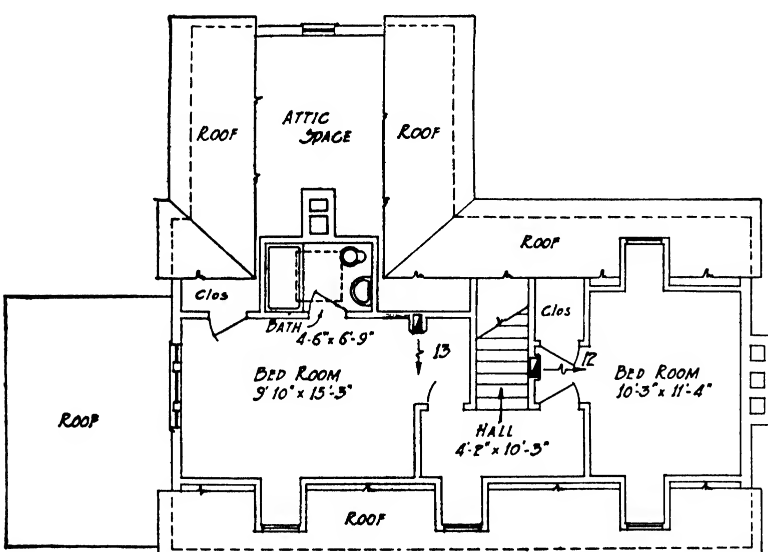

FIG. 11. THIRD-FLOOR PLAN, RESEARCH RESIDENCE

the Warm Air Research Residence of the *National Warm Air Heating and Air Conditioning Association* erected at the University of Illinois[6].

Leaders, Stacks and Registers. (Direct Method)

Living Room, 1st floor:

17,250 ÷ 111 = 155 sq in. leader area. See summary, Table 1; also example under Standard Code[7], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 14 in.

Register size = 155 sq in. net area. Gross area = net area ÷ 0.7 = 14 in. × 16 in.

Owner's Room, 2nd floor:

15,030 ÷ 167 = 90 sq in. leader area. See summary Table 1; also example under Standard Code[7], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 11.4, say 12 in.

Stack area = 0.7 × 90 = 63 sq in. = say 5 in. × 12 in.

Register area = 90 sq in. net area. Gross area = net area ÷ 0.7 = 12 × 12 or 12 in. × 14 in.

In like manner the leaders, stacks and registers are calculated for each room in the house.

Leaders, Stacks and Registers. (Code [7] Method. See Art. 3, Sec. 1, 2, 3)

Living Room (Glass = 90, Net wall = 405, Cubic contents = 2405)

$$\text{Leader} = \left(\frac{90}{12} + \frac{405}{60} + \frac{2405}{800} \right) 9 = 155 \text{ sq in.}$$

Register, same as Direct Method.

Owner's Room (Glass = 68, Net wall = 394, Cubic contents = 2275)

$$\text{Leader} = \left(\frac{68}{12} + \frac{394}{60} + \frac{2275}{800} \right) 6 = 90 \text{ sq in.}$$

Stack and Register, same as Direct Method.

Assuming all air recirculated, the minimum furnace for the plant will be:

Grate area = 0 0042 × 132,370 = 556 sq in.
Use 27 in. diameter grate. (Equation 10).

If provision should be made for certain outside air circulation, then increase the building heat loss by, say 25 per cent and obtain by Equation 10 a 30 in. grate.

Experiments at the University of Illinois[8] have shown that the capacity of a furnace may be increased nearly three times by an adequate fan, with a constant register or delivery temperature maintained, *provided that the rate of fuel consumption can be increased to provide the necessary heat.* In other words, the capacity of a forced circulation system *is limited by the ability of the chimney to produce a sufficient draft, and the ability of the fan to deliver an adequate amount of air.*

[6]Plans used with permission. Bathroom on third floor not heated.
[7]Loc. Cit. Note 3.
[8]University of Illinois, *Engineering Experiment Station Bulletin* No. 120, p 129.

TABLE 1. SUMMARY OF DATA APPLIED TO WARM AIR RESEARCH RESIDENCE

Rooms	From Chapter 7 Estimating Heat Losses Btu Heat Losses *H*	Leader Area Sq In	Stack Area Sq In. 0 7 × *LA*	Leader Diameter Inches	Stack Size Net	Register Size Gross
First Floor		= 0.009*H*				
Living	17250	155	....	14		14 × 16
Dining	6810	61	....	9		8 × 12
Breakfast	2300	21	....	8		8 × 10
Kitchen	9210	83	....	11 or 12		12 × 14
Sun	25710	230	....	Two 12		Two 12 × 14
Hall and stair	12570	113	....	12		12 × 14
Second Floor		= 0.006*H*				
Owner's	15030	90	63	11 or 12	5 × 12	12 × 14
S. W. Bed	9800	59	41	9	3½ × 12	8 × 12
Bath	2450	15	10	8	3 × 10	8 × 10
N. Bed	14800	89	62	11 or 12	5 × 12	12 × 14
Third Floor		= 0.005*H*				
E. Bed	8220	41	29	8	3 × 10	8 × 10
W. Bed	8220	41	29	8	3 × 10	8 × 10

BOOSTER FANS

Booster fans often may be arranged to operate when gas or oil burners are running and to stop automatically when the burners shut down. The booster equipment is most effective in increasing output at low operating temperatures. According to tests, efficiencies may be advanced from 60 per cent for gravity to 70 per cent with boosters at low operating temperatures, but at high operating temperatures gravity and booster efficiencies are almost identical[9].

[9]University of Illinois, *Engineering Experiment Station Bulletin* No 141, p 79, and No 246

PROBLEMS IN PRACTICE

1 ● What may prohibit the use of a gravity warm air system in a large house having several exposed wings?

In a gravity warm air system, excessive vertical distances above the furnace cause little trouble in the design of the wall stacks, but excessive horizontal distances from the furnace should be carefully considered in the design of the leaders. To work effectively, a gravity warm air system should be balanced and leaders over 12 ft in length should be avoided if possible. Long leaders, if used, must be of ample size, well pitched, and well insulated. Large houses having exposed wings may require leaders much longer than 12 ft; infiltration may create severe back-drafts in the exposed wings; and the basement ceiling height may not be sufficient to allow the leaders to have a pitch of more than one inch per foot. These conditions may make the exposed wings very difficult to heat with a gravity system because of its low air head differentials.

2 ● A first story dining room has a calculated heat loss of 12,000 Btu per hour.

a. What size leader pipe should be used for 175 F register air temperature?

b. What size register?

a. Leader area $= \frac{12{,}000}{111} = 108.1$ sq in. Use leader with diameter of 12 in.

b. Register gross area $= \frac{108}{0.7} = 154$ sq in Use 12 in by 14 in register

3 ● A third-story bedroom has a calculated heat loss of 12,000 Btu per hour.
a. What size leader pipe should be used for a 175 F register air temperature?
b. What size stack?
c. What size register?

a. Leader area $= \frac{12{,}000}{200} = 60$ sq in. Use leader with diameter of 9 in

b. Stack area $= 0.7 \times 60 = 42$ sq in. Use stack 3½ in. by 12 in

c. Register gross area $= \frac{60}{0.7} = 85.7$ sq in. Use register 8 in by 12 in

4 ● The calculated heat loss of a house is 130,000 Btu per hour. Find the grate area required for the furnace under the following conditions:

Heating value of coal = 12,790 Btu per pound.
Furnace efficiency = 55 per cent.
Combustion rate = 7.5 lb per sq ft per hr.
Ratio of heating surface to grate area of furnace = 20 to 1.
Register temperature = 175 F.
Loss between furnace and registers = 25 per cent.

See Equations 9 and 10
Grate area $= 0.004205 \times 130{,}000 = 547$ sq in
Grate diameter = 26.3 in.
Use grate with diameter of 26 in.

5 ● If in Question 4 the conditions were the same except that the ratio of heating surface to grate area of furnace was 24 to 1, what size grate would be required for the furnace?

Grate area $= \frac{0.004205 \times 130{,}000}{1 + 0.02\,(24\text{-}20)} = \frac{547}{1.08} = 506$ sq in
Grate diameter = 25.4 in.
Select grate with diameter of 25 in

6 ● Name the items involved in the design of a furnace heating system.

a. Heat loss from each room, Btu
b. Area and dimensions of warm-air pipes in basement, inches.
c. Area and dimensions of vertical pipes, inches.
d. Free and gross area and dimensions of warm-air registers, inches
e. Area and dimensions of recirculating or outside air ducts, inches
f. Free and gross area and dimensions of recirculating registers, inches.
g Size of furnace necessary to supply the warm air to overcome the heat loss
h Area and dimension of chimney and smoke pipe, inches.

7 ● Discuss the design features of recirculating ducts.

a. Their area should be equal to or greater than that of the supply ducts
b. They should be streamlined, and have a minimum number of turns.
c. All runs should be as short as possible
d Account should be taken of all cold walls and window areas in determining sizes and positions of return air inlets.

e. The return line should be pitched downward toward the furnace. It should be designed to minimize friction.

f. The top of the shoe or boot should never be above the grate level

8 ● Discuss the use of a booster fan. What effect has a booster fan at low operating temperatures? At high ones?

A booster fan is useful in accelerating the air flow past the surface of a low temperature furnace, where only a small weight differential in the air is created, and in unbalancing a gravity system so flow is established. The first use involves the entire plant, and increases efficiency about 10 per cent with low temperature operation; the second involves only the leaders in which air flow is accelerated. At high operating temperatures the difference in weight between warm outgoing air and cool incoming air is great enough to make a booster unnecessary with ordinary gravity systems.

9 ● Is it desirable to use high side wall locations for warm air registers in gravity circulating systems?

High side wall locations are not recommended on account of the tendency for stratification of the air in the room resulting in high temperatures at the ceiling.

Chapter 20

MECHANICAL WARM AIR FURNACE SYSTEMS

Furnaces, Fans and Motors, Sound Control, Sprays and Filters, Air Distribution Design, Automatic Controls, Design of Heating System, Selecting the Furnace, Selecting the Fan, Heavy Duty Fan Furnaces, Humidification, Cooling Methods, Cooling System Design

MECHANICAL warm air or fan furnace heating systems[1], which are a special type of central fan systems, are particularly adapted to residences, small office buildings, stores, banks, schools, and churches. Circulation of air is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering its bottom, as in gravity systems described in Chapter 19. The advantages of mechanical systems, as compared with gravity systems are:

1. The furnace can be installed in a corner of the basement, leaving more basement room available for other purposes.

2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view except in the furnace room.

3 Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by sprays or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

Much of the equipment used in central fan systems is the subject matter of other chapters. It is the purpose of this chapter to discuss the coordinated design and to deal in detail only with problems not covered elsewhere which refer particularly to the whole problem of fan warm air furnace heating and air conditioning.

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[1]See University of Illinois, *Engineering Experiment Station Bulletin* No 266 by A P Kratz and S Konzo for details of tests conducted in Warm Air Research Residence.

FURNACES

Furnaces for mechanical warm air systems may be made of cast-iron, steel, or alloy. Cast-iron furnaces are usually made in sections and must be assembled and cemented or bolted together on the job. Steel furnaces are made with welded or riveted seams. The proper design of the furnace depends largely on the kind of fuel to be burned. Accordingly, various manufacturers are making special units for coal, oil and gas. Each type of fuel requires a distinct type of furnace for highest efficiency and economy, substantially as follows:

1. Coal Burning:
   a. *Bituminous*—Large combustion space with easily accessible secondary radiator or flue travel.
   b *Anthracite or coke*—Large fire box capacity and liberal secondary heating surfaces.
2. Oil Burning·
   a. Liberal combustion space.
   b. Long fire travel and extensive heating surface.
3. Gas Burning:
   a. Extensive heating surface.
   b. Close contact between flame and heating surface.

A combustion rate of from 5 to 8 lb of coal per square foot of grate per hour is recommended for residential heaters. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. It is recommended that the system be designed for blow-through installations, so that the furnace shall be under external pressure in order to minimize the possibility of leakage of the products of combustion into the air circulating system.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes it may run as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of any number of furnaces, using one or more fans.

Furnace Casings

Casings are usually constructed of galvanized iron, 26-gage or heavier, but they may also be constructed of brick. Galvanized iron casings should be lined with sheet iron liners, extending from the grate level to the top of the furnace and spaced from 1 in. to 1½ in. from the outer casing. Casings for commercial or heavy duty furnaces, if built of galvanized iron, should be insulated with fireproof insulating material at least 2 in. thick. It is generally believed that either brick or sheet metal casing should be equipped with baffles to secure impingement of the air to be heated against the heating surfaces. Brick furnace casings should be supplied with access doors for inspection.

For furnace casings sized for gravity flow of air, where a fan is to be

used, many manufacturers recommend the use of special baffles to restrict the free area within the casing and to force impingement of the air against the heating surfaces. The method of making these baffles for furnaces with top horseshoe radiators and for furnaces with back crescent radiators is illustrated in Fig. 1.

Either square or round casings may be used. Where square casings are

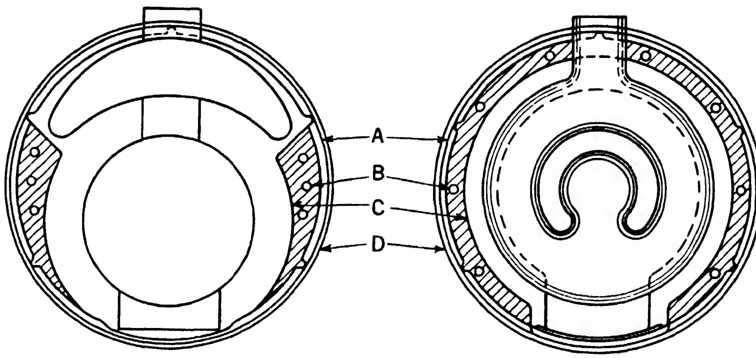

FIG. 1. USUAL METHOD OF BAFFLING ROUND CASINGS FOR FAN FURNACE WORK

A Liner, 1 in from casing *B*. Hole to vent baffle
C Baffle, closed top and bottom *D* Outer casing

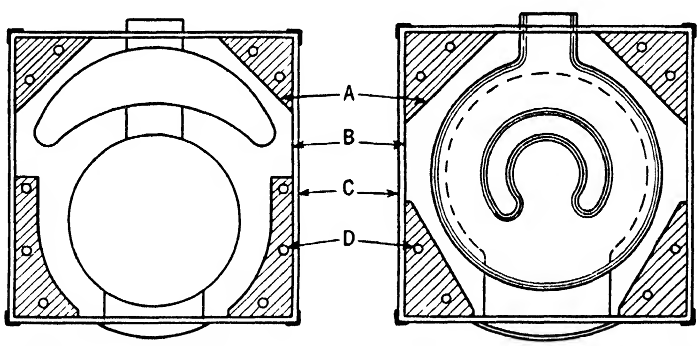

FIG 2. METHOD OF BAFFLING SQUARE FURNACE CASING FOR FAN FURNACE WORK

A Baffle, closed top and bottom *B* Liner 1 in from casing *C*. Outer casing *D* Hole to vent baffle

used, the corners must be baffled to reduce the net free area and to force impingement of air against the heating surfaces. Fig. 2 shows the usual method of baffling square furnace casings for fan furnace work.

The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet

and thus provides a larger plenum chamber. Fig. 3 illustrates a complete residence fan furnace installation showing location of fan, furnace, filters, plenum chamber and method of take-off of warm air pipe.

FANS AND MOTORS

Centrifugal type fans are most commonly used, and these may be equipped with either backward or forward curved blades. Motors may be mounted on the fan shaft or outside of the fan with belt connection. Multi-speed motors or pulleys are desirable to provide a factor of safety and to allow for increased air circulation. For additional information on fans and motors, see Chapters 27 and 38.

SOUND CONTROL

Special attention should be given to the problem of noise elimination. The fan housing should not be directly connected with metal, either to the furnace casing or to the return air piping. It is common practice to use canvas strips in making these connections. Motors and their mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with fan housing The installation of a fan directly under a cold air grille is not recommended on account of the noise objection. (See also Chapter 30).

FILTERS

There are many satisfactory types of filters on the market. These include dry filters, viscous filters, oil filters and other types, some of which must be cleaned, some of which must be cleaned and recharged with oil, and some of which are inexpensive and may be discarded when they become dirty, and replaced with new ones.

The resistance of a filter must be considered in the design of the system since the resistance rises rapidly as the filter becomes dirty, thus impairing the heating efficiency of the furnace, in fact, endangering the life of the furnace itself. Manufacturers' ratings of filters must be carefully regarded, and ample filter area must be provided. Filters must be replaced or cleaned when dirty. (See also Chapter 26).

AIR DISTRIBUTION

The conditions of comfort obtained in a room are greatly influenced by the type of register used and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. Due to the economic considerations involved, it is common practice to locate the supply openings on the inside walls of a residence and the return openings nearest the greatest outside exposure. Many designers prefer, however, to locate the supply registers so that the warm air from the registers *blankets* a cold wall, and mixes with the cold air dropping off from the exposed walls. This may be accomplished by the use of a supply register placed close to an outside wall in such a position that the warm air sweeps

the cold wall surface. The ducts leading to supply registers which are located on exposed walls should be adequately insulated to reduce the heat loss from the ducts.

Register and Grille Openings

Supply registers located in the floor are effective, but as they require frequent attention to keep them clean they should be avoided where another effective register location can be found. Tests conducted in the Warm Air Research Residence[2] have indicated that excellent results are obtainable with the use of a deflecting-diffuser type of baseboard register which throws the air downward toward the floor and diffuses the air at the

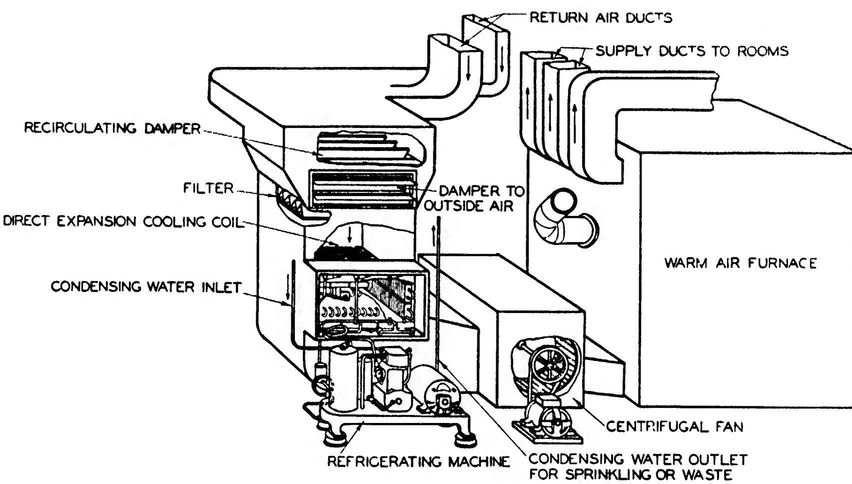

FIG. 3 COMPLETE RESIDENCE FAN FURNACE INSTALLATION FOR WINTER HEATING AND SUMMER COOLING

same time. Unless registers located in the baseboard are well proportioned and designed to harmonize with the trim, they may be unsightly. Better air distribution for cooling is obtained when high side wall registers are used, and this same location is satisfactory for heating when the openings are installed at least 7 ft above the floor line, providing the air velocity through the registers is greater than 600 fpm. Registers which are located in side walls above the baseboard or in the ceiling should be of an effective air-diffusing type. All registers should be equipped with dampers, and should be sealed against leakage around the borders or margins.

Velocities through registers may be reduced by the use of registers larger than the connecting pipes. Some suggestions for equalizing velocities over the face area of the register by means of diffusers are illustrated in Fig. 4. Merely to use a larger register may not result in materially reduced velocities unless diffusers are used.

[2]Loc Cit Note 1

Dampers

Suitable dampers are essential to any trunk or individual duct system, as it is virtually impossible to so lay out a system that it will be absolutely in balance without the use of dampers. Special care must be used in the design of any system to avoid turbulence and to minimize resistance. Sharp elbows, angles, and offsets should be avoided. (See Figs. 1 and 2, Chapter 29)

Three types of dampers are commonly used in trunk and individual duct systems. *Volume dampers* are used to completely cut off or reduce the flow through pipes. (See *A* and *B*, Fig. 5.) *Splitter dampers* are used where a branch is taken off from a main trunk. (See *C*, Fig. 5.) *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. (See *D*, Fig. 5.) It is essential that a damper be provided for each main or duct branch. A positive locking device should be used with each type of damper.

Ducts

The ducts may be either round or rectangular. Rectangular ducts should be as nearly square as possible; the width should not be greater

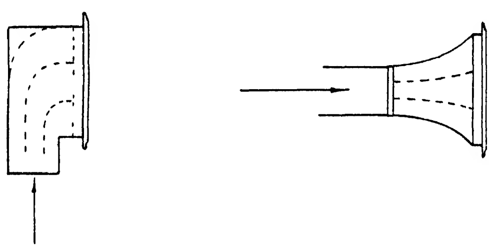

FIG 4. DIFFUSERS IN TRANSITION FITTINGS TO EQUALIZE VELOCITIES THROUGH REGISTER FACES

than four times the breadth. The radii of elbows should be not less than one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts.

AUTOMATIC CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in the system can be largely eliminated through proper care in the planning and installation of the control system.[3] The essential requirements of the control are:

1. To keep the fire burning when using solid fuel regardless of the weather.
2. To avoid excessive bonnet temperatures with resultant radiant heat losses into the basement.
3. To avoid the overheating of certain rooms through gravity action during off periods of blower operation.

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[3]Automatic Controls for Forced-Air Heating Systems, by S Konzo and A F Hubbard (A S H V E. TRANSACTIONS, Vol 40, 1934, p 37)

4. To have a sufficient supply of heat available at all times to avoid lag when the room thermostat calls for heat.
5. To prevent cold air delivery when heat supply is insufficient.
6. To avoid heat loss through the chimney by keeping stack temperatures low.
7. To provide quick response to the thermostat, with protection against overrun.
8. To provide for humidity control.
9. To provide a means of summer control of cooling.
10. To protect against fire hazards.

The following controls are desirable:

1. A *thermostat* located at a point where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. This location would be near an outside wall but not upon it, in a sun room, or in a room with some unusual exposure. The thermostat, of course, should not be located where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct or register.

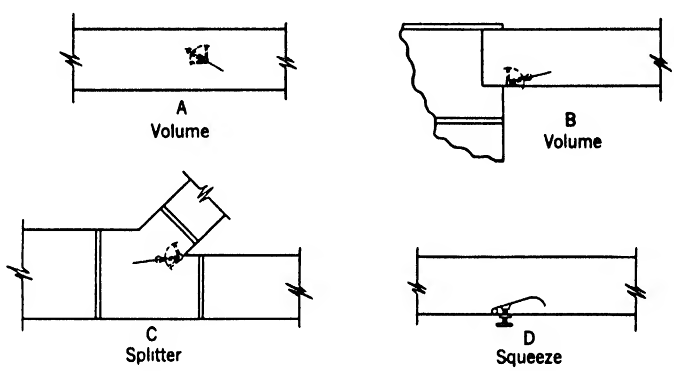

FIG. 5. THREE TYPES OF DAMPERS COMMONLY USED FOR TRUNK AND INDIVIDUAL DUCT SYSTEMS

2 A *furnacestat* located in the bonnet to permit blower operation only between the temperatures of 100 F and 150 F. In certain extreme cases it may be necessary, or weather conditions may make it advisable, to adjust the high limit to a higher temperature than that given. Another location sometimes used for the furnacestat is in the main duct near the frame opening from the bonnet.

3. A *protective limit control* located in the bonnet to shut down the system independently of the thermostat if the bonnet temperature exceeds 200 F.

4. On oil and gas burner installations, a control is usually included which will shut down the system if the fire goes out or if there is a failure of the ignition system.

5. A *humidistat* to regulate the moisture supplied to the rooms

6. On automatic stoker installations, a control is usually included which will start the operation regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying.

METHOD OF DESIGNING FORCED-AIR HEATING SYSTEMS

1. Determine heat loss from each room in Btu per hour. (See Chapter 7).
2. Locate warm air registers and return registers on plans of house, beginning with the upper story rooms.
3. Sketch in duct layout to connect all registers and grilles with the central unit.

4. Determine equivalent length of duct for each register, allowing 10 diameters of straight pipe as equivalent to each 90 deg elbow having an inner radius not less than the diameter of the round pipe or the depth of the rectangular pipe.

5. Select a value for temperature of the air at the furnace bonnet. It is customary to use some value lying between 150 to 165 F. Use lower value if larger number of air recirculations is desired. It is recommended that the number of air recirculations should be in excess of 5 per hour.

6. Determine approximate value of temperature reduction in each duct caused by heat loss from the ducts. A value of from 0 3 to 0 6 F per foot of duct has been obtained from tests conducted in the Research Residence installation for uninsulated duct lengths up to approximately 60 ft.

7. Subtract this temperature reduction from the assumed bonnet air temperature to obtain an approximate value of the register air temperature for each register.

8. Determine the required air volume for each room from the following equation, or from the values listed in Table 1:

$$Q = \frac{H}{60 \times 0\ 24 \times d\ (t_r - 65)} \tag{1}$$

where

Q = required air volume, cubic feet per minute
H = heat loss of room, Btu per hour.
d = density of air at register temperature, pounds per cubic foot
t_r = register temperature, degrees Fahrenheit
0.24 = specific heat of air
65 = return air temperature.

For any given register temperature the solution of this equation simplifies to the following form

$$Q = H \times \text{Factor} \tag{2}$$

in which the values of the Factor may be obtained from Table 1

9. Determine register size from the air volume delivered to each room by the following formula

$$\text{Free area of register, square feet} = \frac{Q}{V} \tag{3}$$

$$\text{Gross area of register, square feet} = \frac{\text{Free Area}}{R} \tag{4}$$

where

Q = required air volume, cubic feet per minute
V = velocity at register face, feet per minute.
R = ratio of free area to gross area of register.

TABLE 1. FACTORS CORRESPONDING TO REGISTER TEMPERATURE FOR EQUATION 2

REGISTER TEMPERATURE	FACTOR
110	0.0221
120	0.0184
130	0.0158
140	0.0140
150	0.0125
160	0.0114
170	0.0105

Allowable register velocities to be used in Equation 3 are approximately as follows:

Baseboard, non-deflecting type, maximum = 300 fpm.
Baseboard, deflecting toward floor, maximum = 500 fpm.
Baseboard, deflecting and diffusing = up to 800 fpm.
High sidewall = not less than 600 fpm.

10. Duct systems for forced-air installations may consist of either trunk systems or individual duct systems.

Trunk Systems. Determine duct sizes and friction losses as outlined in Chapter 29, except that for residence applications the velocities in the main duct and in the various parts of the system should approximate the values recommended in Table 2.

Individual Duct Systems. An individual duct system is one having separate ducts extending from the heating unit to each register. In designing such a system select first the duct having the greatest equivalent length. Select a reasonable velocity using Table 2 as a guide. From friction chart on page 584 determine unit friction loss per 100 ft of run, and from this the total friction loss in the duct selected. If this total friction loss exceeds a reasonable value a lower velocity should be used

The remaining ducts are proportioned so that the total pressure in each duct is the same as that calculated for the longest duct. The added resistance necessary in the shorter ducts is accomplished by increasing the velocity in these ducts. No duct should be less than 6 in in diameter, nor should the velocity in any duct exceed approximately 1200 fpm. The final adjustment in a duct system may be made by employing dampers

TABLE 2. RECOMMENDED VELOCITIES THROUGH DUCTS AND REGISTERS

DESCRIPTION	LOW VELOCITY SYSTEM (FPM)	MEDIUM VELOCITY SYSTEM (FPM)	HIGH VELOCITY SYSTEM (FPM)
Main ducts	500	750	1000
Branch ducts	450	600	750
Wall stacks	350	500	600
Baseboard registers (max)	300	350	400
Wall registers above 5 ft (min.)	500	550	600

Instead of proportioning the ducts as outlined in the preceding paragraph it is more usual in practice to proportion all the ducts so that they have the same velocity as that used in the longest duct and to balance the system by employing dampers in the shorter ducts.

Return duct systems are designed making use of the same principles as those used in the design of supply duct systems. In this case the design may be based on the volume of air corresponding to the density of air existing in the return ducts, or in order to provide a factor for air leakage, it may be based on the same volume as used for the supply ducts.

11. Determine frictional resistance in:

a. Supply side of system as outlined in item 10.
b Return side of system as outlined in item 10.
c. Furnace units, casing or hood, which is usually considered as equivalent to 0 03 to 0 10 in. of water.
d. Accessories such as washers or air filters, from manufacturer's data.
e. Inlet and outlet registers and grilles, from manufacturer's data.
f. Other accessory equipment such as cooling coils, from manufacturer's data

Choose a fan which, according to its manufacturer's rating, is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the items listed in the preceding discussion. In practice it is recommended that liberal allowances should be made so that the

fan will be capable of delivering air against pressures that may not have been foreseen during the design of the duct system.

12. Select a furnace capable of delivering heat at the register outlets equal to the total heat loss of the structure to be heated.

The following formula may be used for coal burning furnaces:

$$G = \frac{H}{f \times p \times E_1 \times E_2\,[1 + 0.02\,(R - 20)]} \tag{5}$$

where

G = required grate area, square feet.
H = total heat loss from building, Btu per hour.
f = calorific value of coal, Btu per pound.
p = combustion rate in pounds of fuel per square foot of grate per hour
E_1 = furnace efficiency based on heat available at bonnet.
E_2 = efficiency of transmission based on ratio of heat delivered at register to heat available at bonnet.
R = ratio of heating surface to grate area.

In practice it is customary to use the following constants:

f = 12,000 (for specific values, see Table 5, Chapter 9).
p = 7.5 lb.
E_1 = 0.65 lower efficiency must be used with highly volatile solid fuel.
E_2 = 0.85.

The foregoing procedure for determining the size of the furnace to be used applies to continuously heated buildings.

13. Although intermittently heated buildings usually have their heat losses computed according to the standard rules for determining such losses, these rules do not take into account the heat which will be absorbed by the cold material of the building after the air is raised in temperature. This heat absorption must be added to the normal heat loss of the building to determine the load which the heating plant must carry through the warming-up process. It is customary to increase the normal heat loss figure by from 50 to 150 per cent depending upon the heat capacity of the construction material, the higher percentage applying to materials of high heat capacity such as concrete and brick Fan furnace systems are well adapted for heating intermittently heated buildings as these systems do not require the warming of intermediate piping, radiators, or convectors, the generation of steam, or the heating of hot water.

14. Follow the same methods for an oil furnace as for coal where a conversion unit is to be used, making sure that the ratio of heating surface to grate area exceeds 20 to 1. If it does not, a size larger furnace should be selected. Use the manufacturer's Btu ratings of furnaces designed for exclusive use with oil, and select a burner with liberal excess capacity.

15. The selection of the proper size gas furnace for a constantly heated building can be easily made by using the following *American Gas Association* formula·

$$R = \frac{H}{0\,9} \tag{6}$$

where

H = total heat loss from building, Btu per hour.
R = official *A.G A*. output rating of the furnace, Btu per hour.

In the case of converted warm air furnaces a slightly different procedure is necessary, as the Btu input to the conversion burner must be selected rather than the furnace output. The proper sizing may be done by means of the following formula:

$$I = 1.59\,H \tag{7}$$

where

I = Btu per hour input.

The factor 1.59 is the multiplier necessary to care for a 10 per cent heat loss in the distributing ducts and an efficiency of 70 per cent in the conversion burner.

16. Specify location and type of all dampers in both supply air and return air sides of system. Specify controls including location of all thermostats. Arrange for proper control of humidifying equipment.

HEAVY DUTY FAN FURNACES

Fan furnaces for large commercial and industrial buildings are available in sizes ranging from 400,000 to 3,000,000 Btu per hour per unit. Heavy duty heaters may be arranged in combinations of one or more units in a battery. A few possible arrangements are shown in Figs. 6 to 9 inclusive.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area on furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 21. Ducts are designed by the method outlined in Chapter 29.

HUMIDIFICATION

Mechanical warm air systems offer a means of proportioning and distributing moisture-bearing air; consequently, during the winter months humidifiers may be employed to deliver water vapor to the fan-driven air stream in proper amounts to produce a more humid atmosphere, with increased comfort for people and increased life for household furnishings. Temperatures and relative humidities should be governed within the limits of the generally accepted standards. See Chapters 3 and 25 for more detailed information on this point.

In earlier types of furnaces, water evaporating pans were usually placed in the cool portions of the air stream, but modern types usually locate them in air which has been heated by contact with the heating surfaces. To change water into vapor capable of being carried in an air stream as part of the mixture, about 1000 Btu per pound are required. Without the addition of this heat, termed the latent heat of evaporation, water injected into the air will be carried along in the form of tiny globules until it falls out of the stream or is deposited upon some surface. Furthermore, when dry air is in contact with water for a sufficient length of time without the presence of a sizable body of water or a source other than air from

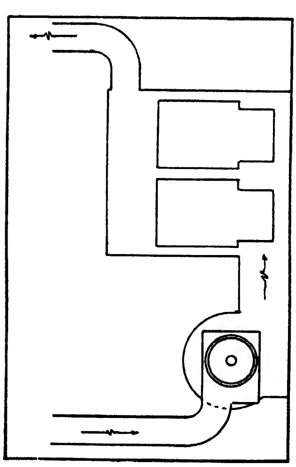

FIG. 6. HEATER ARRANGED FOR COMPLETE RECIRCULATION OF AIR

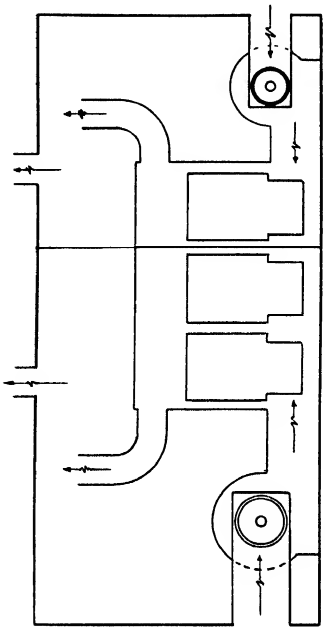

FIG. 7. TWO BATTERIES OF HEATERS AND FANS FOR INDEPENDENT SERVICE USING OUTSIDE AIR AND EXHAUSTING TO ATMOSPHERE

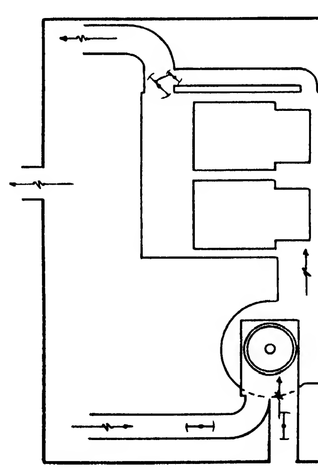

FIG. 8. HEATER ARRANGED FOR PARTIAL RECIRCULATION, ALSO SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR CHAMBERS, AND PARTIAL EXHAUST TO ATMOSPHERE

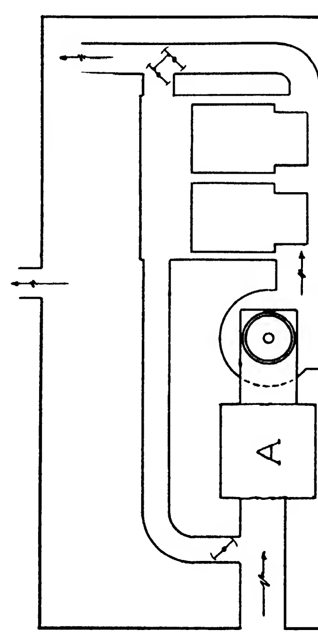

FIG 9. HEATER ARRANGED FOR USE OF AIR WASHER OR FILTER (A) WITH HEATED AIR TO MIX WITH OUTSIDE AIR FOR TEMPERING, SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR AND EXHAUST TO ATMOSPHERE

which this latent heat of evaporation can be taken, such heat is supplied from the air. There is, therefore, a trend in present practice toward heating the water in addition to heating the air. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. (See Chapter 25.)

Sprays for residence systems may be provided in separate housings to be installed on the inlet or outlet side of the fan, or they may be integral with the fan construction. They operate at water pressures of from 10 to 30 lb and use two or more spray nozzles for washing and humidification. The sprays should be adjusted to completely cover the air passages.

Sprays are usually controlled by solenoid valves wired in parallel with the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Sprays used in connection with commercial or heavy duty plants should be a regulation type of commercial spray.

Residence Requirements

The principles underlying humidity requirements and limitations for residences are summarized in *University of Illinois Bulletin* No. 230[4], as follows:

1 Optimum comfort is the most tangible criterion for determining the air conditions within a residence

2 An effective temperature of 65 deg[5] represents the optimum comfort for the majority of people Under the conditions in the average residence a dry-bulb temperature of 69 5 F with relative humidity of 40 per cent is the most practical for the attainment of 65-deg effective temperature.

3 Evaporation requirements to maintain a relative humidity of 40 per cent in zero weather depend on the amount of air inleakage to the average residence, and vary from practically nothing to 24 gal of water per 24 hours.

4. Relative humidity of 40 per cent indoors cannot be maintained in rigorous climates without excessive condensation on the windows unless tight-fitting storm sash or the equivalent is installed.

5. The problems of humidity requirements and limitations cannot be separated from considerations of good building construction, and the latter should receive serious attention in the installation of humidifying apparatus

The following conclusions were drawn from the experimental results reported in the aforementioned bulletin:

1 None of the types of gravity warm air furnace water pans tested proved adequate to evaporate sufficient water to maintain 40 per cent relative humidity in the Research Residence except only in moderately cold weather.

2. The water pans used in the radiator shields tested did not prove adequate to maintain 40 per cent relative humidity in a residence similar to the Research Residence when the outdoor temperature approximated zero degrees Fahrenheit.

[4]See Humidification for Residences, by A P. Kratz (*University of Illinois, Bulletin* No. 230).

[5]66 deg is the optimum winter effective temperature recommended by the A S H V E Committee on Ventilation Standards

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the use of basement air. A more positive cooling effect may be obtained through air washers where the temperature of the water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the temperature of the leaving water is below the dew-point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew-point. Ample coil surface is necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with the fan and duct system for the heating installation, and to cool the building by this method, provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. (See also Chapters 21 and 23).

Results at Research Residence

The following conclusions may be drawn from the studies thus far completed in the Research Residence, subject to the limitations of the conditions under which the tests were run[6].

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hr on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10 year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.

8. In the selection of cooling coils, the frictional resistance of the coil to flow of air must be given careful consideration.

9. Cooling the structure by introducing large quantities of air from outdoors at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

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[6]A S H V E RESEARCH REPORT No 947—Study of Summer Cooling in the Research Residence at the University of Illinois, by A P Kratz and S Konzo (A S H V E TRANSACTIONS, Vol 39, 1933, p 95); A S H V.E RESEARCH REPORT No 979—Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P Kratz and S. Konzo (A S H V E TRANSACTIONS, Vol 40, 1934, p. 167).

METHOD OF DESIGNING COOLING SYSTEM

The general procedure for the design of a cooling system in a forced-air installation is as follows:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 5 and 8). Allowance for addition of outside air must be included in this calculation.

2. Select a temperature of air leaving supply inlets In Research Residence tests[7] a value of from 65 to 70 F was found satisfactory.

3 Determine indoor conditions to be maintained In Research Residence 80 F dry-bulb and 45 per cent relative humidity was found satisfactory.

4. Determine the quantity of air to be introduced into each room. (See Chapter 21).

5 Estimate heat loss in duct system between cooling unit and supply registers.

6. Calculate the heat to be removed by the cooling unit, in the form of sensible heat and latent heat.

7 Determine size of ducts in duct system and size of registers, as explained in this chapter under the heading of Method of Designing Forced-Air Heating Systems.

8. Determine pressure loss in duct system and select fan as also explained in the same section.

9. Select cooling unit from manufacturer's data Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing See Chapter 24.

11 If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation

[7]Loc Cit Note 6

PROBLEMS IN PRACTICE

1 ● A residence furnace, having a ratio of heating surface to grate area equal to 20 to 1, is to be selected to heat a house which has a computed load of 225,000 Btu per hour. If coal having a calorific value of 12,000 Btu per pound is to be burned, if the furnace will burn 7.5 lb of coal per square foot of grate per hour, and if the furnace efficiency is 65 per cent, determine the square feet of grate area necessary in the furnace to be selected.

Substituting in Equation 5:

$$G = \frac{225,000}{12,000 \times 7.5 \times 0\,65 \times 0\,85} = 4.53 \text{ sq ft of grate area.}$$

A furnace having at least 4.5 sq ft of grate area should therefore be selected

2 ● Why should secondary surface be designed for easy cleaning?

If the combustion is not perfect, soot is formed immediately above the fire and is apt to form a deposit on the secondary surface from which it should be removed. If the secondary surface is so designed that there are horizontal passages, fine gray ash will settle out in these to form an insulation between the hot gases of combustion and the metal of the furnace; consequently, these should be readily cleaned. If the passages are vertical they are largely self-cleaning of ash, but provision should be made for easy and thorough cleaning of the collection chamber below them.

3 ● Why is baffling inside the casing necessary on fan systems?

Because the movement of air is independent of its temperature, air must be guided by baffles of one form or another to bring it in contact with the hot surfaces so it will not pass through the casing unheated. On the other hand, if the air is held against a hot surface too long it might become overheated, for the average register temperature on a fan system should not exceed 120 F.

4 ● What practical points should be observed in designing a fan system in order to eliminate noise?

a. Use a large fan so it can be run at slow speed.

b. Set the fan and motor on a solid foundation.

c. Insulate the fan and motor from the foundation with rubber, cork, or other springy material according to the principles given in Chapter 30, provided, of course, that such insulation is of value.

d. See that the air velocity is not too high in the ducts Properly designed splitters in the elbows will avoid high velocities at the turns in cases where the velocity through the ducts themselves is not too high.

e. Use canvas connections between the ducts and any running equipment.

f. Be sure the ducts have a relatively smooth interior and are rigid.

5 ● Why do furnaces designed to burn bituminous coal, oil, or gas require larger combustion spaces than those designed for anthracite?

Anthracite burns largely as fixed carbon whereas gas and oil burn as gases, and as much as 50 per cent of bituminous coal burns as a gas. Ample space must be provided for the intimate mixture of these gases with the oxygen of the air to secure proper combustion.

6 ● A furnace has the following dimensions: Grate diameter, 24 in.; casing diameter for gravity air flow, 56 in.; combustion chamber diameter, 30 in. What is the unobstructed area required for passage of air across the heating surface when a motor-driven blower, operating at an outlet velocity of 1200 fpm, delivers 1600 cfm into the casing near its bottom?

For residence applications using small blowers, an air outlet velocity of about one-third of the blower outlet velocity is considered good practice.

$$\text{Air-pass velocity} = \frac{1200}{3} = 400 \text{ fpm}$$

$$\text{Air-pass area} = \frac{1600}{4} = 4 \text{ sq ft} = 576 \text{ sq in.}$$

7 ● In Question 6 what would be the gap between the chamber and the baffle when the chamber is centered in the casing?

Area of combustion chamber (30-in diam)	706 9 sq in
Area of air pass	576 0 sq in
Total area	1282.9 sq in.

The diameter of a circle with an area of 1282.9 sq in. is 40.4 in One-half of the difference between the diameters is the amount of gap.

$$\text{Gap} = \frac{40.4 - 30.0}{2} = 5.2 \text{ in} = \text{approximately } 5\tfrac{1}{4} \text{ in.}$$

Chapter 21

CENTRAL SYSTEMS FOR COMFORT AIR CONDITIONING

Types of Systems for Heating, Humidifying, Cooling, Dehumidifying with Modifications, Design Details, Load Calculations, Selection of Equipment

A CENTRAL system for comfort air conditioning includes a central fan with complete supply and recirculating duct system which is designed to serve one or more conditioned spaces from one central apparatus. The air conditioning equipment includes the heating and cooling coils, humidifying and dehumidifying apparatus, air cleaning equipment, and the controls, all designed so that the air introduced to the space will maintain the desired temperature and humidity in that space *after* it has been affected by the losses and gains in heat or moisture which occur in the space. This chapter is intended to serve as a general guide in the selection and design of the central system for year-round air conditioning for commercial installations. References are made to other chapters which give complete data and discussion on certain portions of the work. Many systems will not include all of the functions of the complete year-round system and it is impractical in the space allowed here to show all the different combinations of equipment which could be used for central system work to suit special conditions or the preferences of the designing engineer.

In January, 1938 the Society adopted an application Code[1] for the design of comfort air conditioning installations which was prepared by a Joint Committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers.* In some states and municipalities local codes have been prepared giving specific design requirements for those localities and the engineer should refer to these standards if they are in existence.

CLASSIFICATION OF SYSTEMS

Central systems which are designed to produce dehumidification by the use of cold surfaces or sprays may be classified as:

1. Central unit with heating and cooling coils and humidifying sprays (refer to Fig 1) This system is very common and used extensively for summer and winter conditioning.

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[1]Code of Minimum Requirements for Comfort Air Conditioning (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1938, p 276) Reprints of this code are available at $ 10 a copy

2. Central unit with preheating coil, washer and reheating coil (refer to Fig. 2). This system is widely used for winter conditioning on larger installations and summer conditioning is accomplished by using cold water in the washer.

3. Blow through system with heating and cooling coils and mixing dampers (refer to Fig. 3). This system is used where several different zones are served from one central system and the conditioning of the entering air for each zone is determined by mixing varying quantities of air after passing through the conditioner.

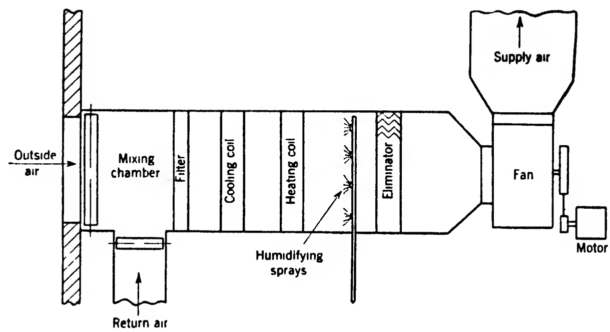

FIG 1 CENTRAL UNIT WITH COILS AND SPRAYS

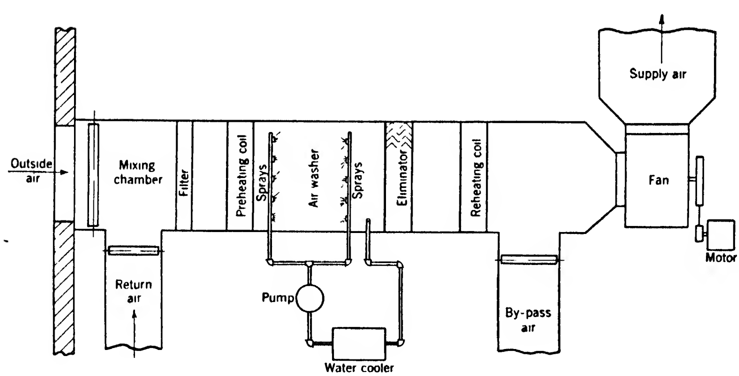

FIG 2. CENTRAL UNIT WITH WASHER AND PREHEATING AND REHEATING COILS

4 Central outside air conditioning units with unit recirculating conditioners (refer to Fig 4) This system is used on installations having large zones requiring independent control to meet local requirements for office and apartment buildings, hotels, etc.

5. Central conditioning unit either washer or coils with booster reheating coils (refer to Fig 5). This system is used for special zone arrangements where there is considerable variation in requirements of different zones such as hospital operating rooms or an auditorium served in conjunction with office and store cooling systems.

An entirely different method of dehumidification is used in some installations where the extraction of moisture is accomplished by methods of adsorption or absorption. Materials which are commonly used for these

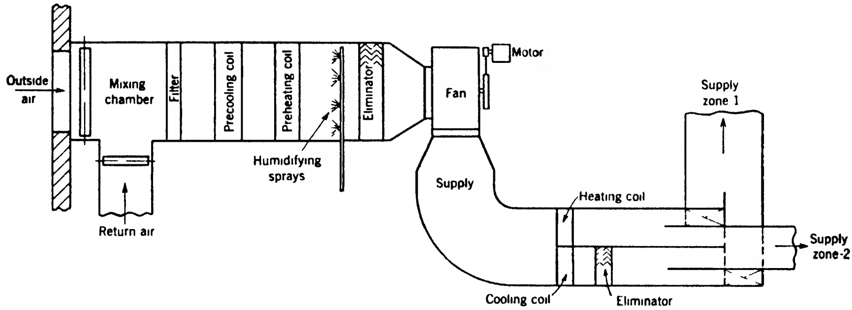

FIG. 3 BLOW THROUGH SYSTEMS WITH COILS AND MIXING DAMPERS

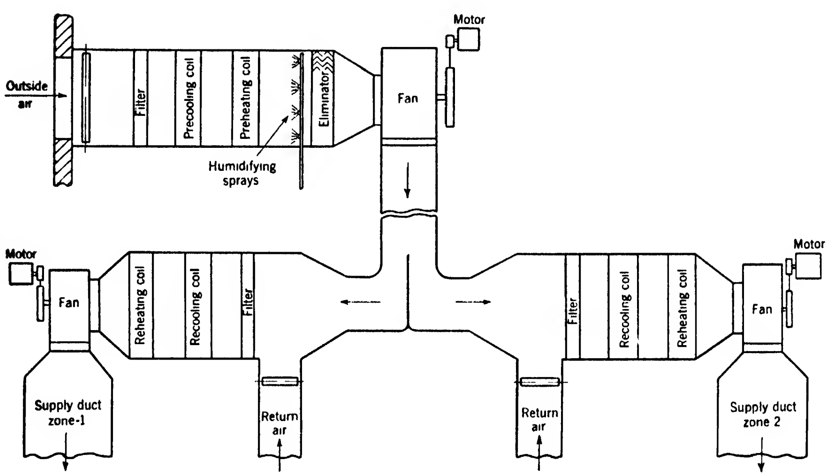

FIG. 4. CENTRAL UNIT WITH RECIRCULATING CONDITIONERS

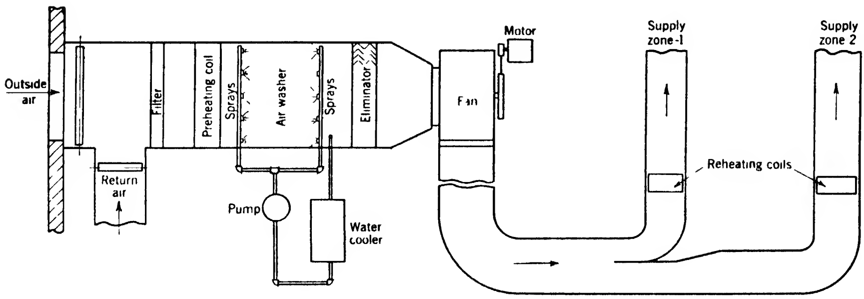

FIG 5. CENTRAL UNIT WITH EITHER AIR WASHER OR COILS AND BOOSTER COILS

systems are silica gel, lithium chloride, activated alumina and calcium chloride. These systems accomplish dehumidifying entirely separate from sensible heat cooling. The air passing through the material gives up its moisture to the material and there is a conversion of latent heat energy to sensible heat energy so that the temperature of the air leaving the equipment is considerably higher than the entering air. These systems can be used for dehumidifying outside air only, or for a mixture of outside and recirculated air, or for recirculated air only. The sensible heat cooling is accomplished by directing the air over a cooling coil. For further details of this method refer to Chapter 23.

DESIGN OF SYSTEM

In designing a complete air conditioning system it is logical to consider several factors which, for convenience, may be enumerated herewith:

1. Design conditions
    a. Outside dry-bulb temperature for winter.
    b. Outside dry- and wet-bulb temperatures for summer.
    c. Inside dry-bulb temperature and relative humidity.
2. Design heating load
    a. Heat transfer through windows, walls, partitions, doors, floors, sky-lights, ceilings and roofs.
    b. Heating capacity to warm incoming outside air.
    c. Heat required to evaporate moisture for humidification.
    d. Heat loss through ducts and casings.
    e. Allowances for heat emitting sources
3. Design cooling load
    a. Heat transfer through windows, walls, partitions, doors, floors, sky-lights, ceilings and roofs
    b. The sensible and latent heat gains from occupants
    c. Heat emission of electrical, chemical, gas, steam, hot water or other devices, apparatus or lights.
    d. Transfer of solar radiation through windows, walls, doors, sky-lights or roofs
    e. The sensible and latent heat to be removed from incoming outside air
    f. Heat gain through ducts, casings and fans between conditioning unit and enclosure.
4. Design of distribution system
    a. Establish the temperature of air leaving the supply inlets.
    b. Calculate the quantity of air to be circulated.
    c. Estimate the temperature rise or loss in the duct system
    d. Calculate the heating or cooling requirements of apparatus.
    e. Design the duct system including the air inlets and outlets.
    f. Consider noise control problems.
    g. Design the control system.
    h. Calculate total static pressure of the system.
    i. Select the air filtering equipment.
    j Select the fan, motor, drive and auxiliary equipment

Design Conditions

The design outside dry-bulb temperature in various localities for use in computing the winter heating requirements may be referred to in Chapter 7. In computing the summer cooling requirements for various cities recommended design outside dry- and wet-bulb temperatures may be

found in Chapter 8. Recommended inside temperatures and relative humidities for winter and summer conditions and for various types of buildings are given in Chapters 3 and 7.

Load Calculations for Heating

Complete tabular information is given in Chapter 5 for determining the heat loss through windows, walls, partitions, doors, floors, sky-lights, ceilings and roofs.

The heating capacity required for warming the outside air which enters by uncontrolled infiltration should be calculated according to the information given in Chapter 6. Mechanical ventilation systems are frequently designed which produce either positive or negative pressures within an enclosure that are greater or less than the outside prevailing wind pressure. Under such circumstances uncontrolled infiltration is over or under balanced by the supply or exhaust ventilation system. If the rate at which air is specified to be introduced to or removed from the enclosure by positive means exceeds the estimated infiltration rate it is common practice to use the greater rate in heating capacity calculations.

The heat required for warming the outside air introduced for ventilation purposes should be calculated according to the basic data given in Chapter 3, and the methods outlined in Chapter 6. The first requirement for any air conditioning system is to provide satisfactory ventilation and the design should err on the side of being liberal in the amount of outside air introduced for the purpose of maintaining good ventilation without objectionable odors.

Code[2] requirements state that the assumed rate at which air is to be positively introduced into the enclosure per occupant, when the contamination of air within the enclosures results entirely from respiratory process shall not be less than 10 cfm per stated number of occupants which is indicated as being the maximum number of people within the enclosure when the sum of the remaining loads are a maximum. The Code further states that the assumed ventilation rates shall not be less than 15 cfm per stated number of occupants in enclosures where smoking is customarily permitted, and that provision shall be made for air removal from the enclosure either by natural or mechanical means at not less than the assumed ventilation rate. For the purpose of the Code air quality or purity are assumed to be met if means are provided for the positive introduction of outside air in the amounts previously mentioned and for removal of 95 per cent by count of all dust particles over 10 microns in diameter from all air delivered to the enclosure.

The heat required to evaporate the necessary water required for winter humidification and superheat the resulting vapor in order to raise the moisture content of the outside air assumed to enter the enclosure by infiltration or positively introduced for ventilation should be calculated according to the information included in Chapter 1.

The heat loss through the ducts and casings between the condition unit and the treated space may be determined from data given in Chapter 39.

The heat gain due to lights and people or other heating sources in the conditioned space may be calculated as referred to later in this Chapter.

[2]Loc Cit Note 1

This heat gain is then used as a credit against the heat loss calculations. In general, however, the design for heating disregards these gains as in most cases these values are not a continuous or uniform source of heat and the heating system must be adequate to maintain the required temperature at all times including nights, Sundays and holidays, when the space is not in normal use.

The summation of these heat losses will give the total heat to be supplied by the air entering the conditioned space and the additional heat necessary to be added to the conditioning unit to provide the prescribed entering air temperature.

Load Calculations for Cooling

The heat gain through the windows, partitions, doors, floors, skylights, ceilings or roofs of the enclosure due to the air dry-bulb temperature difference assumed to exist between the air on the opposite sides of the construction may be determined from data given in Chapter 5.

The heat gain from occupants may be calculated from data given in Chapter 3, which gives the metabolic rate for people engaged in various activities. In addition charts are included which give a separation of the sensible and latent heat losses from the body which should be itemized separately in all calculations.

The heat emission from various appliances should be calculated according to the information and data given in Chapter 8, with special consideration being given to the division of latent and sensible heat requirements of the apparatus. Complete details may also be found in Chapter 8 for determining the heat gain resulting from electric lights and motors within the enclosure.

The transfer of solar radiation through windows, walls, doors, skylights or roofs of enclosures may be determined from the charts and tables in Chapter 8.

The heat to be removed from the outside air in cooling load calculations is determined in exactly the same manner as described previously under Load Calculations for Heating. The minimum ventilation requirements should be carefully considered.

The heat gain through ducts, casings and fans between the conditioning unit and the treated space may be determined from information in Chapter 39, and such calculations demonstrate the value of insulation in applications of this type.

Air Distribution System for Heating

The total heating load to be supplied by the central system is determined from the several components of the load listed under Item 2. The quantity, air motion, and temperature of the treated air and the method of introducing it to the conditioned space should be designed so as to limit the variation in dry-bulb temperature to 3 F or less at a 5 ft level throughout that portion of the enclosure which is normally frequented by persons. It is desirable to avoid air velocities exceeding 50 linear feet per minute in the occupied zone between the floor and the 5 ft level. When architectural or other construction requirements necessitate the location of a supply or return grille below the 5 ft level in an occupied

space, special consideration should be given to the air velocities in that region to avoid uncomfortable drafts.

It is desirable to use reasonably low temperature differences between the entering air and room conditions where possible. Air temperatures from 80 to 90 F will generally be satisfactory, although where the quantity of air to be circulated is kept at a minimum and where the arrangement of air inlets permits adequate mixing with the room air before reaching the breathing zone, higher temperatures from 100 to 120 F can be used. Having selected the desired temperature of the entering air, the quantity of air is determined as follows:

$$Q = \frac{H}{60d \times 0.24\,(t_y - t)} \tag{1}$$

where

Q = volume of air to be introduced, cubic feet per minute
H = sensible heat loss of space to be conditioned, Btu per hour
d = density of air, pounds per cubic foot
t_y = outlet temperature at the grille, degrees Fahrenheit.
t = design room temperature, degrees Fahrenheit.

If the air quantity calculated is excessive, it may be decreased by using a higher entering temperature. If the quantity is too small to provide adequate distribution and ventilation, it may be increased by using a lower entering temperature. Air motion has a cooling effect on the individual, and ordinarily the air quantity circulated should provide an overall air change in the conditioned space in not less than 5 min or more than 12 min. Best results are secured when the entering air temperature and method of distribution permit uniform mixture of air without excessive motion in the occupied zone.

The temperature drop between the heating element and the supply grille may be calculated according to the following formula

$$t_d = \frac{H_d}{60\,d \times 0.24 \times Q} \tag{2}$$

where

t_d = temperature drop, in degrees Fahrenheit.
H_d = heat loss in ducts, etc , Btu per hour.

The temperature drop t_d is added to the grille outlet temperature to determine the air temperature leaving conditioning unit heating coil. In general, duct losses will vary from 2 to 5 F and if greater than 5 F special consideration should be given to insulation.

The duct distribution system is designed using velocities as recommended in Chapter 29, and grille locations as discussed in Chapter 28 In most installations it is advisable in order to permit economical heating prior to occupancy to design the return duct system of sufficient area to convey 100 per cent of the air handled by the fan. Also in mild weather certain economies of operation may be affected by designing the outside air duct of sufficient area to convey approximately the total quantity of air handled by the fan and means should be provided for the escape of this air quantity. In every case, however, the outside air duct should be of sufficient area to permit the minimum ventilation requirements to be met.

Air Distribution System for Cooling

The total cooling and dehumidifying load to be supplied by the central system is determined from the several components of the design load listed under Item 3. The entering air temperature is determined by selecting the proper relationship between the quantity of air to be handled, the heat gain in the conditioned space, and the location of the air inlets. In cooling applications it is desirable that the difference between the temperature of air currents in the space frequented by occupants and the average temperature in such space, be not greater than 2 F for air velocities of 40 linear feet per minute and over and not greater than 3 F for velocities of less than 40 linear feet per minute.

There is a fairly wide range of permissible entering air temperatures With high velocity jets or diffusing nozzles, located at some distance from the occupied space, entering air temperatures may be as much as 30 F below room temperature. Where the air is introduced through supply inlets fairly close to the occupied zone the entering air should be within 10 to 15 F of a desired room temperature. The problem of preventing drafts in summer air conditioning is important as air, cooler than room air, tends to fall without diffusing and proper design must consider the relationship between temperature and diffusion to secure satisfactory results.

The quantity of air circulated for cooling may be determined from Equation 1 which was given previously for heating. However, in this case H is the total sensible heat gain of the space to be conditioned Having established the entering dry-bulb air temperature and quantity, the relative humidity is then determined. If there is no moisture gain in the room (i.e., no latent heat gain) the dew-point of the entering air will be the same as that of the room air, as all of the necessary air cooling will be used for removing the sensible heat. If there is a latent heat load to be absorbed in the room then a procedure as outlined herewith may be used for determining the entering air wet-bulb temperature:

Total all the latent heat gains in the room and convert them to equivalent grains of moisture. Divide the total grains of moisture by the number of pounds of air delivered to the room which will give the difference in weight of moisture between the entering air and room air conditions. Subtract this amount from the grains of moisture corresponding to the dew-point temperature in the room and refer it to psychrometric charts or tables to establish the required dew-point temperature of the entering air. The intersection of this new dew-point condition with the dry-bulb temperature line of the entering air at the supply inlet to the room will establish the entering wet-bulb temperature condition

The temperature rise from the air cooling equipment to the supply inlet may be determined from Equation 2. Obviously, the dry-bulb air temperature leaving the apparatus must be lower than the supply inlet temperature by the difference in temperature rise calculated. These duct gains will usually be from 1 to 3 F and if greater than 3 F special consideration should be given to the application of insulation.

With the dew-point of air leaving the air cooling apparatus as previously determined and the dry-bulb air temperature as established from the temperature rise in the duct system, a wet-bulb temperature of air leaving the apparatus may be determined from a psychrometric chart which also shows the total heat of the air. In some instances, a conventional air

washer or simple cooling coil will not produce the desired combination of dry-bulb and dew-point temperatures for the air leaving the apparatus. The relative amount of dehumidification can be increased by reducing the air velocity through the dehumidifier, and by reducing the temperature of the dehumidifier. In some cases, it is necessary to add a reheating coil in order to obtain the desired combination of dry-bulb and dew-point temperatures.

The design of a duct distribution system for cooling is accomplished in the same manner as that previously described for heating installations. For cooling spaces prior to occupancy, it is also desirable to design the return air ducts of sufficient area to convey 100 per cent of the air handled by the fan and the same recommendations with regard to the outside air duct as referred to in the heating design would be applicable for summer air conditioning.

CORRELATION OF SUMMER AND WINTER DESIGN

Frequently the quantity of air required for the central system in summer conditioning is considerably greater than the quantity required for winter conditioning. In practice, volume control should be provided using a speed regulator on the fan, or dampers, so that the air quantity may be changed for the cooling and heating cycles. Sometimes a recalculation using different entering air temperatures will permit using the same quantity of air all year round. There is no fixed rule or method for determining the most practical design for air quantity and the engineer should use discretion to a large extent in working out a balanced system.

When air is introduced above room temperature it tends to rise, while incoming air below room temperature tends to fall. It is therefore common to introduce air for heating only through baseboard or low air inlets; and air for cooling is introduced through high side-wall inlets or ceiling inlets. Uniform diffusion is the important part of the design, and successful installations have been made introducing cold air from low outlets and warm air from high outlets. An overall air change in the treated space of once in 5 to 8 min prevents stratification. In a system designed for all year-round operation, the overhead diffusing openings are used for supply air with very satisfactory results. Particular care is taken in locating the exhaust or recirculating grilles to prevent short-circuiting of the supply air directly to the exhaust. Special care is also taken to prevent cold down-drafts from outside windows sweeping across a room during the heating season. This subject is covered more thoroughly in Chapter 28.

SELECTION OF EQUIPMENT

The system to be used is selected to meet the requirements of the installation. There are numerous modifications which can be made to any of the systems mentioned. Recirculating air is used in both heating and cooling for the sake of economy. However, in many cases the system is designed for 100 per cent outside air with no recirculation and an exhaust system added to remove the air to complete the circuit. In planning any system, it should be remembered that there is always some exfiltration through door cracks, windows, and even through building

materials, so that it is not possible to recirculate or exhaust all of the air mechanically. If the conditioned space is practically air tight, 90 per cent of the air can be recirculated, but in ordinary construction it is better not to assume more than 80 per cent. On the same basis exhaust fans are sized for 75 to 80 per cent of the supply fan capacity, where recirculation is not used.

In some localities, the cooling is accomplished by evaporative cooling without any mechanical refrigeration or dehumidifying equipment. This is practical where the outside air dry-bulb is high and the relative humidity is low, as the washer temperature will tend to equalize at the outside wet-bulb temperature and will deliver saturated air at this temperature for cooling. There is no change in the wet-bulb temperature of the air going through the conditioner and therefore no change in the total heat, but the rapid circulation of the cold, moist air is often considered sufficient for comfort. These systems do not use recirculation.

In making the selection between spray and surface dehumidifiers, certain characteristics of each should be considered. A spray dehumidifier, (i.e., a dehumidifying air washer) will deliver practically saturated air, the temperature of which is determined by the temperature of the air washer water. The control in this case is the control of the water temperature and cooling effect can be accomplished by an external water cooler, a natural cold water supply, or coils directly installed in the washer. The washer system is used in the winter time for humidifying in the same way; that is, by controlling the water temperature, water can be evaporated into the air. This may require preheating the air before entering the washer, or the use of an external water heater, or steam coils directly in the washer. Air washers also have the ability to eliminate certain kinds of dirt and dissolved gases and some odors.

Surface coil dehumidifiers seldom deliver saturated air. Most comfort conditioning systems require that the air be delivered at low dew-point and relatively high temperatures. It is often simpler to produce this condition with a so-called dry coil than with an air washer. Surface coils can be used with cold water in the tubes or with direct-expansion refrigerant. When dehumidifying, the coil surface is wet due to the condensed moisture, and this water has some small cleaning effect and does absorb some gases and odors. These coils must be kept clean, however, as the water condensed is not sufficient to wash off the accumulations of dirt, and in the course of time a stale odor may be continuously added to the air stream from dirty coils.

The relationship between sensible and latent heat is given in Table 1 for typical classes of comfort conditioning and the required entering air condition. Unfortunately, there is yet no uniform practice in the statement of this ratio, and hence in Table 1 several ways of stating the load ratio are given. Examples of the solution of a typical problem of treating air to produce the room conditions of 80 F dry-bulb and 50 per cent relative humidity are presented in Table 2. In these examples it is assumed that as the air is discharged into the room and diffuses with the room air, it will absorb the sensible and latent heat in the ratio indicated and thus arrive at the designed condition. These are grille or outlet conditions which will not be the same as the condition leaving the dehumidifier.

The heating coils may be selected as indicated in Chapter 24. It is customary to use heating coils of convenient shape to fit the balance of the conditioning equipment, using 1, 2, or 3 rows of coils as needed.

On comfort cooling installations, the 4-row cooling coil is most widely used although some installations use 3-row coils and some 5 rows or more, depending upon sensible-latent ratio, air quantity and refrigerant temperature. With any given face velocity, the resistance to air flow increases with the number of rows of depth of coil. (See Chapter 24).

If a system using an air washer is selected, the washer is designed to fit the permissible space both as to width, height and length. Refer to Chapter 25.

If the surface coil system is used, winter humidifying may be accom-

TABLE 1. ROOM HEAT LOAD RATIOS FOR TYPICAL SUMMER COMFORT CONDITIONING

Room Heat Load Ratios[a]	Typical Classes of Room Service or Load				
	No Occupants or Sources of Vapor	Private Office or Residence	Restaurant or Crowded Office	Auditorium at Capacity or Crowded Restaurant	Ballroom at Capacity
$\frac{\text{Sensible Heat}}{\text{Total Heat}}$	1.00	0.90	0.80	0.70	0.60
$\frac{\text{Total Heat}}{\text{Sensible Heat}}$	1.00	1.11	1.25	1.43	1.67
$\frac{\text{Latent Heat}}{\text{Total Heat}}$	0	0.10	0.20	0.30	0.40
$\frac{\text{Total Heat}}{\text{Latent Heat}}$		10.00	5.00	3.33	2.50
$\frac{\text{Sensible Heat}}{\text{Latent Heat}}$		9.00	4.00	2.33	1.50
$\frac{\text{Latent Heat}}{\text{Sensible Heat}}$	0	0.11	0.25	0.43	0.67

[a]The overall heat load ratio for the dehumidifier will be different from the heat load ratio for the room. The extent of the difference will depend on the quantity and condition of the outside air used, upon the magnitude of the duct losses, and upon whether or not reheat or by-pass are used.

TABLE 2. DRY-BULB TEMPERATURE OF AIR AT ROOM INLETS

To Maintain Typical Room Conditions of 80 F Dry-bulb, 50 per cent Relative Humidity

$\frac{\text{Sensible Heat}}{\text{Total Heat}}$	1.00	0.90	0.80	0.70	0.60
Air entering saturated[a]	60.0	58.6	56.5	53.0	35.0
Air entering with 4 F wet-bulb depression	66.5	65.4	64.1	61.8	56.0
Air entering with 8 F wet-bulb depression	72.6	72.1	71.6	70.5	68.0

[a]Typical air conditions leaving the central conditioner are: With spray dehumidifier, 0 to 2 F wet-bulb depression. With surface-type dehumidifier, 1 to 6 F wet-bulb depression. With by-pass or reheat, 4 to 10 F wet-bulb depression.

plished with a separate humidifier system. On small installations, the simplest method is to use a warm water spray through atomizing nozzles, using a pressure of 15 to 25 lb and sufficient nozzles to atomize about twice the amount of water needed for humidifying. The grains of moisture to be added to the incoming air at outside design temperature to bring it up to the required room dew-point are calculated. This is converted to total pounds of water per hour for the system and sprays designed for twice this amount. Cold water will not vaporize as completely as warm water, and water temperatures from 120 to 150 F are commonly used. Another method is to install a water tank in the air stream with a steam coil submerged in the tank, the humidification being accomplished by the steam boiling the water into vapor, which in turn will be absorbed by the passing air. In both of these systems there is a tendency to deposit lime and other impurities on any surface the water touches, and this scale should be cleaned regularly before it becomes excessive. Water spray should not touch the steam heating coils as they will quickly become coated with scale. The preferred practice is to install sprays between coils and eliminator plates.

Sound Control

Problems of sound control should be jointly considered by the acoustical and air conditioning engineer for satisfactory results. Many installations require noise levels which are relatively low and for that reason equipment must be selected having a very low noise rating. In central systems consideration should also be given to the lining of ducts for the reduction of noise levels within an enclosure. Often reduced speeds of equipment and low air velocities are helpful in eliminating undesirable noise conditions. Information is given in Chapter 30 with regard to acceptable noise levels for various types of rooms and methods are outlined for computing length of duct lining materials.

Automatic Control

The control of an air conditioning system is very important. A simple comfort cooling or heating installation requires a minimum of control, whereas a more complex installation justifies a more complete control. In this connection, it should be mentioned that there are many patents allowed and pending on air conditioning equipment including control, and the designer should consider these factors in selecting all of the equipment. Refer to Chapter 37.

Static Pressure

The total static pressure against which the system must operate may be found by summing up the static losses through the complete system from the outside air intake to the discharge outlets or nozzles. This means that the loss due to friction must be determined for each piece of apparatus involved. Most of these values may be obtained from manufacturers' data tables. For a simple system, the following static pressure drops may be assumed.

1. Outside air inlet, comprised of screen, louver and short duct, may have a loss of 0.2 in. of water.

2. A typical viscous filter at rated capacity and velocity has a drop of 0.25 in. water.
3. The loss of one row of a standard make tempering stack equals 0.09 in. water.
4. The loss of one row of a standard make preheater equals 0.10 in. water.
5. A standard humidifier at rated velocity may have a loss of about 0.35 in. water
6. The loss through one row of a standard make reheater equals 0.12 in. water.
7. A fair assumption for duct losses on a simple system is 0.25 in. water.
8. The static pressure for a nozzle type outlet may be taken as 0.1 in. water.

The sum of these values equals $0.2 + 0.25 + 0.09 + 0.10 + 0.35 + 0.12 + 0.25 + 0.1 = 1.46$ in. which is the static pressure against which the system must operate.

Fans, Motors and Filters

The selection of Fans may be based on data contained in Chapter 27 and for Motors in Chapter 38. Because centrifugal fans reach their maximum efficiency when working against the resistance offered by the average central fan heating system, they are well adapted to such systems and are generally used. Complete information is given in Chapter 26 for the selection of all types of air filtering equipment.

SPECIAL CONSIDERATIONS

In designing a central system for air conditioning there are a number of special considerations not referred to previously which must be considered. Certain features of building constructions are important. The building must be suitable in construction so that the conditions designed can be maintained economically.

For instance, excessive sun load on roofs or glass windows may not only cause excessive heat gain to be absorbed by refrigeration, but direct sun radiation heating converts a surface into a panel heater. This radiant heat is not absorbed until it strikes another mass such as a building wall, or a person. It is therefore possible to have comfortable air conditions surrounding a person, and yet have him uncomfortably warm from radiant heat from a hot wall, window, or ceiling near him. As another example, it is much better to provide hoods over steam tables, coffee urns, etc., than to try to remove this heat by mechanical refrigeration.

Another problem is presented with winter conditioning for maintaining satisfactory relative humidity. If 30 to 40 per cent is desired at all times, no matter how cold it is outside, excessive condensation may collect on single glass windows, or on hardware which connects to the outside such as latches and hinges. Condensation on windows can sometimes be prevented by applying a small amount of local heat under the window. In other cases double glass or storm window construction is used. Refer to Chapters 5 and 7 for condensation temperatures.

In many cases, different zones of a building will require entirely different treatment. In some buildings where offices are exposed on all four sides to sun and wind effect, cooling is required on the sunny side and heating is required on the shady side simultaneously, in spring and fall seasons. The central system must be carefully zoned to apply cooling or heating as they may be needed for each zone, independent of other zones.

In many existing buildings, the central system will be added to a radiation system of heating. The designer should take full avantage of

this radiation for heating the outside walls and windows. At the same time, it must be controlled to prevent overheating the whole system. A few uncontrolled radiators will often change the heat balance and cause excessive overheating of the whole zone, without occupants near the radiators realizing the source of the trouble. All radiation used locally in connection with the control system should be equipped with automatic control to prevent overheating.

The apparatus should be placed for minimum piping and duct work but it must be accessible for maintenance, repair, and cleaning. The air conditioning unit should be designed to provide access for cleaning coils, drip pans, eliminators and for very easy maintenance on filters. This equipment will be in operation for many years, probably for the life of the building, and a little thought spent in the plan will simplify maintenance and assure successful results.

Example 1. With the assumed values as indicated perform the essential calculations to determine the design loads for heating and cooling and the necessary factors for designing the distribution system.

Solution: Design Conditions from Item 1

Outside air dry-bulb, winter	0 F
Outside air dry-bulb, summer	95 F
Outside air wet-bulb, summer	75 F
Inside air dry-bulb, winter	72 F
Inside air wet-bulb, winter	56 F
Inside air relative humidity, winter	35 per cent
Inside air dry-bulb, summer	80 F
Inside air wet-bulb, summer	66.7 F
Inside air relative humidity, summer	50 per cent

200 people—4 kw light load

Design Load for Heating from Item 2

Sensible heat loss through walls, etc	200,000 Btu per hour
Outside air—2000 cfm $\times$ 0.075 $\times$ 60 $\times$ 0.24 $\times$ (72 − 0) =	155,500 Btu per hour
Humidification— $\frac{2000 \times 0.075 \times (40 - 5) \times 60 \times 1040}{7000}$ =	46,900 Btu per hour
Heat loss through ducts	33,200 Btu per hour
Heat gain from lights, etc	Disregard
Total heating capacity	435,600 Btu per hour

Design Load for Cooling and Dehumidifying from Item 3.

	Sensible	Latent
Heat gain through walls, etc	128,160	
Heat gain from occupants (200)	44,000	36,000
Heat emission from appliances	2,000	
Heat gain from lights (4 kw)	13,840	
Heat gain from solar radiation	12,000	
Totals	200,000	36,000 Btu per hour
Outside air:		
2000 $\times$ 15 $\times$ 0.075 $\times$ 0.024 $\times$ 60 =	32,350	
$\frac{2000 \times 0.075 \times (110 - 80) \times 60 \times 1040}{7000}$ =		40,100
Heat gain through ducts	22,200	
Totals	254,550	76,100 Btu per hour

Total Cooling Capacity = 330,650 Btu per hour
Tons Cooling Effect = 27.5

Heat Load Ratio $= \dfrac{200,000}{236,000} = 0.85$

Design Distribution System for Heating

a. Total heating loss in space = 200,000 Btu per hour
Using 72 F plus 18 F = 90 F entering air

$$Q = \frac{200,000}{60 \times 0.075 \times 0.24 \times 18} = 10,300 \text{ cfm}$$

b Total heat loss in ducts between unit and grilles = 33,200 Btu per hour

$$t_d = \frac{33,200}{60 \times 0.075 \times 0.24 \times 10,300} = 3 \text{ F}$$

c 90 F plus 3 F = 93 F temperature leaving coil

Design Distribution System for Cooling

a Total sensible heat gain in space = 200,000 Btu per hour
Using 80 F minus 18 F = 62 F entering air

$$Q = \frac{200,000}{60 \times 0.075 \times 0.24 \times 18} = 10,300 \text{ cfm} = 770 \text{ lb}$$

Latent heat gain = 36,000 Btu per hour = 600 Btu per minute
= 4020 grains per minute = 5 2 grains per pound

Room condition = 80 FDB, 66 7 FWB, 50% RH, 60 FDP, = 77 3 grains
Entering air = 62 FDB, 59 5 FWB, 87% RH, 58 FDP, = 72.1 grains

b Duct gain = 22,200 Btu per hour

$$t_d = \frac{22,200}{60 \times 0.075 \times 0.24 \times 10,300} = 2 \text{ F temperature rise}$$

Leaving coil = 60 FDB, 58.7 FWB, 58 FDP

These calculations are based on maximum load conditions as set forth in the design. For intermediate loads the calculations may show entirely different relationship. For example, the entering air temperature will approach the space temperature as the sensible heat gain or loss decreases, due to outside temperature change, entrance or exit of people, use of artificial lighting, direction and intensity of sun's rays. The entering dew-point will remain much more uniform as it is affected by changes in room moisture gain only and this does not fluctuate greatly. If intermediate load conditions are important the calculations should be repeated for those loads

PROBLEMS IN PRACTICE

1 ● Consider a central heating system handling 10,000 cfm. The resistance to air flow offered by one coil arrangement is 0.9 in. of water and by another coil arrangement is 0.2 in. of water. The fan operates 4000 hours per year and the combined efficiency of motor and fan is 60 per cent. Determine the annual energy saving if the second coil is used.

Difference in system resistance = 0.9 − 0.2 = 0.7 in. of water.

Reduction in power input $= \dfrac{10,000 \times 0.7}{6356 \times 0.60} = 1.83$ hp.

Annual energy saving = 1.83 × 0.764 × 4000 = 5480 kwhr.

2 ● A group of three drafting rooms, having a total volume of 27,000 cu ft, a transmission loss of 110,100 Btu per hour, and an infiltration loss of 34,200 Btu per

hour on the basis of 0 F outdoors and 70 F room temperature, is to be heated by a recirculating central heating system with air entering the rooms at 116 F. How many cubic feet per minute, measured at 70 F, will be required?

Substitute in Equation 1. $H = 110{,}100 + 34{,}200 = 144{,}300$ Btu per hour; $t_y = 116$ F, $t = 70$ F; $Q = \dfrac{144{,}300}{60 \times 0.07492 \times 0.24\ (116 - 70)} = 2900$ cfm.

3 ● In the preceding question, if the warm air loses 4 F between heater and rooms, how many pounds of steam per hour at 1-lb gage will the heating sections condense?

$Q = 2900$ cfm, from solution of Question 2, $\Delta t = 116 + 4 - 70 = 50$ F; $h_{fg} = 968$ Btu, from steam table in Chapter 1.

$$W = \frac{60\ dQ \times 0.24 \times \Delta t}{h_{fg}} = \frac{60 \times 0.07492 \times 2900 \times 0.24 \times 50}{968} = 161.8 \text{ lb per hour.}$$

4 ● In central systems for cooling and dehumidifying can the dry-bulb temperature change be fixed arbitrarily?

No, because the change depends on factors at both the conditioner and the room At the conditioner, temperature of the available water supply may limit the dry-bulb temperature of the leaving air. At the room, the dry-bulb temperature of the entering air may be further limited by: 1. The duct and supply grille arrangement permitted by architectural and structural requirements for the particular space, *e.g.*, ceiling height and obstructions on ceilings, such as beams. 2. The state of activity of the occupants. 3. The velocity at the inlet grille, as limited by noise level requirements. 4. The direction of the jet relative to the occupants.

5 ● Why must the air leaving a dehumidifying type air washer often have its dry-bulb temperature raised before delivery to the occupied zone of room?

The air leaves the dehumidifying air washer saturated at a relatively low temperature which in most cases is lower than the allowable delivery dry-bulb temperature as fixed by factors outlined under Question 4 Also, the air may possibly be carrying a small amount of entrained water which might settle out in the ducts near the washer and cause corrosion difficulties.

6 ● What methods may be used to raise the dry-bulb temperature of the air after it leaves the dehumidifying air washer and before it enters the room?

a. Sensible heat may be added by a reheating method from a source outside the air stream. This method passes all or part of the cold, dehumidified air over steam or hot water coils at the central conditioner or in the ducts, or over electric grids or similar devices. Any available source of sensible heat can be used.

b. A mixing method using sensible heat already in the air stream. In this method the cold, dehumidified air is mixed with air at a higher dry-bulb temperature and the dry-bulb temperature of the resulting mixture is higher than that of the air when it left the conditioner. The air at high dry-bulb temperature is obtained by not passing it through the dehumidifying washer. The mixing may take place at a central conditioner or in the rooms themselves.

c. Combinations of these methods.

Chapter 22

UNIT HEATERS, VENTILATORS, AIR CONDITIONING, COOLING UNITS

Classification of Unitary Equipment and Related Systems, Unit Heaters, Unit Ventilators, Split and Combined Systems, Cooling Units, Air Conditioning Units, Heating, Humidifying and Dehumidification, Filtering, Location of Units, Air Distribution, Residential Central System Units, Costs, Accessories to Unitary Equipment

IN other chapters, complete descriptions have been given of heating, cooling, ventilating, humidifying, and dehumidifying systems. These descriptions have covered the detailed principles of each and have, in general, described the assembled equipment included in the complete systems. The success of such completely engineered, heating, cooling, and air conditioning systems has inevitably led to the production of smaller factory-assembled equipment employing a majority of the principles of these complete systems. As a result, present day practice involves the use of this unitary equipment in the majority of smaller installations where capacity demands are within the limits of such units. Thus, unit heaters, unit ventilators, cooling units and air conditioning units have come to occupy a place of their own in the industry.

With the growth of this unitary industry, it becomes increasingly evident that there is no sharp line of demarcation, on the basis of capacity, between a *unit* and a *central station* system. Definitions contained in a code, Standard Method of Rating and Testing Air Conditioning Equipment[1], have helped to clarify and identify the various types available.

A *unit* is a factory-made encased assembly of the functional elements indicated by its name, such as air conditioning unit, room cooling unit, humidifying unit, etc. Such units are shipped substantially complete or built and shipped in sections so that the only field work necessary is the assembling together of the sections, without resorting to any field fabrication. A unit of this type may be complete in itself, employing its own direct means of air distribution and sources of refrigeration or heating, in which case, it thus represents a complete self-contained unit. Or it may be coupled with separate means of air distribution such as duct work and outlets, in which case, it will still be considered as a unit system, in distinction to the generally accepted term of a field fabricated central station system. The manufacturer of the unit is responsible for the output and

[1]Prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association* and *Air Conditioning Manufacturers' Association*.

performance of the unit under rated conditions, whereas the contractor installing the complete unitary system is normally held responsible for the complete performance of the system.

Unit equipment justifies its existence due to the following features:

1. Lower cost per unit capacity. Standardized design and volume production makes possible low cost factory assembly thereby eliminating individual design and handling of every part for each installation.

2. Flexibility and mobility of equipment. Unitary equipment can be readily located in existing buildings without the necessity of running large ducts through floors and many partitions. Such equipment can be shifted to meet changing requirements. Tenants may obtain the advantages of conditioning when the entire building is not equipped with a conditioning system. In industrial process work, the flexibility of unitary equipment is also advantageous.

3. Lower installation costs. The fact that the equipment arrives on the job in an assembled condition, coupled with the lesser problems of duct work and connecting piping, materially reduces installation costs.

4. Small capacities. The small capacities available in unitary equipment have brought the advantages of controlled air conditions to a number of small offices, stores, shops, and individual rooms where specially designed and built central system equipment would have been uneconomic

SUB-DIVISION OF UNITARY EQUIPMENT

For descriptive purposes unitary equipment is sub-divided on a purely functional basis. The following definitions are included in the previously referred to code[2].

1. A *Heating Unit* is a specific air treating combination consisting of means for air circulation and heating within prescribed temperature limits.

2. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.

3. A *Humidifying Unit* adds water vapor to and circulates air in a space to be humidified.

4. A *Dehumidifying Unit* removes water from and circulates air in a space to be dehumidified.

5. An *Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for maintaining temperature and humidity within prescribed limits.

6. A *Cooling Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for cooling and maintaining humidity within prescribed limits.

7. A *Heating Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for heating and maintaining humidity within prescribed limits.

8. A *Self-Contained Air Conditioning* or *Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements. Self-contained Air Conditioning Units are classified[3] according to the method of rejecting condenser heat (water cooled, air cooled, and evaporatively cooled), method of introducing ventilation air (no ventilation, ventilation by drawing air from outside, ventilation by exhausting room air to the outside, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure type)

9. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

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[2]Loc Cit. Note 1

[3]Proposed Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association*, and *Air Conditioning Manufacturers' Association*

10. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.

There has grown up in the industry definite branches, in which the engineering and application of the equipment vary quite widely. Thus common acceptance recognizes the following groups of the unitary equipment defined above.

1. *Unit Heaters* consisting of an encased heating surface through which air is forced by means of a fan or blower, located either in or closely adjacent to the heated space, and normally employed only for industrial and commercial applications.

2. *Unit Ventilators* which are similar in principle to unit heaters but are designed to use outside air with or without provision for recirculation of the air. While unit heaters are largely used for commercial and industrial applications, unit ventilators are intended primarily for school, offices and semi-commercial applications.

3. *Cooling Units* which are similar to unit heaters except that a cooling medium is used in place of a heating medium and provision is made to collect and remove the condensate. Cooling units are normally applied to the cooling of products for their preservation or processing (commercial air conditioning) and air conditioning units are used for cooling for comfort.

4. *Air Conditioning Units* which consist of equipment to provide control of heating with humidifying or cooling with dehumidifying, coupled with air circulation, all compactly housed in a single casing.

5. *Miscellaneous Unit Equipment and Accessories* such as filtering equipment, attic fans, humidifying units and special controls

UNIT HEATERS

A *unit heater* consists of the combination of a heating element and fan or blower having a common enclosure and placed within or adjacent to the space to be heated. Generally no ducts are attached to inlets or outlets, although it is common practice with many unit heaters to equip them with directional outlets or adjustable louvers.

While unit heaters are designed primarily to handle all recirculated air, they may be installed to handle either partial or total outdoor air. Compared with the older method of heating by means of radiation, properly designed and applied unit heaters should:

1. Circulate air in the building at a rapid rate but without objectionable draft
2. Reduce the temperature differential between the floor and ceiling.
3. Direct the heated air so that uniform temperature distribution be obtained throughout the heated space
4. Prevent or remove the cold stratum of air commonly found at the floor level
5. Reduce the number of heating elements required and thereby decrease the cost and extent of the piping necessary.
6. Maintain a closer control of room temperature either manually or by means of simple thermostats.
7. Produce an economy in heating costs resulting from the sum total of the above advantages
8. Provide a means of saving floor area or room space due to the compactness of the equipment and flexibility of application

TYPES OF UNITS

There are two major types of unit heaters, propeller fan type and centrifugal housed fan type. The housed fan, high velocity (1500 to 2500 fpm) discharge units with outlets adjustable to deliver air in several directions, are able to project their heating effect over distance of from

30 ft to as much as 200 ft from the unit. This makes possible the location of these units at considerable distances from each other, thus reducing greatly the piping and loss of floor space due to the heating equipment. Propeller-type units, illustrated in Fig. 1, with outlet velocities of from 300 to 1000 fpm are usually placed from 30 up to 100 ft apart.

Two methods of application of unit heaters are commonly used. Floor mounted units, shown in Fig. 2 are available either with or without the air by-pass, and withdraw the cold air from the floor and discharge the heated air above the working zone. Suspended type units are located in an elevated position withdrawing air from this higher level and discharging the heated air down into the working zone. In closely occupied spaces where direct air drafts into the working zone are not permitted,

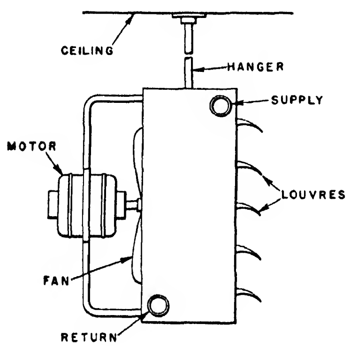

FIG. 1. SUSPENDED UNIT HEATER, PROPELLER TYPE FAN

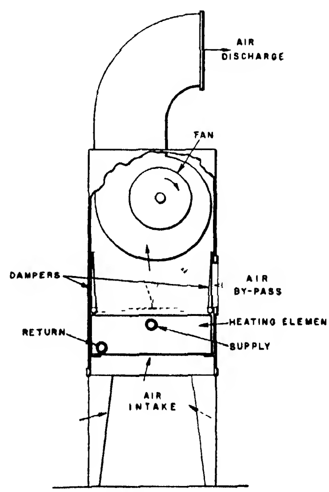

FIG. 2. FLOOR MOUNTED UNIT HEATER, HOUSED TYPE FAN

the floor mounted unit will give more uniform temperature distribution. On the other hand, if opportunity is provided to deliver the heated air from suspended units down into the working zone, excellent temperature distribution is possible. A suspended high-velocity type unit heater connected to an outside air intake with damper to control the volume of ventilation is shown in Fig. 3. A code[4] governing the number of sizes of propeller type unit heaters offered for sale by a manufacturer, as well as a standard method of specifying outlet air velocities has been adopted.

A wide variety of structural designs is available. All employ some form of convector, supplied with either steam or hot water, although occasionally equipped for gas or electric heat. Air is always forced over these convectors by a fan of either the propeller or centrifugal type. Heating surfaces may be in the form of steel pipe coils, non-ferrous tubes or pipes with extended surfaces, cast-iron, and pressed or built-up sections of the cartridge or automotive type.

[4]Standards for Propeller Type Unit Heaters prepared and adopted by the *Industrial Unit Heater Association*, June, 1938

AIR TEMPERATURES[5]

For recirculating heaters with intakes at the floor level, the temperature to be maintained in the room should be considered as the temperature of the air entering the heater. Where outside air is introduced, the temperature of the mixture must be calculated and used as the entering air temperature to the heater. Where suspended heaters are used without any intake boxes extending down to the floor level, a higher entering air temperature should be used than that at which the room is to be maintained.

With suspended unit heaters taking air at some distance above the floor, the temperature variation from floor to ceiling may reach as much as 1 deg for each foot of elevation during the periods when the maximum capacity of the heaters is required. Thus this allowance should be made in calculating the capacity of suspended heaters. High velocity discharge

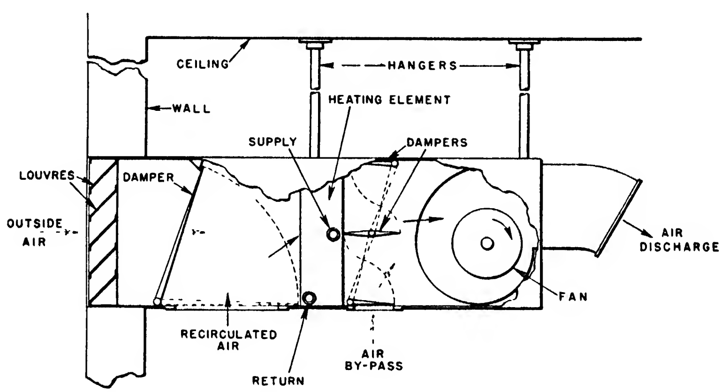

FIG. 3. SUSPENDED TYPE UNIT HEATER, HOUSED TYPE FAN

units (blower type) will maintain slightly lower temperature differences than will low velocity units (propeller type). Unit heaters taking in recirculated air at the floor level should maintain temperature differentials of less than 0.5 deg per foot of elevation when the maximum capacity of the heaters is required. This temperature difference per foot of elevation is less than the corresponding variations for spaces heated by direct radiation.

OUTPUT OF HEATERS

It is standard practice to rate unit heaters in Btu per hour at a given temperature of air entering the heater and at a given steam pressure maintained in the coil. Steam at 2 lb pressure and air entering at 60 F are used as the standard basis of rating[6]. The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature

[5]A S H.V E RESEARCH REPORT No 958—Temperature Gradient Observations in a Large Heated Space, by G L Larson, D. W. Nelson, and O. C Cromer (A S H V E TRANSACTIONS, Vol 39, 1933, p. 243).

A S H V E RESEARCH REPORT No 1011—Tests of Three Heating Systems in an Industrial Type of Building, by G L. Larson, D W Nelson, and John James (A S H V E TRANSACTIONS, Vol 41, 1935, p 185)

[6]A S H.V.E. Standard Code for Testing and Rating Steam Unit Heaters (A S H V.E TRANSACTIONS, Vol 36, 1930, p. 165).

TABLE 1. CONSTANTS FOR DETERMINING THE CAPACITY OF UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2-lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In	Temperature of Entering Air											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.538	1 446	1 369	1 273	1 191	1.110	1 034	0.956	0 881	0 809	0.739	0 671
2	1.585	1.495	1 405	1 320	1 237	1 155	1 078	**1.000**	0 926	0 853	0.782	0.713
5	1.640	1.550	1.456	1.370	1.289	1 206	1 127	1.050	0 974	0.901	0 829	0.760
10	1.730	1 639	1.545	1 460	1 375	1 290	1 211	1.131	1 056	0 982	0 908	0 838
15	1.799	1.708	1 614	1 525	1.441	1 335	1.275	1.194	1 117	1 043	0.970	0.897
20	1 861	1 769	1 675	1 584	1 498	1 416	1 333	1.251	1 174	1.097	1 024	0.952
30	1 966	1.871	1 775	1 684	1 597	1 509	1 429	1.346	1 266	1.190	1.115	1.042
40	2 058	1 959	1 862	1 771	1 683	1 596	1.511	1.430	1 349	1 270	1.194	1.119
50	2.134	2 035	1 936	1 845	1 755	1 666	1 582	1 498	1.416	1 338	1 262	1.187
60	2.196	2.094	1 997	1 902	1 811	1.725	1 640	1 555	1.472	1 393	1.314	1.239
70	2 256	2.157	2 057	1.961	1 872	1 782	1 696	1 610	1 527	1.447	1 368	1.293
75	2 283	2 183	2 085	1 990	1 896	1 808	1.721	1 635	1 552	1 472	1 392	1.316
80	2.312	2 211	2.112	2 015	1.925	1.836	1 748	1 660	1 577	1.497	1 418	1.342
90	2.361	2 258	2 159	2 063	1 968	1 880	1.792	1.705	1.621	1.541	1.461	1.383
100	2 409	2.307	2 204	2.108	2 015	1 927	1.836	1.749	1 663	1 581	1 502	1.424

Note.—To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

increases. The heat capacity for any condition of steam pressure and entering air temperature may be calculated approximately from any given rating by the use of factors in Table 1. This table is accurate within 5 per cent.

Unit heaters are customarily rated as free delivery type units. If outside air intakes, filters, or ducts on the discharge side are used with the heater, proper consideration should be given to the reduction in air and heat capacity that will result because of this added resistance.

The percentage of this reduction in capacity will depend upon the characteristics of the heater and on the type, design, and speed of the fans employed, so that no specific percentage of reduction can be assigned for all heaters for a given added resistance. In general, however, disc or propeller fan units will have a larger reduction in capacity than housed fan units for a given added resistance, and a given heater will have a larger reduction in capacity as the fan speed is lowered. When confronted with this problem the ratings under the conditions expected should be secured from the manufacturer.

DIRECTION OF DISCHARGE

Heaters may be distributed through the central portions of a room discharging toward exposed surfaces, or may be spaced around the walls, discharging along the walls and inward as well, especially when there are considerable roof losses.

In general, it is better to direct the discharge from the unit heaters in such fashion that rotational circulation of the entire room content is set up by the system rather than to have the heaters discharge at random and in counter-directions.

Various types and makes of unit heaters are illustrated in the *Catalog Section* of this edition. Usually hot blasts of air in working zones are objectionable, so heaters mounted on the floor should have their discharge outlets above the head line and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air shall be brought as close to the head line as possible, yet not into the working zone. In general, the higher the elevation of the unit, the greater the volume and velocity required to bring the warm air down to the working zone, and consequently, the lower the required temperature of the air leaving the unit.

BOILER CAPACITY

The capacity of the boiler should be based on the rated capacity of the heaters at the lowest entering air temperature that will occur, plus an allowance for line losses. Ordinarily for recirculating heaters the lowest entering temperature will occur at the beginning of the heating period and is usually taken as 40 F, while for ventilators taking air from outdoors the lowest entering temperature will be the extreme outdoor temperature expected in the district. No greater allowance in boiler capacity beyond the calculated heat demand need be added in order to supply unit heaters than for any other type of system.

It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attendance to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, however, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller heating units instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their coils are designed for the purpose and when proper provision is made for returning the condensate. If ventilators are

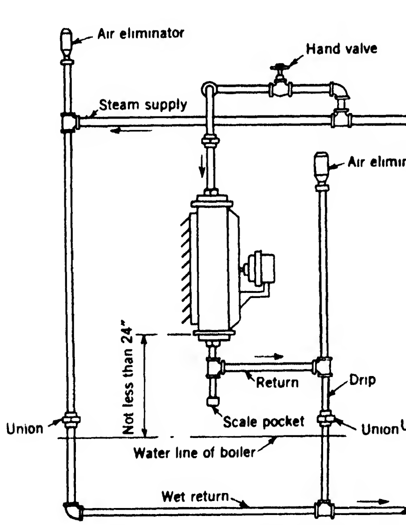

FIG. 4. UNIT HEATER CONNECTION TO ONE-PIPE GRAVITY STEAM SYSTEM

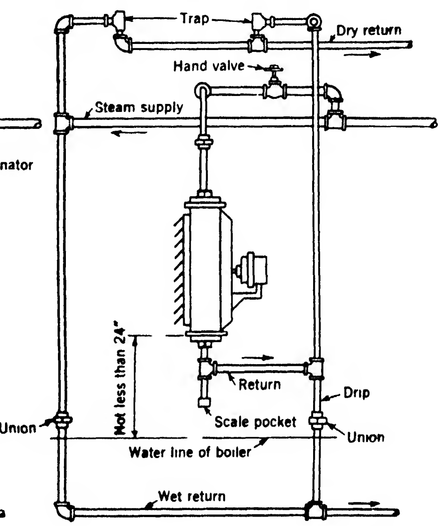

FIG. 5. UNIT HEATER CONNECTED TO GRAVITY SYSTEM WITH WET AND DRY RETURNS

to take in air that may be at a temperature below freezing, however, a steam pressure of not less than 5 lb should be maintained on the convector or a corresponding differential in pressure between the supply and returns be maintained by means of a vacuum.

PIPING CONNECTIONS

Piping connections for unit heaters are similar to those for other types of fan-blast heaters. The piping around the unit heaters must strictly conform to the system requirements while at the same time permitting the heaters themselves to function as intended. Some of the more frequently encountered piping arrangements are illustrated in Figs. 4, 5, 6 and 7. Unit heaters may be applied to one-pipe gravity and vapor systems if the piping connections are arranged with proper care and attention.

A method of connecting a unit heater to a one-pipe gravity system is illustrated in Fig. 4. In those cases where the unit heater is to be con-

nected to a dry return instead of a wet return it is necessary to provide a water pocket or loop about 5 ft in depth to prevent steam passing into the return and thus into other equipment.

A method of connection is shown in Fig. 5, where there is a wet return and a dry return. In this case the condensate from the heater and the drip from the supply main drop to the wet return by gravity, while the air passes upward through the traps to the dry return and is vented from the system at any suitable location.

A sketch of an arrangement where there is a dry return line through which both air and condensate pass to be handled by some suitable means, such as a condensation pump and receiver is given in Fig. 6. The return line is not subjected to vacuum, and consequently all arrangements must facilitate gravity flow of the condensate toward the receiver. Traps must

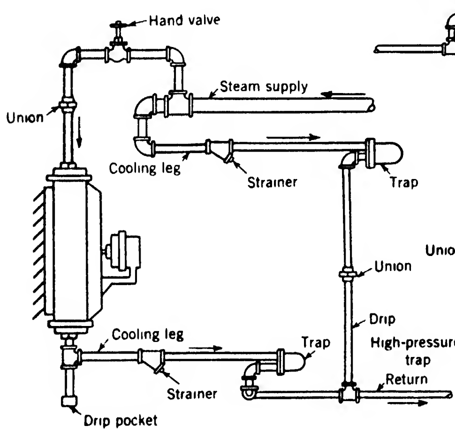

FIG. 6. UNIT HEATER CONNECTION FOR VACUUM OR VAPOR SYSTEM DISCHARGING CONDENSATION INTO DRY RETURN

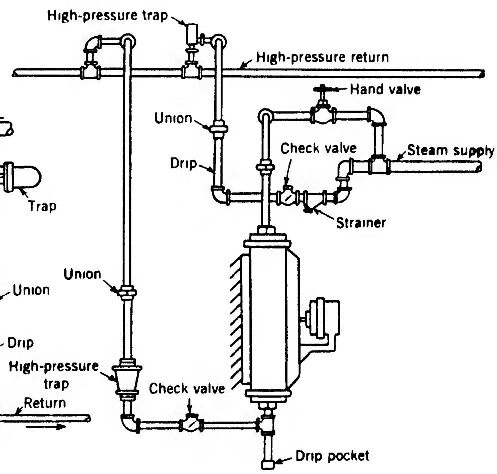

FIG. 7. METHOD OF CONNECTING UNIT HEATER TO HIGH PRESSURE RETURN

pass air and condensate rapidly to keep the return piping only partially full of water.

Since unit heaters are often constructed with sufficient strength to resist high pressures, use of high pressure steam in them is a common practice. In Fig. 7 the condensate and air reach the return overhead through traps, and check valves are located in the return piping.

For two-pipe closed gravity return systems, the return from each unit should be fitted with a heavy duty of blast trap, and an automatic air valve should be connected into the return header of each unit. Pressure drop must be compensated for by elevation of the heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual air valves on the heaters, or it may be carried into the returns the same as for vacuum systems and the entire return system be free-vented to the atmosphere, provided all units, drip points, and radiation are properly trapped to prevent steam entering the returns.

On vacuum or open vented systems the return from each unit should be fitted with a large capacity trap to discharge the water of condensation and with a thermostatic air valve for eliminating the air, or with a heavy-duty trap for handling both the condensation and the air, provided the air finally can be eliminated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, with some danger of return pipe corrosion and the problem of its elimination at some other point in the system.

OTHER TYPES OF UNITS

All Electric

The foregoing discussion relates generally to units in which steam or hot water is used as the heating medium. Electric unit heaters are applied where electric power is abundant and cheap and where other forms of fuel are scarce and expensive. The low first cost, easy control, and inexpensive installation of this type of heating have also accounted for many other installations in which electricity has conveniently provided heat for short periods of time. (See Chapter 40).

Direct Fired

A recent development in gas burning equipment is the direct-fired industrial unit heater. These heaters are of the warm-air type and are equipped with fans which cause the air to pass over the heating surfaces at a fairly high velocity and then direct the warm air in to the space to be heated. As is the case with the steam-fed unit heaters, the gas-fired appliances may be used for heating stores, shops, and warehouses. They usually are suspended in the space to be heated and in most instances leave the entire floor and wall area free for commercial use. Partial or complete automatic control also may be secured on appliances of this type. This type of heater is often used for temporary heat during building construction or where the installation of a steam or hot water plant is for some reason not justified. For permanent installations, it is usually advisable to provide an exhaust duct from the gas-fired unit heaters to remove products of combustion from the occupied space. While this is not necessary in large open industrial plants, in smaller closed rooms, it becomes essential.

Turbine Driven

Where high pressure steam is available it is sometimes used to drive a steam turbine direct-connected to the unit heater. The exhaust from this turbine, reduced in pressure, is then passed into the heating coil where it is condensed and returned to the boiler.

INDUSTRIAL USES

In addition to their prime function of heating buildings, unit heaters may be adapted to a number of industrial processes, such as drying

and curing, with which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye-houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in which process moisture is given off. When such conditions are severe, it is necessary that the heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. (See discussion of condensation in Chapter 7).

UNIT VENTILATORS[7]

Unit ventilators while designed primarily for ventilation must incorporate controlled heating. A typical unit ventilator is illustrated in Fig.

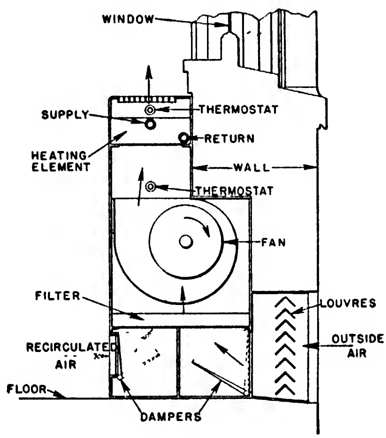

FIG. 8. TYPICAL UNIT VENTILATOR SHOWING ONE OF MANY ARRANGEMENTS OF DAMPERS AND HEATING COILS

8. They usually consist of a semi-decorative cabinet containing the following necessary or optional parts:

1. Outside air inlet.
2. Inlet damper for closing the opening to the outside air inlet when the unit is not in use.
3. Adhesive or dry type filters for cleaning the air (optional).
4. A heating element usually of special design and intended for low pressure steam.
5. Motor and fan assembly.
6. Mixing chamber where warm and cold air streams are brought together. (No mixing chamber is normally provided where sectional type convectors are used).
7. Outdoor air inlet and recirculating air mixing damper (optional).
8. Discharge grille or diffuser.
9. Temperature control arrangement.

The primary functions of a unit ventilator are:

[7]A roof ventilator is sometimes termed a *unit ventilator* For information on roof ventilators, see Chapter 36.

1. To supply a given quantity of outdoor air for ventilation or to mix indoor and outdoor air. (See A.S.H.V.E. Ventilation Standards, Chapter 45).

2. To warm the air to approximately the room temperature if the unit is intended for ventilation only, or to a higher temperature if it is intended to take care of all or a part of the heat transmission losses from the room.

3. To control the temperature of the air delivered so as to prevent both cold drafts and overheating. (See Chapter 37).

4. To deliver air to the room in such a manner that proper distribution is obtained without drafts.

5. To recirculate room air for the purpose of heating or promoting comfort when ventilation is unnecessary. (Ordinances should be consulted).

6. To perform all its functions without objectionable noise.

7. To clean the air properly

SPLIT AND COMBINED UNIT VENTILATOR SYSTEMS

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at very near the room temperature, and enough separate direct heaters are placed in the room to warm it to the desired temperature, independently of the unit. Their principal advantage lies in offsetting the cooling effect of window and wall surfaces long before these can be heated to room temperature and in retaining heat for this purpose after the ventilation is shut down.

Where the unit ventilator selected has a capacity more than sufficient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct heating surface installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current, since the direct radiators will furnish heat, but it permits a careless operator to avoid operating the ventilating equipment.

A combined system employs the unit ventilator alone, its capacity being sufficient both for ventilation and for supplying the heat loss. Direct heating surface is omitted altogether. It becomes necessary then that the fan be running whenever the room is to be heated but this also gives assurance of ventilation, especially if automatic dampers are used in the air intake from out-of-doors and in the recirculating intake arranged so as to give a certain quantity of air from the outside (commensurate with weather conditions) whenever the unit is operating and after the room is heated. The cost of installation of a combined system is usually less than that of a split system and there is less danger of overheating, but if the electric energy fails there will be practically no heating.

LOCATION OF UNIT

The location of the unit ventilator in a room is important. Wherever possible it should be placed against an outside wall. It is difficult to obtain proper air distribution if the unit is erected either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set away from the wall or partially recessed into the wall to save space without materially affecting the results. The air inlet may enter the cabinet at the back at any point from top to bottom.

TABLE 2. TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO

Cubic Feet of Air per Minute	Total Capacity in Square Feet of Equivalent Direct Heating Surface (Radiation)		Capacity Available for Heating the Room in Square Feet of Equivalent Direct Heating Surface (Radiation)		Final Air Temperature (Deg Fahr)
	EDR	Mbh	EDR	Mbh	
600	285	68	95	23	105
750	350	84	115	28	105
1000	455	110	150	36	105
1200	565	136	190	46	105
1500	705	169	235	56	105

VENTS[8]

The size and location of the vent outlet is important. In many cases the sizes for public buildings are regulated by law, but the location of the vents generally is left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system.

The cross-sectional area of the vent flue itself may be figured on the basis of 15 sq in. of flue for each 100 cfm. Thus the vent flue area of a flue for a room equipped with one 1200 cfm unit ventilating machine would be 180 sq in. The area of vent flue opening from the room may be figured on the basis of 25 sq in. per 100 cfm.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, heating and ventilating them with air which otherwise would be passed to the outside without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

There has been much controversy over the use of corridor ventilation in school building practice, one group holding the view that when each classroom has a separate vent flue there is a minimum fire risk and less likelihood of cross-contamination, while others emphasize the economy features of the corridor discharge and minimize the fire, contamination, and other hazards.

CAPACITIES

Unit ventilators are available in air capacities ranging from 450 cfm to 5000 cfm and with corresponding heat capacities (above that required for ventilation purposes based upon an outside temperature of zero and an inside temperature of 70 F) ranging from 15 Mbh to 144 Mbh (1 Mbh = 1000 Btu per hour). Some manufacturers furnish a unit with several heating capacities for each air capacity, thus enabling the engineer to

[8]A S H V E RESEARCH REPORT No 936—Investigation of Air Outlets in Class Room Ventilation, by G L. Larson, D W Nelson, and R W Kubasta (A S H V E TRANSACTIONS, Vol 38, 1932, p 463)

A S H V E RESEARCH REPORT No 1017—Air Supply to Classrooms in Relation to Vent Flue Openings, by F C Houghten, Carl Gutberlet, and M F Lichtenfels (A S H V E TRANSACTIONS, Vol 41, 1935, p 279)

select the unit best adapted to the heating and ventilating load. Typical capacities are given in Table 2[9].

If no direct heating surface (radiation) is installed, the combined heating and ventilating requirements must be taken care of by the unit ventilators, and the total heat to be supplied is obtained by means of the following formulæ:

When all of the air handled by the unit is taken from the outside,

$$H_t = 0.24\, W\, (t_y - t_o) \tag{1}$$

$$W = d\, 60\, Q \tag{2}$$

$$t_y = \frac{H}{0.24 W} + t \tag{3}$$

where

d = density of air, pounds per cubic foot.
H = heat loss of room, Btu per hour.
H_v = heat required to warm air for ventilation, Btu per hour.
H_t = total heat requirements for both heating and ventilation, Btu per hour $= H + H_v$.
Q = volume of air handled by the ventilating equipment, cubic feet per minute
t = temperature to be maintained in the room.
t_o = outside temperature.
t_y = temperature of the air leaving the unit.
W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure.

From Equations 1, 2 and 3:

$$H_t = H + 0.24\, d\, 60\, Q\, (t - t_o) \tag{4}$$

Example 1. The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70 F and all air is taken from the outside at zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution. $H = 24{,}000$; $d = 0.075$ $Q = 1000$ cfm; $t = 70$ F; $t_o = 0$ F.

Substituting in Equation 4:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\, (70-0) = 99{,}600 \text{ Btu}$$

$$t_y = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F}$$

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$H_t = 0.24 W_o\, (t_y - t_o) + 0.24\, W_i\, (t_y - t) \tag{5}$$

where

W_o = weight of air, pounds per hour taken from out-of-doors.
W_i = weight of air, pounds per hour taken from the room.

$$W_o = d_o\, 60\, Q_o \tag{6}$$

$$W_i = d_i\, 60\, Q_i \tag{7}$$

[9]A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25).

where

d_o = density of air, pounds per cubic foot at temperature t_o.
d_i = density of air, pounds per cubic foot at temperature t
Q_o = volume of air taken in from the outside, cubic feet per minute
Q_i = volume of air taken in from the room, cubic feet per minute

$$t_y = \frac{H}{0.24\,(W_o + W_i)} + t \qquad (8)$$

$$H_t = H + 0.24\,d_o\,60\,Q_o\,(t - t_o) \qquad (9)$$

Equations 5, 6, 7, 8, and 9 may be used in the same manner as is illustrated above for Equations 1, 2, 3, and 4. It may be noted in Equation 9, representing the total heat requirements, that as the quantity Q_o is diminished the heat requirements for the unit diminish very materially.

In Example 1, if the quantity of air taken in from the outside is reduced to zero, or all of the air handled by the unit is recirculated, the total heat requirements H_t reduce from 99,600 Btu to 24,000 Btu, or to about one fourth. Such a unit handling one third of its air volume from the outside and two thirds from the room would show a total heat requirement of $24{,}000 + \frac{99{,}600 - 24{,}000}{3} = 59{,}200$ Btu. Units designed and operated on this principle show an average heat requirement and, therefore, a boiler capacity requirement of less than 50 per cent of that required for units taking all their air from the outside.

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24\,W\,(t_y - t) \qquad (10)$$

If the heat loss of the room is to be taken care of by the direct heating surface, the unit ventilators will be required to warm the air introduced for the ventilating requirements. Therefore:

$$H_v = 0.24\,W\,(t_y - t_o) \qquad (11)$$

In this case t_y should be equal to or slightly higher than t. If the unit ventilator were of such capacity as to exactly provide for the ventilating requirements, the direct radiation would be selected on the usual basis. However, it is necessary to employ a unit which may not exactly meet the ventilating requirements, since standard units are usually rated in terms of the volume of air that will be delivered at a certain temperature t_y for an initial temperature of t_o. Therefore a certain amount of heat (H_h) may be available from the unit ventilator for heating purposes, as previously stated, and the amount of equivalent direct heating surface may, if desired, be deducted from the amount required for heating the room.

COOLING UNITS

Cooling units as applied to industrial product conditioning and processing are similar in construction to unit heaters except that the heat transfer surface is supplied with refrigeration instead of with steam or hot water. They are normally installed within the space to be served, or at

least closely adjacent thereto. Occasionally they are provided to receive outside air in which case this air is invariably filtered or washed to prevent any possible contamination of the product.

Cooling units are provided in two major types similar to unit heaters, either floor mounted with housed fan, or suspended with propeller type fans. Normally, air outlet velocities are lower than for heating, due largely to the effect of high velocities on the product. Cooling units are normally of the free delivery type although they occasionally are supplemented with duct work to provide more careful air distribution.

Product cooling originally was accomplished by means of stationary pipe coils. This was later supplemented with the forced fan bunker systems in which air was passed over banks of coils. The present trend in this field is toward a more accurate control of both temperature and humidity, thus placing these units in the classification of complete air conditioning units as discussed in the next section. However, in the majority of these cases dry-bulb temperature is controlled separately from the control of humidity, thus classifying these units as cooling units.

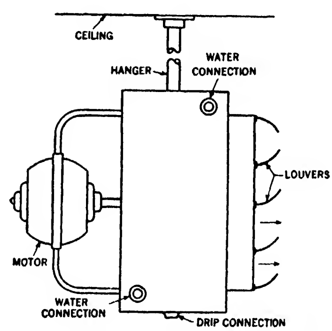

FIG. 9. CEILING TYPE COOLING UNIT

The principal field for cooling units is in cold storage plants, fur storage, fruit packing houses, provision stores, brewery fermentation and stock rooms, candy plants, and other industrial process work. In replacing bunker and wall coils in meat storage plants, cooling units give distinct advantages in compactness, lower first cost and maintenance expense, ease of defrosting, freedom from drip and the maintenance of sanitary conditions, as well as uniform temperature and humidity under variable load conditions. Cooling units by means of their positive air circulation prevent dead-air spots, frequently objectionable in this industry.

Typical cooling units are shown in Figs. 9 and 10. The former indicates a suspended type cooling unit which may be designed with or without a moisture eliminator. If high air velocities are maintained, an eliminator will be necessary to prevent the drops of moisture from being carried through with the air. The condensation that occurs is collected in a drip pan and removed from the system through a drain pipe. Fig. 10 indicates a typical floor-mounted unit of the housed fan type. The illustration shows a common form of distributing outlet designed to give low outlet velocities together with a controlled distribution. In process work, it is often important that direct air distribution does not impinge on the

product. Cooling units are normally constructed of galvanized steel or non-ferrous material in order to reduce the corrosive effect of their constant wetted condition.

Cooling units are often called upon to operate in rooms where a temperature below freezing is maintained and low refrigerant temperatures are required. This results in the collection of frost on the heat transfer surface which in turn leads to a rapid loss in capacity and requires eventual defrosting. Such defrosting is accomplished by the following methods:

1. When the room is above freezing the source of refrigeration is cut off and the fan allowed to operate until the unit has defrosted

2. A reversal of the refrigeration system may be provided and the so-called hot gas defrosting method used. This is accomplished by reversing the flow of the hot

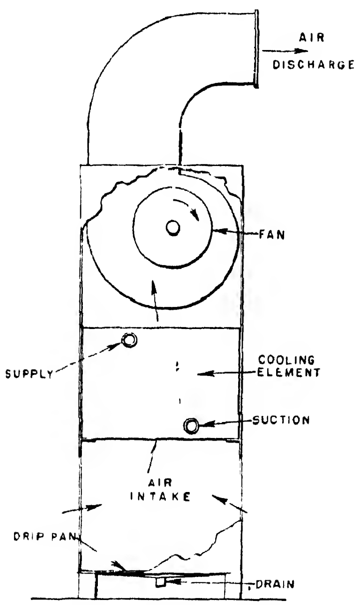

FIG 10 SURFACE TYPE COOLING UNIT

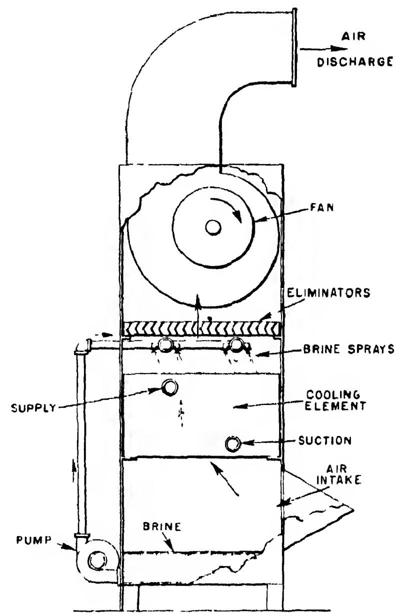

FIG. 11. BRINE SPRAY TYPE COOLING UNIT

gas so that it is delivered directly from the compressor to the evaporator of the cooling unit As soon as the ice and frost has been melted, the system is again returned to its normal cycle

3 Where brine is used as a refrigerant, heated brine may be sent through the cooler to remove the ice

4 When the room is at very low temperatures, warm air defrosting is sometimes used by providing for the admission and removal of warm air from outside the cooled space

5 The surface may be sprayed with a strong brine solution

In order to prevent the collection of frost in low temperature rooms where high latent heat loads are present, unit coolers equipped with a constant brine spray are frequently used. These are normally of the housed fan type similar to Fig 10, but equipped with a pump for recircu-

lating brine over the coil as shown in Fig. 11. It is, of course, necessary to strengthen the brine at intervals to maintain a non-freezing mixture.

Ratings of cooling units may be expressed in Btu per hour, or in tons of refrigeration and should specify the quantity, temperature and humidity of the air entering the unit with a stipulated refrigerant temperature within the coil. When chilled water or brine is used, the rate of circulation of the cooling media as well as its entering temperature must be given.

AIR CONDITIONING UNITS

Air conditioning with unit equipment has gained in popularity during the last few years and this type of apparatus now represents the bulk of the production of the industry. It is to be noted that some equipment does not fulfill all of the basic requirements of a true air conditioning unit. True air conditioning equipment involves not only the ability to alter temperature and humidity conditions within the conditioned space, but it must also be able to *control* these conditions.

The means for accomplishing these functions are outlined herewith:

Heating

The normal air conditioning unit derives its heating function from a heating coil, usually of the non-ferrous finned tube type supplied with either steam or hot water. Steam may be supplied directly from a self-contained and built-in oil or gas fired unit, from a separate domestic steam boiler, or even from an outside source of a central heating plant. Hot water is supplied either from a separate hot water boiler or in rare instances from a domestic water heater.

In domestic or household conditioning units, adapted from a warm-air heater to which humidification is added, or possibly all-year-round conditioning, a direct-fired air interchanger is frequently used. The source of heat in this case may be from the combustion of coal, oil, or gas. A wide variety of designs and structures are used. In such direct-fired systems, the bulk and volume of the heat transfer surface is necessarily large in proportion to the rest of the equipment.

Where electric power is low in cost, electric heat has been furnished for air conditioning units either in the form of encased heaters, or open wire heaters. (See Chapter 40). Radiant electric heaters are seldom used except as their radiant heat is absorbed by some receiving wall and there transmitted to the air in the form of convected heat.

Another method of applying electric heat is by means of the reversed refrigeration cycle, whereby electric energy is used to compress a refrigerant and to deliver the heat of compression, withdrawn from a lower temperature source, to the conditioned space by locating the condensing coils in the air circulation circuit of the air conditioning unit. While this method of heating has gained wide interest, it is practical only in a limited number of applications.

Humidifying

A variety of methods have been used to furnish humidification in winter to air conditioning units. The oldest and best known is by means of a direct spray which is used in many different ways. The simplest

system is where the spray water is furnished from a constant water source, such as city water, and is permitted to run to waste. Under such conditions, the spray may be either of the direct atomizing type where, by means of the nozzles, the water is broken into fine particles, or of the so-called target spray type, where a fine stream of water under pressure is caused to impinge upon a flat surface or target. Such methods are normally rather inefficient in the use of water.

In some units, in order to increase the humidifying capacity and to utilize a greater portion of the spray water, the atomized spray is permitted to impinge against a heated surface thereby forcing its evaporation. While this is practical in some instances, there is danger of scale formation where hard water is employed.

One of the simplest methods of humidification in winter is by means of a direct steam spray. This is seldom used in air conditioning units for comfort applications due to the resulting odors. In industrial applications, however, it finds frequent use. The steam is usually introduced to the air through a perforated tube or through some type of porous material.

If a small atomizing spray is not used in comfort conditioning units, the evaporative pan type of humidifier is usually employed. This consists of a container offering as much water surface as possible and equipped with means of heating the water. This heat may be applied either electrically, by steam, hot water, or by circulation of the water from a heated space through the evaporating pan. Since the humidification is accomplished by surface evaporation only, it is essential that the air stream be directed across the surface and that the evaporating surface be large. While this system eliminates the dusting hazard when hard water is used with a spray system, hard water tends to scale the heating surface and results in loss of capacity and the need for frequent cleaning.

The rate of evaporation per unit surface exposed is low, thus it frequently becomes difficult to provide sufficient surface for adequate capacity. The higher the temperature of the water, the lower the relative humidity of the air; the greater the velocity over the surface, the greater is the rate of humidification. The evaporative pan type of humidification limits the water wastage and is usually supplied with water through a float valve. Due to the collection of salts in this evaporating pan such humidifying systems require occasional drainage and cleaning.

Other methods of humidification attempted in air conditioning units are through the use of wetted fabrics, porous eathernware plates, or other capillary surfaces. These methods rely upon the capillary absorption of the moisture up from the liquid level into the portion exposed to the air. They have a tendency to lose their effectiveness due to the resulting deposit of mineral salts at the evaporating surfaces thereby clogging the pores and reducing the contact of the air with the water. Also they frequently become foul and often support bacterial growth.

Cooling and Dehumidification

Those units that employ recirculated water sprays will undoubtedly use such sprays as their means of cooling and dehumidification by furnishing refrigeration to the water in circulation. Occasionally where

an adequate source of cold well water is available, this may be used as a direct spray and run to waste.

Other methods of dehumidification accomplished by direct contact with the transfer medium are by means of the so-called *adsorption* and *absorption* systems. (See Chapter 23). It must be recognized that these methods of dehumidification do not in themselves provide cooling. The substance removes the water vapor from the air thereby heating it. This highly dehumidified air may then be cooled either by partial rehumidification or by direct contact with a cooling medium of cold water or direct expansion refrigerant. There are now on the market solid adsorbents such as silica gel and activated alumina. Water solutions of the chloride of various inorganic elements such as calcium and lithium chloride are the absorbents most frequently used.

Finally a common direct means of cooling and dehumidification is through the use of ice. In such units the ice is brought into as intimate contact as possible with the air handled. Provision is made for the removal of the moisture as rapidly as it is formed from the melting of the ice. Ice is also used to cool water which is circulated through the sprays

In conditioning units, the use of surface cooling is probably more common than direct spray or other direct transfer means. The type of surface employed may, of course, be cast or fabricated from tubes. In present day practice finned tubes or plate fins through which tubes are passed form the most generally used cooling surface. The detailed fabrication of this surface and the arrangement of the tubes will depend largely upon the type of refrigerant for which it is intended.

The simplest construction is where chilled water or brine is used as the refrigerating medium. With direct expansion refrigerant it is usually necessary to provide a special arrangement of headers so that proper distribution of refrigerant through all the surface is obtained. In some cases, ordinary brine coils can be used when operated as a flooded refrigerant system. In some units a combination of a direct spray and a refrigerant surface is used, the spray being directed against the surface. Such systems claim the advantage of air washing together with the maintenance of a clean and effective cooling coil.

It should be noted that when surface coolers are used, adequate protection in the form of filters or at least lint screens are necessary to prevent fouling of the surface from the air borne dirt. Surface not so protected frequently becomes completely matted with lint, grease, and similar dirt.

The sources of refrigeration used with these surface type conditioning units are discussed in Chapter 23. However, they may be divided into the following groups:

1. Direct expansion refrigerant in which the liquid refrigerant is evaporated within the coils of the unit. The vapor from these coils may be recompressed in centrifugal, rotary, or reciprocating type compressors, and the refrigerant again returned to the evaporator coil.

2. Indirect refrigeration by means of:
   a. Cold well water.
   b. Cold city water.
   c. Artificial refrigerated water provided by direct expansion of refrigerant in a water cooler, direct steam jet refrigeration, or by the melting of ice.

Filtering—Air Cleaning

A variety of methods are employed as a means of controlling air purity. In unit systems where filtering alone is considered satisfactory, the degree of filtering varies widely and in proportion to the actual needs. If the air is chiefly recirculated with but little outside air used for ventilation, filtering requirements are largely limited to keeping the coils in a clean and operable condition. Thus such units are frequently furnished with simple lint screens of low resistance and formed of moderately close meshed wire. Where outside air is used for ventilation, more complete filtering of dust particles is necessary and for this purpose, there are a large number of filters available on the market. Some of these filters are of the so-called *throw-away* type, constructed of inexpensive material so that when they become dirty or clogged they may be thrown away and replaced with new ones. All of these filtering methods are described in detail in Chapter 26.

Ventilation

Inasmuch as air purity is one of the factors that constitute true air conditioning, ventilation or the introduction of outside air is an essential part of any air conditioning unit or system. While a unit that recirculates all its air capacity is still considered an air conditioning unit, the better type system provides for the introduction of a certain proportion of outdoor air. In some instances one of several units may operate entirely on outside air, while in other cases only a portion of the air handled by the unit is drawn from out-of-doors. In such cases a damper is provided either in the unit or in the duct connections for controlling the proportion of outdoor air.

Location of Air Conditioning Units

The characteristics of the conditioned space, the building construction, the type of system employed, the duct connection, the accessibility of the unit for servicing, as well as the source of power, piping and refrigeration influence directly the location of air conditioning units.

Fundamentally, there are two types of unit air conditioners. The first type is entirely the self-contained. These units are usually finished in decorative cabinets designed to harmonize with the interior finish of residences, stores, and small commercial establishments. They are, however, sometimes located outside the conditioned space with small lengths of ducts transmitting the conditioned air to the space where it is required. The primary problem in locating units within the conditioned space is to insure proper air distribution. If ventilation air is required, or if the condenser is of the air cooled type, the proximity to a source of fresh air should be considered when locating the units. Units with water cooled condensers should be placed close to the water supply and drain, and care must be exercised that the ambient temperature is never below 32 F, to prevent freezing the water in the condenser.

Proper care should be observed to locate the unit so that all parts are easily accessible in case of trouble.

The second type of unit is the air conditioner which is designed for connection to some remote source of refrigeration. In the smaller sizes these units are sometimes placed within the conditioned space. The

larger sizes are frequently located externally to the occupied and conditioned space and are connected thereto by means of delivery and return ducts. Such an arrangement permits the location of the conditioning unit convenient to either the source of refrigeration or outside air or both. It frequently permits the use of the basement or of space less valuable than that on the level or floor of the occupied zone. The design then approaches that of a *Central System*, (see Chapter 21). Oftentimes the same type of unit may find application in an exposed position for one job and in a concealed location for another. Thus it can be seen that it is not possible to define a unit merely on the basis of its location. Frequently conditioning units are built into the structure or into the architectural design of a room so that they are entirely concealed except for the discharge and return grilles which are designed so as to correspond to the decorative scheme of the room.

Air Distribution

With self-contained units, or any units exposed within the conditioned space, the air distribution is usually through grilles or louvres built into the equipment. The grilles or louvres are usually adjustable to assist in directing the air properly. The discharge of the air from this type of unit is usually directed upward at some angle with respect to the horizontal so that the cool air is not directed at the occupants, but at the same time is carried to the most remote part of the room. In general, the air discharge should be designed to distribute cool air over the entire zone, dropping slowly and returning to the unit below the breathing line and along the floor. The location of doorways, air vents and heat-exposed walls should be carefully observed as they have a marked effect on the direction of air flow and on its uniformity of temperature.

With the suspended type of unit, located within the conditioned space, sufficient outlet air velocity should be provided to give adequate induction and mixing with the room air thereby preventing the immediate dropping of the air stream and resulting objectionable cold drafts.

Where the units are located outside the conditioned space, air distribution is more frequently provided through multiple outlets located in ducts from the conditioning unit. The location of these outlets is quite critical and is influenced both by the building construction, economies of connections and by the distribution of load.

There are a wide variety of outlet types used, and most of these have fixed delivery characteristics, thus requiring careful consideration in their location. Some types of outlets are now available with adjustable vanes thereby permitting some alteration in the delivery of the air stream after installation. This frequently eliminates objectionable down drafts resulting from the impingement of the air stream against posts, pillars, lighting fixtures, and beams.

TYPES OF UNITS

Several types and designs of air conditioning units in production and proposed are available for selection. New designs are constantly appearing, with new improvements, greater capacities, wider range of application and superior construction. It will be impossible to cover in

this chapter the many types of construction on the market. Illustrations of current makes and models will be found in the *Catalog Data Section*. A few typical designs of conditioning units will be described in detail.

An all-year floor type heating and cooling unit for an exposed location and with direct expansion coil supplied with refrigerant from a remotely located compressor is shown in Fig. 12. A cooling coil for use with chilled water may be substituted for the direct expansion coil indicated. The fans below the separate cooling and heating elements deliver the air against deflectors thereby obtaining distribution across the face of the element and preventing condensate from dripping down into the fans. The plate upon which the fans are mounted serves as the drip pan from which the water is conducted to the drain. Separate elements are used for heating and for cooling. Thus this unit may be used automatically for

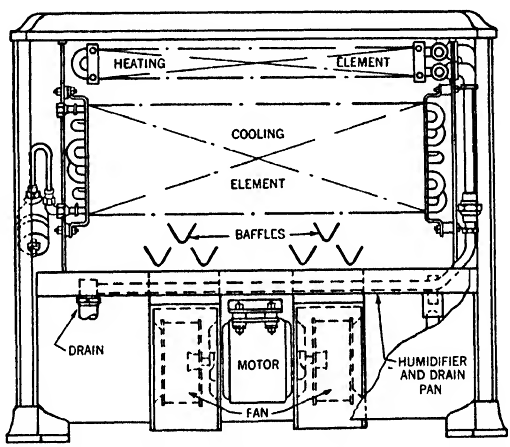

FIG. 12. FLOOR TYPE HEATING AND COOLING UNIT

heating and cooling without manual control. When the unit is used for summer conditioning only, the heating coil may be omitted for the installation. The illustration indicates an evaporative type humidifier and drain pan.

Other units are available in which a target spray humidifier is substituted for the evaporative type thereby providing a unit for summer cooling and dehumidification, at the same time supplying humidification in winter for application in rooms with other existing heat sources.

Still another unit is available in which the fans are mounted at the top of the unit delivering directly through a grille and drawing their air supply through the cooling and heating coils. Other variations in proportion and details of construction of this general arrangement are common. With this type of unit, ventilation is usually provided by means of a separate duct connected to the inlet of the unit.

An entirely different arrangement shown in Fig. 13 places both the air inlet and the discharge at the top of the unit. The fan at one side discharges the air downward to the bottom where it turns and passes horizontally through an atomizing spray air washer. The path then con-

tinues upward through eliminators, a cooling surface and a heating surface before it leaves the unit. With steam or hot water connected to the heating element, tempered water to the sprays and refrigerated water to the cooling element, this unit gives controlled temperature, humidity, air cleaning, and air movement in both summer and winter. Air washing may be connected in summer, or in intermediate season to remove room odors. Excess water is run to waste. Acoustical treatment of the housing and outlet baffles permits installation where noise requirements are exacting.

A common type of suspended type unit for exposed location utilizing a propeller type fan and suitable for summer conditioning only is illustrated in Fig. 14. Such units are equipped with either a direct expansion coil or

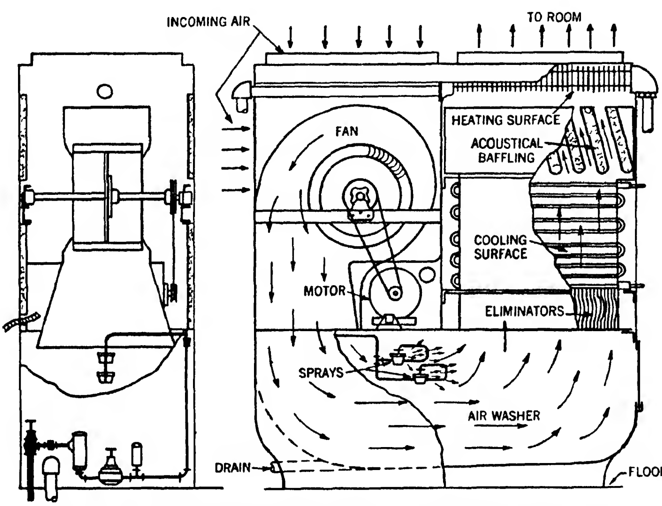

FIG. 13. CONDITIONING UNIT WITH TOP INLET AND OUTLET

one for chilled water or brine circulation. The outer cabinet is sometimes constructed of a well polished wood, but more commonly is made of wood-grained steel or baked enamel and is insulated from the cool air chamber to prevent external condensation. The drip from the coil is collected in an insulated drip pan and carried to a drain. The inlet to the unit is provided with a lint screen to protect the cooling surface. Such units are normally used for recirculation only but may be connected for ventilation through short full-size ducts. Similar units are available with twin housed fans of the same general construction, although usually such fans draw the air instead of blow it through the coils.

A self-contained completely portable cooling air conditioning unit is illustrated in Fig. 15. For the operation of this unit, it is only necessary that it be located adjacent to a window or shaft to which air connections can be made and to plug in the motors to a convenient light socket. In this unit, the conditioned air enters on the side, passing through a grille, filter, and cooling coil and is delivered vertically to the room through a special motor and fan assembly. Refrigeration is furnished by a recipro-

cating compressor driven from a motor located in the base. This compressor utilizes an air cooled condenser. Air is drawn into the base by a fan mounted on the compressor motor, so arranged that the air passes through the refrigeration condenser and is again discharged out through the window connection. A novel feature of this design is that the condensate from the cooling coil is sprayed over the condenser surface and there vaporized, thus eliminating the need for drain connections. One advantage of this type of conditioning unit is that it may be removed from

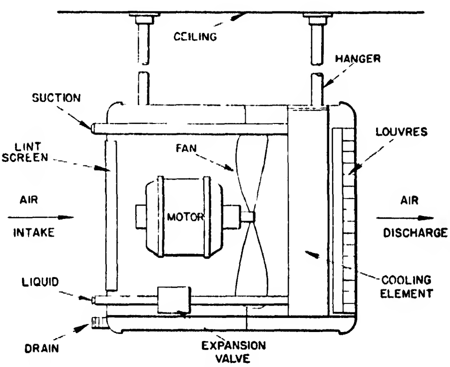

FIG 14 SUSPENDED PROPELLER FAN TYPE COOLING AIR CONDITIONING UNIT

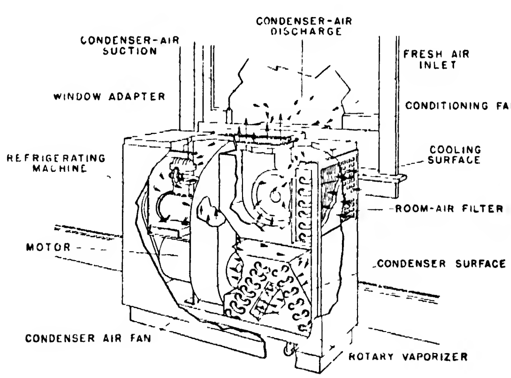

FIG. 15. PORTABLE SELF-CONTAINED CONDITIONING UNIT FOR COOLING

the occupied space during the winter season when cooling is not needed. The portable air-cooled units are now available in sizes up to and including ⅓ hp.

Another type of portable unit for occupied space locations differs from the former in that the compressor is water cooled and a connection to water and drain must be provided in addition to the electric connection. Water and drain lines are carried in a composite hose, especially built for this purpose and connections are usually made to a nearby washbowl. In order to reduce the starting load, one model has two separate motors brought on to the line at delayed intervals thereby decreasing the initial

line surge and reducing light flicker. These water cooled units either eliminate or reduce the need for outdoor air connections. Due to the necessity of water and drain connections they are not as portable as the air cooled type.

A new type of self-contained air conditioning unit has achieved prominence, recently, for application in small commercial establishments. These units range in capacity from 2 to 10 tons, and are designed primarily for use in the conditioned space. They are enclosed in steel casings, designed and finished to harmonize with the interior of commercial establishments. These units generally use water-cooled condensers and are designed for use with 100 per cent recirculated air and free discharge although both fresh air connections and discharge ducts can be used with the units.

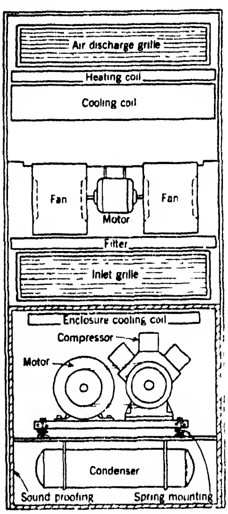

FIG. 16. SELF-CONTAINED AIR CONDITIONING UNIT

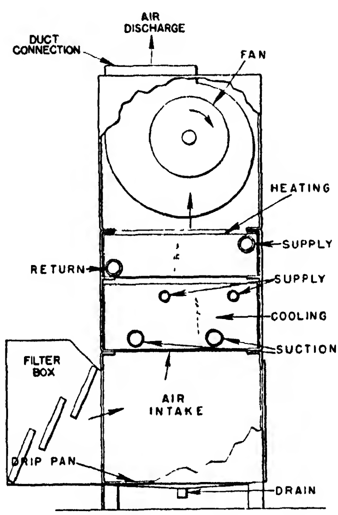

FIG. 17. VERTICAL ALL-YEAR-ROUND CONDITIONING UNIT

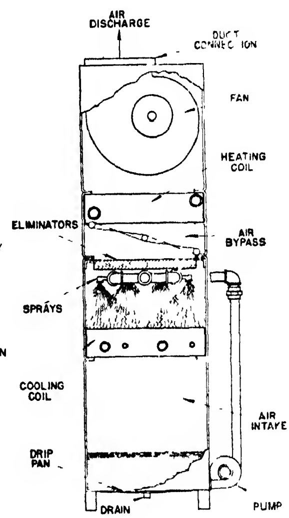

FIG. 18 SPRAY TYPE AIR CONDITIONING UNIT

A typical unit of this type is illustrated in Fig. 16. In this particular unit the air enters a grille located at the front of the unit, passes through a throw-away type filter, then through the cooling coil and is discharged by a blower through an adjustable discharge distributor head. The air is discharged in such a manner to insure good distribution without being directed at the occupants. The air may be discharged from the front only or from the front and either (or both) side. The refrigerating effect is furnished by a reciprocating compressor belt connected to a motor, all mounted on a resilient base to reduce vibration. The heat dissipated by the condensing unit is removed by water cooling. Panels are removable for servicing and replacement of filters. The motor starter and other controls are mounted inside the enclosure and the unit may be operated either as a cooling unit or circulating unit by using the manual switches mounted on the side panel, or it may be automatically controlled by means of a time switch or thermostat.

These units usually provide for the addition of a heating coil as an optional item and some units also have available humidifying equipment as accessory equipment for winter operation.

Remotely located conditioning units vary widely in details of construction. Figs. 17, 18 and 19 indicate one type built-in sections thereby permitting interchangeability of application with a minimum change in parts. The vertical unit shown in Fig. 17 consists of a fan section, housing one or more fans, mounted on a coil section in which are located a heating coil and a cooling coil, which may be built for either direct expansion refrigerant, chilled water, or brine. These two sections are supported on a third or drip pan section. The distributing duct system is attached to the fan outlets and returned fresh air connections are made to the drip pan. A filter box is illustrated attached to the drip pan section. By eliminating the vertical type drip pan and substituting a horizontal drip pan, this unit is converted into a horizontal suspended type conditioning

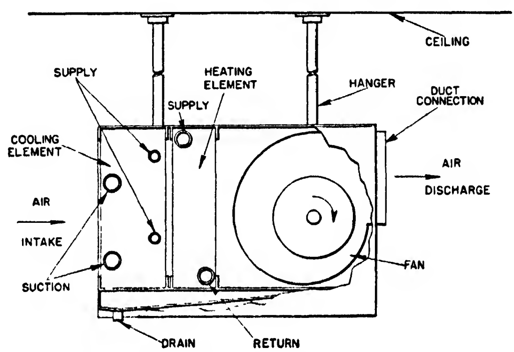

FIG. 19. HORIZONTAL REMOTE TYPE ALL-YEAR-ROUND CONDITIONING UNIT

unit for connection to duct work, both with and without filters, as shown in Fig. 19.

A spray type conditioning unit is illustrated in Fig. 18. This spray type unit, which is similar to the arrangement given in Fig. 17, provides for the complete washing of the air and the cooling coil. For winter operation the spray provides means for humidification. The units may also be obtained with by-pass dampers as shown in Fig. 18, to provide control of cooling in summer and humidification in winter. The spray type unit without the cooling coil may be used for humidification and heat control. This type of air conditioning unit is used in industrial process air conditioning as well as for comfort air conditioning.

Residential Central System Units

The previous figures have largely confined themselves to the illustration of all-year-round or summer conditioning units for office or commercial application. The smaller units have, of course, been applicable to residences. There remains a field of air conditioning units primarily adaptable to residence work. They are in general the outgrowth or adaptation of mechanical warm air systems to conditioning, which are covered in Chapter 20. However, the following illustrations will cover details not included in that Chapter.

In Fig. 20 is shown a conditioning unit which may be operated in conjunction with a hot water or steam boiler. Heat generated in the boiler is supplied to an exchanger which raises the air temperature as it is circulated through the unit. The connection of a cooling coil to a source of refrigeration will provide year-round air conditioning. This type of system is particularly adaptable to a *split* system in which a

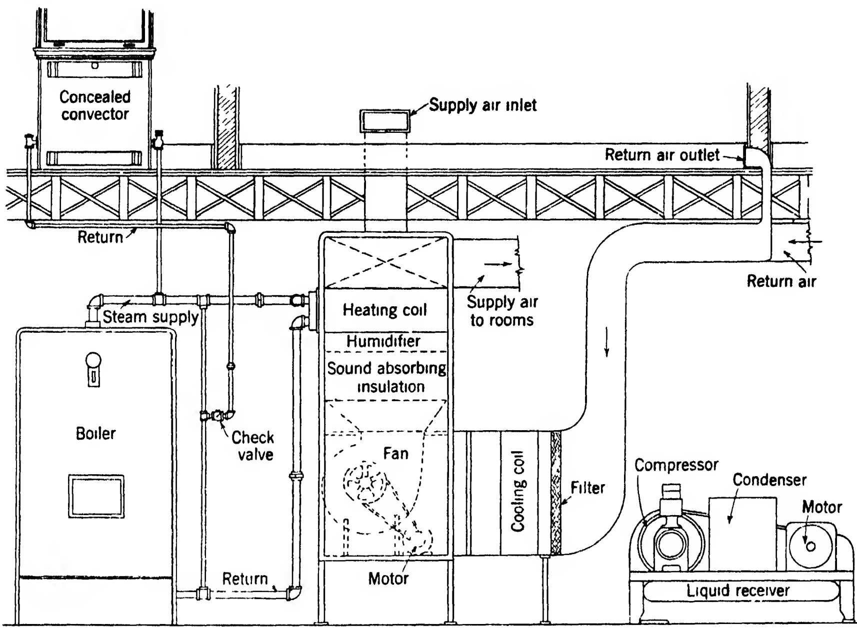

FIG. 20. RESIDENTIAL CONDITIONING UNIT WITH STEAM BOILER

portion of the residence may be conditioned in the winter and summer while the garage, servants' quarters and less freqently used rooms may be provided with radiator or convector heating directly from the boiler in the winter.

Gas-fired winter conditioning units are available equipped with apparatus to filter, heat, humidify and circulate the air in a residence. If a cooling coil is added to the arrangement this unit may also become a year-round conditioner.

A diagram of a direct oil-fired conditioning unit is given in Fig 21. A circulating fan forces the filtered air over a heat exchanger through which the combustion gases from the oil burner are being directed. A cooling section may be placed in the air inlet, with cold water, or refrigerant being circulated through the cooling element.

COSTS

Due to the rapid development of the air conditioning industry and the great progress that is being made each year, it is impossible to give any

cost figures that will be of value. There are, however, certain factors that influence the cost of unit air conditioning installations.

1. Since the cost of the total job involves material cost plus installation labor and since through the use of unitary equipment, material costs can be kept to a minimum, every effort should be made to simplify installation.

2. Self-contained units in the small sizes now available, probably represent the lowest cost individual installations. They have, however, their limitations.

3. The floor type all-year-round air conditioning units for the occupied space with a remotely controlled compressor, heating sources being either the existing heat system or steam connections to the unit, probably afford the lowest cost all-year-round service for most individual rooms.

4. For multiple rooms or offices, the remotely located unit with connecting ducts probably represents the most economical installation. The larger self-contained air conditioners are particularly adaptable to stores, residences and small commercial installations

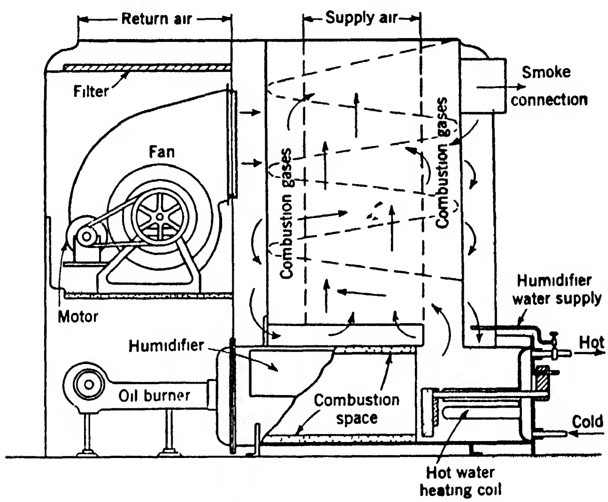

FIG. 21. OIL-FIRED CONDITIONING UNIT

Costs of operation vary widely depending entirely upon the cost of power and water. Water costs in the larger installations are being materially reduced through the use of cooling towers and special types of condensers. The normal expense of operating the cooling system is considerably in excess of that of winter heating both as to the first cost and as to operation. It is difficult to make any comparison of operating costs of cooling vs. heating equipment because the relative operating expense depends upon many factors including climatic conditions; i e , in the South the cost of operating cooling equipment greatly exceeds the operating cost of heating equipment, whereas in colder climates where cooling equipment is used about two months per year, the heating costs are probably higher than those for cooling. The more rapid growth of air conditioning, at present, has been along commercial lines where it has represented an actual profitable investment resulting in increased business returns and where the first cost and operating cost per occupant is considerably less than in the residential type of application where comfort cooling is still considered a luxury by the ordinary home owner.

MISCELLANEOUS UNITARY EQUIPMENT

There are a number of units available which were not covered in the previous discussion that accomplish only one or two of the functions of air conditioning.

Unit Window Ventilators

Window ventilators consist of filters and motor-driven fans enclosed in a cabinet to be mounted on the window sill of homes or offices. These units accomplish ventilation, air cleaning and air circulation. The direction of air discharge is manually adjustable for seasonal operation.

Attic Fans

Attic fans, used during the warm months of the year to draw large volumes of outside air through a house, offer a means of using the comparative coolness of outside evening and night air to bring down the inside temperature of a house.

Because the low static pressures involved are usually less than 1/8 in. of water, disc or propeller fans are generally used instead of the blower or housed types. The fans should have quiet operating characteristics, and they should be capable of giving about twenty air changes per hour. The two general types of attic fan installations in common use are:

Open attic fans, in which the fan is installed in a gable or dormer and one or more grilles are provided in the ceilings of the rooms below. Fresh air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged out-of-doors by the fan. An attic stairway may be used in place of the central grille. It is essential that the roof and the attic walls be free from air leaks.

Boxed-in fans, in which the fan is installed within the attic in a box or housing directly over a central ceiling grille, or in a bulkhead enclosing an attic stair. The fan may be connected by a duct system to the grilles in individual rooms. Fresh air entering through the windows of the rooms below is discharged into the attic space and escapes to the outside through louvers, dormer windows, or screened openings under the eaves.

The locations of the fan, the outlet openings, and the grilles should be selected after consideration of the room and attic arrangement in order to give uniform air distribution in the individual rooms served. If the outlet for the air is not on the side away from the direction of the prevailing wind, openings should be provided on all sides. Kitchens should be separately ventilated because of the fire hazard, and to prevent the spread of cooking odors.

Some typical data on an attic fan installation in an average six room house of frame construction containing 14,000 cu ft and located in the southern part of this country are:

Installation cost	$75 to $400, average $250
Fan data	9000 cfm average, 280 rpm if belt driven, 570 rpm if direct connected, 500 watts input
Operating period	April 15 to October 15, intermittently as weather conditions demand
Power consumption	500 kwh per year for 8 months' operation

Humidifiers

Humidifying units may be installed as part of an air conditioning unit system, or may be installed individually to furnish additional humidity. Fig. 23 illustrates a humidifying unit for installation in connection with a warm air heating system, and as such it is located at the intake of the furnace. The air passes through a lint filter, then through the fans and finally through an air washer or spray system. Surplus spray is eliminated and the air delivered to the air distribution system. In other cases, similar spray type apparatus is used to deliver humidified air through ducts to openings beneath existing radiators in a steam heated residence.

For other steam heated homes, there is a humidifying unit as illustrated in Fig. 24. This unit is normally placed at some central location on the

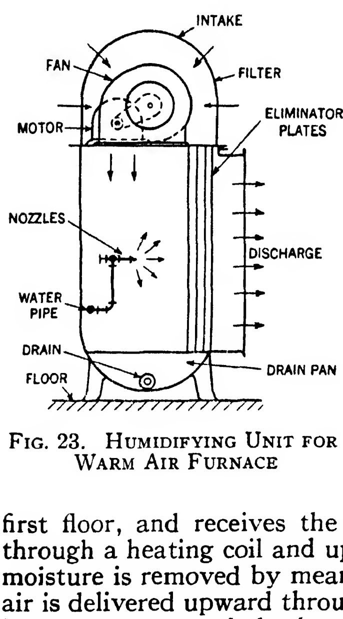

FIG. 23. HUMIDIFYING UNIT FOR WARM AIR FURNACE

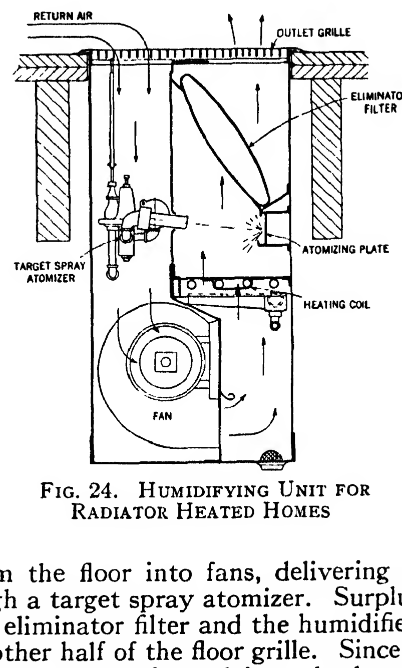

FIG. 24. HUMIDIFYING UNIT FOR RADIATOR HEATED HOMES

first floor, and receives the air from the floor into fans, delivering it through a heating coil and up through a target spray atomizer. Surplus moisture is removed by means of an eliminator filter and the humidified air is delivered upward through the other half of the floor grille. Since a large percentage of the heater capacity is transformed into the latent heat of humidification, this unit does not eliminate any existing steam radiation. It may also be used with hot water systems but its capacity is considerably reduced.

PROBLEMS IN PRACTICE

1 ● Is it satisfactory to use superheated steam in unit heaters?

Superheated steam can be satisfactorily used in unit heaters provided the capacity is based on the saturated steam temperature and not on the total temperature. If unusually high superheat is used, trouble may be experienced from the excessive expansion and contraction of the heating elements.

2 ● Is it satisfactory to install one unit heater as the total load on a coal fired boiler?

Such an arrangement is impractical if the unit heater is started and stopped in keeping with the room temperature. However, if the room temperature controls the steam pressure and the unit heater is arranged to start when there is steam in the mains and to stop when there is no steam in the mains, such an installation will be satisfactory.

3 ● Will a unit heater with a slow speed fan be more quiet than one with a high speed fan?

Quietness is a function of the type, diameter, blade form, and location of the fan, as well as the speed. For a given fan, slower speeds mean less noise.

4 ● Do all unit ventilators introduce a constant amount of outdoor air?

Certain types employ full recirculation except when outdoor air is obtained by throttling the steam valve on the heating element so the proportion of outdoor air to room air is varied. This is a very economical type of unit ventilator but in some communities it cannot be used because of existing laws which require that some fixed amount of outdoor air be introduced whenever the room is occupied. Certain types of units are designed to always take in a minimum quantity of air from the outside and to automatically vary this with the weather.

5 ● Why are metal surface cooling elements instead of liquid spray chambers used in the design of most air conditioning units and cooling units?

The first cost of the surface cooling type of unit is considerably less than the cost of spray type equipment. Further, the requirements of many industrial air conditioning jobs and of all comfort cooling jobs where unit equipment is applicable can often be effectively met with the use of surface type units, with a reduction in the space required for making the installation. Where space conditions are especially limited, the cross-sectional area of the surface cooler can be reduced because the resulting increase in velocity over the coil surface increases the effectiveness of the surface, whereas an increase in velocity through a liquid spray would reduce its effectiveness.

6 ● Why are air conditioning units with metal cooling surfaces not desirable for all industrial jobs?

Wherever unusually close control of relative humidity is required, a spray type unit will prove to be more satisfactory. Relative humidity control and accurate temperature control, however, can be maintained without difficulty with the use of metal surface units

7 ● Why is accurate control of relative humidity with surface coolers more or less complicated?

A surface cooler cannot add moisture to the air, and moisture is removed only when the surface temperature is below the entering dew-point temperature. Any change in condition of the entering air will result in a change in the dry-bulb depression of the leaving air. This change in entering condition requires not only a readjustment of the air volume but also a change in the coil temperature, if accurate control over the relative humidity is to be maintained.

8 ● What in general are the characteristics of operation of a unit using surface coils?

For a constant entering dry-bulb temperature and a constant refrigerant temperature any increase in the entering wet-bulb temperature will produce a rise in the leaving dry-bulb temperature with an accompanying reduction in the wet-bulb depression of the leaving air. The sensible heat removed by the unit decreases and the latent heat increases, while the total heat removed also increases. When the dry-bulb temperature of entering air is increased, with constant refrigerant temperature and constant wet-bulb temperature of entering air, the wet-bulb depression of the leaving air increases, and since it is this depression which determines the maintained relative humidity it must be carefully considered when selecting the unit.

Chapter 23

COOLING AND DEHUMIDIFICATION METHODS

Air Cooling Processes, Dehumidification Processes, Practical Combination Methods, Compression Systems, Mechanical Refrigeration, Steam Jet System, Condensers, Evaporators, Refrigerant Pipe Sizes, Operating Methods, Adsorption System, Absorption System, Evaporative Cooling, Reverse Cycle, Ice Systems

COOLING and dehumidifying are closely related in most air conditioning work. Usually a reduction in both temperature and humidity is necessary to produce comfort. Also, it should be borne in mind that. (1) there is a reduction in moisture content whenever air is cooled below its dew-point, and (2) there is a rise in temperature whenever moisture is removed from air by either adsorption or absorption. Consequently, cooling and dehumidification must in most cases be considered together, not as two separate problems, although each can be accomplished separately.

AIR COOLING PROCESSES

In air conditioning either of two arrangements, or a combination of them, is used to accomplish air cooling The two arrangements are (a) surface cooling, where the air is passed across a cold metal surface; and (b) spray cooling, where the air is passed through a cold liquid spray, usually water. In either case the surface or the spray liquid must be at a temperature sufficiently low so that heat may be removed from the air. Suitable temperatures for the purpose are obtained by the proper use of

1. Refrigeration; or
2. Water from a cold natural source such as a well, from melting ice, or from application of refrigeration; or
3. Evaporative cooling.

The choice of most suitable source of cooling in any specific case will depend on the accompanying circumstances and can be determined only by a thorough analysis made by a competent engineer.

DEHUMIDIFICATION PROCESSES

Dehumidification may be accomplished in any of three ways, or by a suitable combination of them:

1. By cooling the air below its dew-point temperature thus causing a part of the moisture contained to condense and precipitate.
2. By extracting moisture by adsorption.
3. By extracting moisture by absorption.

As in the case of air cooling, the best dehumidification method can be determined only by a complete analysis taking into account all the circumstances of the particular case being considered. In Chapter 2 the nature of the adsorption and absorption processes are explained and the principal properties of the materials used are presented.

PRACTICAL COMBINATION METHODS

As applied in actual practice these several processes frequently have to be combined in order to produce the desired results. Any or all of the three processes of air cooling listed may be combined with any or all of the three dehumidifying processes to produce both air cooling and dehumidification. One form of combination consists of a multi-stage method whereby moisture is removed from the air and then the resulting mixture is cooled. Stage methods are common where dehumidification is accomplished by the use of adsorbent or absorbent substances. Another method, and one in common use, is to combine the air cooling and dehumidification processes into one step. This is made possible by keeping the temperature of the surface or liquid spray used for cooling below the dew-point temperature of the air to be conditioned It is the method most commonly associated with comfort air conditioning in current practice. Still another general method consists of what may be called a parallel-flow method wherein the cooling or dehumidification, or both, may be performed by splitting the air stream, performing the process on part of it and then bringing the two parts back together again.

Obviously with so many possible combinations much leeway is left to the designer to determine what shall be done in a practical case. The remainder of this chapter is devoted to a discussion of some of these possible practical methods and the equipment used in applying them. Space does not permit discussing all the great variety possible and only those in reasonably frequent use are included here. Others will occur readily and can be analyzed in a fashion similar to those here treated.

COMPRESSION SYSTEMS

Comfort air conditioning imposes requirements on refrigeration equipment not usually found in general cooling applications so that specially designed apparatus is often required to replace that normally used for industrial cooling. Standard equipment can be adapted to meet air conditioning requirements but extreme care must be taken to determine the limits of its applicability.

In industrial or process cooling systems the load is fairly constant,

noise in operation is not of paramount importance, space is available or relatively cheap, and the cooling system is to a great extent separate or independent of other mechanical equipment. By contrast, air conditioning for space cooling and comfort work in office buildings, theaters and places of public assemblage requires special consideration of all these factors. Space in public buildings is limited, noise interferes with the occupants and the cooling equipment must be adaptable to the other air handling apparatus. Most important, the load fluctuates tremendously and is seasonal.

Types of Compressors

There are many different types of compressors, a number of refrigerants, different types of evaporators, condensers and arrangements of cycle and each type has its particular place in usage. Compressors generally used are of the following types·

1. Reciprocating compressors using a volatile refrigerant.
2. Centrifugal compressors.
   a. Using a volatile refrigerant.
   b. Using water as a refrigerant.
3. Rotary compressors using a volatile refrigerant.
4. Steam jet or vacuum systems using water as a refrigerant.

Reciprocating compressors are generally used with any low pressure refrigerant such as dichlorodifluoromethane, monofluorotrichloromethane, methyl chloride, ammonia and sulphur dioxide. These compressors have been developed to a point where their efficiency is high and their operation very satisfactory. Relatively low speed operation makes them desirable for general use in large installations. Generally they are of two types, vertical and horizontal either single or double acting. The horizontal double-acting compressor is not generally used in air conditioning, except when carbon dioxide is used as a refrigerant in the larger industrial systems. Vertical, single acting, encased crank, reciprocating compressors of the uniflow type with valves in the pistons have proven reliable and are used in capacities from 1 hp to more than 100 hp. At present reciprocating compressors are used with more refrigerants than any other type of compression unit. When carbon dioxide is used as a refrigerant, a reciprocating compressor is required because of the extremely high pressures and the relatively high ratio of compression.

Centrifugal compressors using monofluorotrichloromethane, methylene chloride or water vapor can theoretically be used with any of the other refrigerants, but the resulting loss in efficiency with the higher pressure gases limits the centrifugal compressor to the refrigerants cited. At the present time centrifugal compressors are limited to air conditioning systems of a minimum of about 50 tons. Centrifugal compressors are usually built in two or more stages where the compression ratio is high and their design follows closely that of any other centrifugal equipment such as is found in general service pumps and fans.

Rotary compressors are expanding in use due to the development of new refrigerants. These units are of four common designs, consisting of rotating elements generally referred to as centrifugal, eccentric, gear and blade types. The rotation of the shafts and blades traps the refrig-

erant vapor between the moving elements and the case and delivers it to the condenser at the required pressure. The rolling together of the impellers as in the case of the gear compressor prevents the return of the refrigerant vapor to the low side of the system.

Steam jet compressors which are particularly adapted to large tonnage installations are simple, compact, have no moving parts and produce practically no vibration. However, they are not economical for water temperatures much below 40 F or where the cost of generating steam is higher than the cost of operation with other prime movers.

MECHANICAL REFRIGERATION

While the mechanical refrigeration systems differ in the methods used for compression of the refrigerant vapor, they are fundamentally similar.

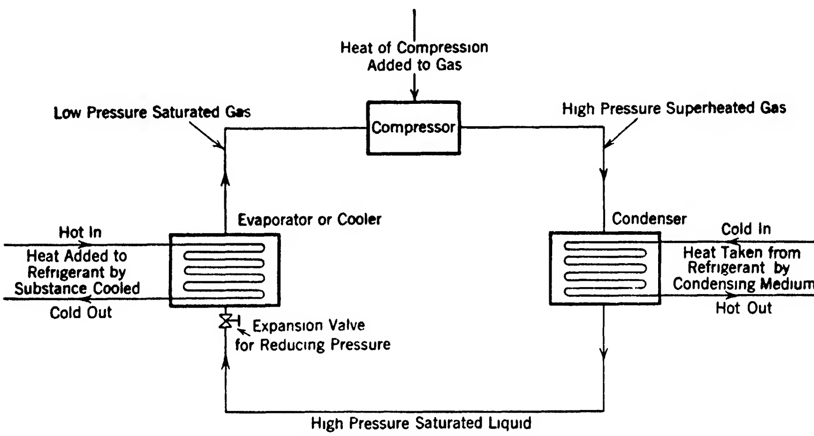

FIG. 1. MECHANICAL REFRIGERATION SYSTEM

Refrigerant vapor usually saturated or slightly superheated, is drawn into the compressor as diagrammed in Fig. 1. It is then compressed and discharged at a higher pressure to a condenser. The vapor is condensed as it contacts a heat transfer surface over which is flowing a cooling medium such as water, air or a combination of the two. The liquid refrigerant flows to the evaporator through an expansion valve which reduces its pressure and regulates its flow. In the evaporator the refrigerant absorbs heat from the medium which is to be cooled. When this medium is water or brine, the evaporator is known as a water or brine cooler and the refrigeration system, if used for air cooling, is known as an indirect system. When the medium cooled is air, the evaporator is known as a direct expansion cooler and the system is known as a direct expansion system.

Fundamentally, the function of the system is to absorb heat at one temperature and *pump* it to a higher temperature, where it may be removed by an available cooling medium. In order to conserve refrigerant, virtually all refrigeration systems are completely closed and the same refrigerant is recirculated.

Theoretical Mechanical Refrigeration Cycle

The complete mechanical refrigeration cycle may be illustrated on the temperature-entropy diagram, and also on the pressure-volume diagram both of which are shown in Fig. 2.

Considering the theoretical cycle, saturated vapor is drawn into the compressor at *a* and compressed at constant entropy (adiabatically) and then delivered to the condenser at *b*. Condensation occurs at constant temperature T_2 from *b* to *c* with a contraction from the vapor to the liquid volume. The line *cd* represents cooling from the temperature of the condenser to that of the evaporator by an external cooling means. At the same time, the pressure is lowered to P_1. Evaporation then occurs from *d* to *a* at temperature T_1, completing the work cycle *abcda*. Since no external means of cooling the refrigerant liquid is normally available, the

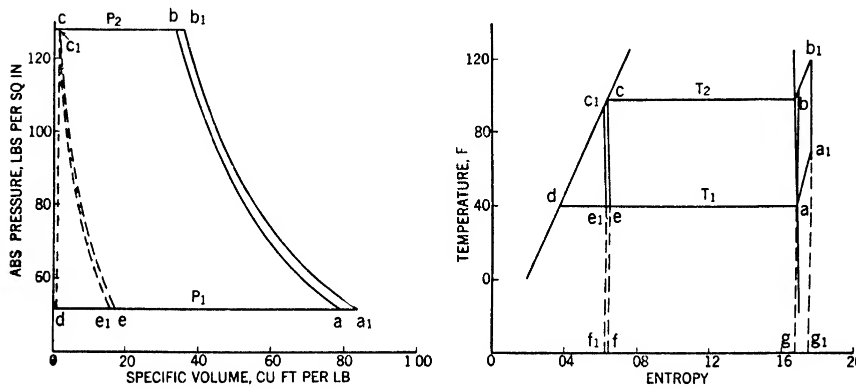

FIG. 2. THEORETICAL DICHLORODIFLUOROMETHANE (F_{12}) CYCLES

cooling is generally accomplished by evaporation of a portion of the refrigerant. Since the work of expansion is usually used up as friction in the expansion valve, this process is assumed to be carried on at constant total heat, as represented by the line *ce* on the temperature-entropy diagram. Thus the refrigerating effect is represented by an area *eagfe*. While the normal theoretical cycle starts with saturated vapor, operation is common at a condition of superheated vapor (as at a_1). Moreover, expansion may start either with a mixture of liquid and vapor or with a sub-cooled liquid, as at c_1, with expansion to e_1. It is obvious that this latter is desirable as it increases the refrigerating effect. Area $a_1b_1cdaa_1$ represents the work of such a superheated cycle, while the area $e_1a_1g_1f_1e_1$ represents the refrigerating effect of the cycle with superheated vapor and sub-cooled refrigerant liquid.

It will be noted on the pressure-volume diagram the volume of the saturated liquid is indicated by a dotted line close to and parallel to the ordinate.

In the discussions in this chapter a slight error is introduced by not including all of the work of pumping the liquid from the low to the high

pressure. This occurs because the liquid line is not a line of equal pressure but of saturation pressures. The error in work per pound of refrigerant figured from total heats, which should be added to the indicated figures is roughly the specific volume of the liquid at the lower pressure and temperature multiplied by the pressure difference in appropriate units. This error may become of some importance in calculations involving carbon dioxide or in problems involving the liquid of any of the refrigerants, as in figuring expansion valve orifices.

Theoretical Work per Pound

The temperature-entropy and pressure-volume diagrams are based on one pound of the refrigerant. Likewise, the theoretical work and the refrigerating effects are conveniently based on a pound of refrigerant. The compression work per pound may be found by several methods.

The temperature-entropy method starts with state point a. Since the quality of a is known, the heat content of the vapor H_a is known, and also the entropy S_a. Since point b lies near the saturation curve it is customary to assume $S_a = S_b$ and with T_2 given, H_b can be determined. If W = work in foot-pounds per pound of refrigerant, then

$$W = (H_b - H_a) \times 778 \tag{1}$$

The pressure-volume method starts with state point a, whose pressure and specific volume are known. The work of compression is the adiabatic work of compression from P_1 to P_2, plus the work of expelling the vapor at constant pressure P_2 minus the external work of evaporation of the vapor to volume V_1 at pressure P_1.

$$W = \frac{n}{n-1} \times P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \tag{2}$$

It is frequently helpful to think of the compression of the vapor in terms of head. The head may be likened to a vertical column of vapor in which is located the vapor to be compressed. The compression occurs when the vapor is moved down from a level corresponding to P_1 to a new level corresponding to P_2, in equilibrium with the surrounding vapor. If this process is carried on isentropically, the result will be the same as indicated previously. Then if h is the head in feet,

$$W = h \tag{3}$$

This relationship may easily be seen from the fact that a small difference of head dh divided by the specific volume of the vapor V is equal to the increment of pressure difference dP.

Head is very useful in considering the performance of centrifugal compressors, which merely substitute a centrifugal for the gravity head. It is also useful in considering problems of fluid flow. In these problems, the head per degree can be obtained either by direct calculation or approximately by dividing the total head by the temperature difference $T_2 - T_1$. The velocity head loss can then be calculated in degrees, using the customary formula $V^2 = 2gh$.

Refrigerating Effect per Pound

The refrigerating effect per pound is computed by the same method, regardless of the type of refrigeration system. The solution is indicated on the temperature-entropy diagram of Fig. 2. Assuming that the vapor leaving the evaporator is saturated, the refrigerating effect in Btu per pound is obtained by subtracting from the heat content of the vapor at temperature T_1, the heat content of the liquid at T_2, or if the liquid is sub-cooled, the liquid temperature.

Thus, the refrigerating effect in Btu per pound is equal to

$$H_a - H_c - H_a - H_e \tag{4}$$

If the vapor entering the compressor is superheated or supersaturated, a correction in the heat of the vapor is made accordingly.

The unit of refrigeration is the *ton*, based on the latent heat of fusion of one ton of ice in 24 hr. Thus one ton = 200 Btu per minute = 12,000 Btu per hour.

Coefficient of Performance

The coefficient of performance of a refrigeration system is the ratio of the refrigerating effect to the work of compression, both expressed in the same units.

The ideal or Carnot coefficient of performance depends upon the temperatures T_1 and T_2 in much the same way as the ideal efficiency of a steam engine depends upon its working temperature, with an inverse relationship.

$$\text{Ideal C. of P.} = \frac{T_1}{T_2 - T_1} \tag{5}$$

Evidently the smaller the compression range, the less power will be required to produce a given refrigerating effect.

The theoretical coefficient of performance of actual refrigerants is always less than the ideal due to the tendency of most refrigerants to superheat when compressed, and due to the heat of the liquid which must be removed. The cycle efficiency is the theoretical C. of P. divided by the ideal for the same temperatures. The cycle efficiency usually changes as the compression temperatures change.

Practical Cycle

Fig. 3 illustrates the pressure-volume and temperature-entropy diagrams for an actual cycle. These diagrams are based upon the compressor receiving vapor superheated and upon sub-cooling of the liquid going to the evaporator. The theoretical cycle is $a_1b_1cc_1e_1a_1$. However, the vapor during compression actually follows line a_1b_2 due to superheating as a result of the inefficient work of compression. The theoretical work of compression is $a_1b_1cda_1$. Added to this is the area $b_2b_1g_1h_1b_2$ on the temperature-entropy diagram which represents the inefficient work of compression (assuming no compressor heat losses). The sum of these areas represents the total work of the compressor per pound of refrigerant, and the ratio of theoretical cycle work to the actual work represents the overall efficiency. It should be noted that area $a_1b_2b_1a_1$ is considered as part

of the inefficient work and is commonly termed the superheat loss. The refrigerating effect per pound is the same for the practical as for the theoretical cycle, working with the same sub-cooling of liquid and superheating of vapor, that is, area $e_1a_1g_1f_1e_1$.

Sources of loss which are usually recognized as reflected by the overall efficiency referring particularly to reciprocating and rotary systems, are as follows:

1. The superheat loss.

2. A pressure loss to and from the cylinder of the compressor. (The line pressure drop between the compressor and the evaporator and condenser, respectively, is usually taken into account separately in the design of the refrigeration system).

3. Leakage loss through valves and past pistons is quite small in most compressors.

4. With an oil soluble refrigerant, there may be an absorption loss due to absorption and re-evaporation of refrigerant in the oil of the cylinder.

5. Mechanical losses are always present and are usually a large part of the total

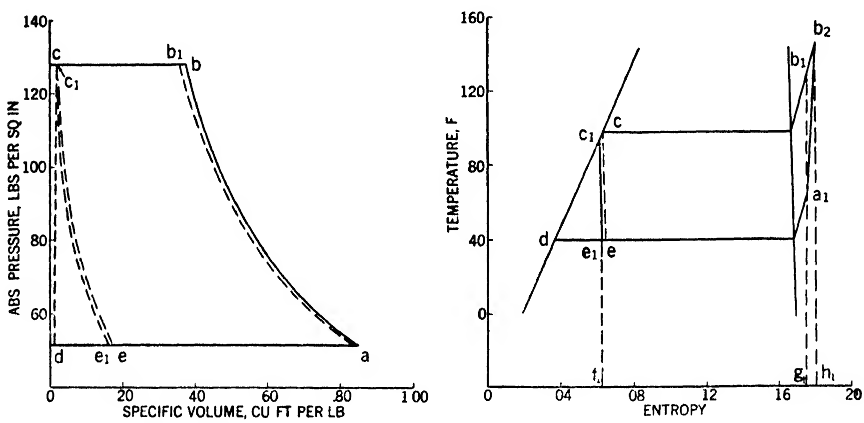

FIG. 3. PRACTICAL DICHLORODIFLUOROMETHANE (F_{12}) CYCLES

Reciprocating and rotary compressors always take in less vapor than that which corresponds to the displacement. The overall volumetric efficiency is the ratio of the suction vapor volume to the piston displacement. On reciprocating compressors part of the loss is the re-expanded volume, at suction pressure, of the vapor which was in the clearance volume. This is expressed by the following equation:

$$\text{Volumetric Efficiency} = 1 - \frac{v_c}{v_d} \times \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right] \quad (6)$$

where

v_c = clearance volume
v_d = cylinder displacement volume.

The balance of the overall volumetric efficiency is known as the superheat volumetric efficiency even though it includes some other sources of capacity loss.

The mechanical efficiency of a reciprocating and rotary compressor must be multiplied by the superheat volumetric efficiency to give the overall efficiency of the compressor.

$$\text{Eff.}_{\text{overall}} = \frac{\text{Vol. Eff.}_{\text{overall}}}{\text{Vol. Eff.}_{\text{reexp.}}} \times \text{Mech. Eff.} = \text{Super. Vol. Eff.} \times \text{Mech. Eff.} \qquad (7)$$

Normally, the volumetric efficiency of a compressor varies with the ratio of compression, while the mechanical efficiency remains virtually fixed. Good standard practices for dichlorodifluoromethane compressors are:

	Low comp. ratio = 2.5 to 1	High comp. ratio = 5 to 1
Vol. Eff.$_{\text{reexp.}}$	94 to 96 per cent	88 to 92 per cent
Vol. Eff.$_{\text{super.}}$	75 to 85 per cent	73 to 77 per cent
Vol. Eff.$_{\text{overall}}$	70 to 81 per cent	64 to 71 per cent
Mech. Eff.	75 to 85 per cent	75 to 85 per cent

These values are for one ton or larger compressors. Part of the difference expresses the change with capacity. With other refrigerants and other types of compressors there will be some further variation.

STEAM JET SYSTEM

The steam jet type of compressor, under certain circumstances, is desirable for use in air conditioning. The power used for compressing the refrigerant is steam, taken directly from the boiler, thus eliminating the mechanical losses of manufacturing electric current. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiencies of the equipment are somewhat lower than those of the positive mechanical type of compressor; also the condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures, and steam ejectors of the type commonly used in power plants for various processes will produce the necessary low absolute pressure to cause evaporation of the water.

A diagrammatic representation of a typical steam ejector water cooling system is shown in Fig. 4. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam, and as this requires heat and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point, determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per hour per ton of refrigeration developed. The remainder of the water at the desired low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed in the evaporator, plus any entrained air taken out of the water circulated, to a somewhat higher absolute pressure, and the vapor and air mix with the impelling steam on the discharge side of the jet. The total mixture of

entrained air, evaporated water, and impelling steam is discharged into a surface condenser at a pressure which permits the available condensing medium to condense it. The resulting condensate is removed from the condenser by a small pump, from which it can be discharged to the sewer

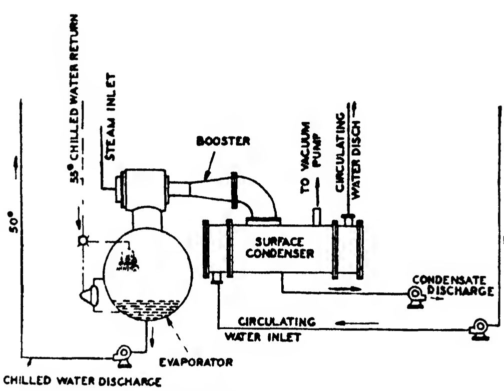

FIG. 4. STEAM EJECTOR COMPRESSION REFRIGERATION SYSTEM

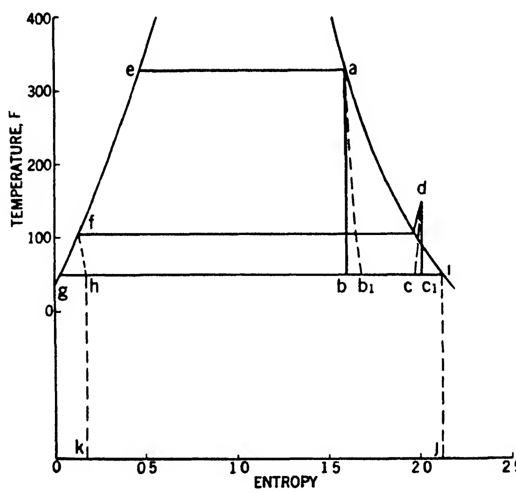

FIG. 5. STEAM EJECTOR TEMPERATURE-ENTROPY DIAGRAM

or returned to the system in the form of make-up water, or part of it may be returned to the boiler feed pump.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A small secondary condenser, of course, is necessary to condense the steam used in the secondary jet.

The performance of the steam ejector may be studied theoretically by the use of the temperature-entropy diagram, Fig. 5. Unlike its usual

application, however, the amount of working fluid is different for one portion of the cycle than for the other. Dry saturated steam under high pressure, for example 100 lb per square inch gage, at *a*, is expanded through the nozzle of the steam ejector. With 100 per cent efficiency, the expansion would occur along isentropic line *ab*. Actually, however, most nozzles are only about 90 per cent efficient, the real expansion being along the line ab_1. Since the exact path of the line ab_1 is not known, the work area is normally assumed by using the isentropic giving a work area *abgea*. The velocity at the mouth of the nozzle may be determined in the usual manner using this area and the velocity coefficient of the nozzle.

At the evaporator pressure or slightly below, the vapor from the nozzle mixes with virtually dry saturated vapor from the evaporator. An impact loss also occurs at this point due to the mixture of vapors at different velocities. This results in bringing the state point of the mixture to *c*. Compression then occurs along the line *cd*, the work of compression per pound being *cdfgc*. In computing the work area, however, the point *c* is not actually known. Therefore, the work area c_1dfgc_1 is used in expressing the efficiency of the ejector, the line c_1d being an isentropic. The losses are expressed by nozzle efficiency, impact loss and diffuser efficiency. The work of compression, however, is performed on the mass of the mixture. Thus, the available work is reduced in proportion to:

$$\frac{M_{\text{primary}}}{M_{\text{mixture}}}$$

The impact loss is commonly determined from the formula:

$$MV_{\text{primary}} + MV_{\text{secondary}} = MV_{\text{mixture}} \tag{8}$$

Common efficiencies for commercial ejectors are: nozzle efficiency 90 per cent, diffuser efficiency 60 to 70 per cent. Customary steam rates in pounds per ton are approximately as follows:

Evaporator temp	50 F	Steam press	100 lb per sq in	Steam press	12 lb per sq in
Condenser temp	105 F	Steam rate	30 lb per hour per ton	Steam rate	45 lb per hour per ton
Evaporator temp	40 F	Steam press	100 lb per sq in.	Steam press	12 lb per sq in
Condenser temp	105 F	Steam rate	40 lb per hour per ton	Steam rate	70 lb per hour per ton

CENTRIFUGAL VAPOR VACUUM SYSTEMS

The centrifugal vapor vacuum system functions on the same general principle as the steam jet system, except that a centrifugal evacuator is used to produce the low absolute pressure instead of the velocity of the steam through a jet. Less condenser water is required and a vacuum pump is employed instead of a steam jet purge.

CHARACTERISTICS OF COMPRESSION SYSTEMS

The different types of compression systems have quite different characteristics of capacity and power with varying evaporator temperature and with varying condenser temperature, as will be seen from curves in Figs. 6 and 7.

The capacity of the reciprocating and rotary compressor varies slowly with a change of evaporator temperature, and the variance of power

requirements, in the air conditioning range of operation, is small for a change of evaporator temperature. On the other hand, the capacity and power of the centrifugal machine vary rapidly, and the capacity of the steam ejector also varies considerably. Thus, both these latter types tend to be more nearly self-regulating than the reciprocating and rotary compression type. On the other hand, the operating range of the latter near standard capacity is superior. Although the capacity of the reciprocating and rotary compressor is little affected by the condenser temperature, the power of the compressor is greatly affected, while the reverse is true

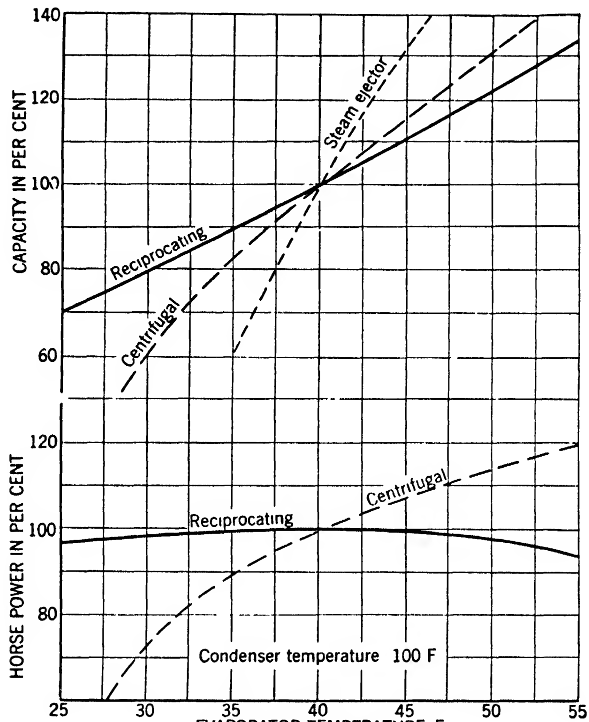

FIG. 6. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

for the centrifugal compressor. As previously indicated, the condenser temperature has no effect on the capacity of the steam ejector type of compressor until a certain point is reached, beyond which the capacity is zero. The steam consumption for the performance characteristic curves shown in Figs. 6 and 7 remains constant for all evaporator and condenser temperatures.

Steam jet refrigeration requires from 3 to 10 times as much condenser water as other types of mechanical refrigeration, but its capacity is not effected by condensing water temperature as long as the water does not greatly exceed 100 F. Consequently, steam jet systems are well suited to those applications where condensing water is cheap, or where con-

densing water is rather high in temperature. From Fig. 6 it is evident that steam jet refrigeration is better suited for use with evaporator temperatures *above* rather than below 40 F.

CONDENSERS

Condensers used in connection with refrigerating equipment for absorbing the work of compression are of three general designs: (1) air, (2) water, and (3) evaporative.

Air Cooled

Air cooled condensers are seldom used for capacities above 3 tons of refrigeration, unless an adequate water supply is extremely difficult to

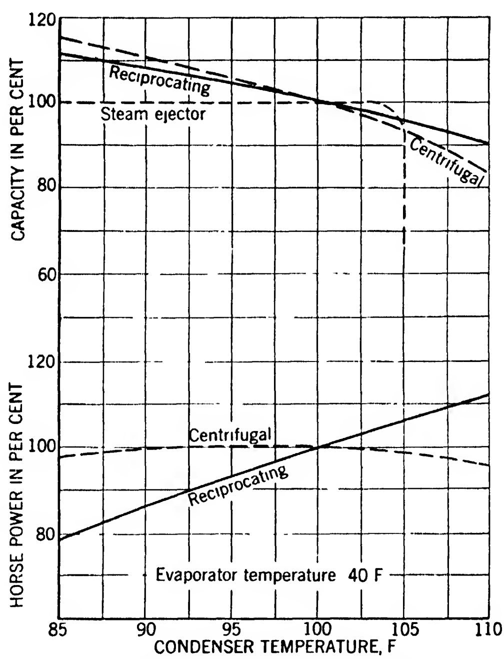

FIG. 7. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

obtain, as for instance in railway air conditioning.[7] Even on fractional tonnage installations, air is used as the condensing medium only where water is expensive or where simplicity of installation warrants the higher condensing pressure, and consequent higher power costs than can be obtained using water as the condensing medium.

The conventional air cooled condenser consists of an extended surface coil across which air is blown by a fan. The hot refrigerant gas enters the coil at the top and as it is condensed flows to a receiver located below the condenser. Air cooled condensers should always be located in a well ventilated space so that the heated air may escape and be replaced by cooler air.

The principal disadvantages to air cooled condensers are the power required to move the air and the reduction of capacity on hot days. This loss of capacity due to high condensing pressures on warm days requires that equipment of reduced capacity be selected to meet the peak load. Thus, at normal loads the equipment is oversized.

Water Cooled

Water cooled condensers are usually of the double pipe type or the shell and tube type. Double pipe condensers are arranged so that water passes through the inner of two concentric pipes, and refrigeration circulates through the annular space in the outer pipe. Where possible, there should be counter-flow of the refrigerant and the condensing water to maintain maximum temperature differences.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is, therefore, necessary to determine a balance so that the quantity of water insures economical compressor operation.

Because there is a decided tendency to conserve the water in city mains and because most large cities are restricting the use of water for air conditioning and refrigeration equipment, it is often necessary to install cooling towers or evaporative condensers. Cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet not only the maximum load at normal conditions, but also the maximum load at abnormal condensing water temperatures. If properly designed, this makes little difference in the efficiency of operation throughout the year except at those times when the condensing water temperature is highest. As this occurs only for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs.

The cooling tower has a certain advantage over the use of water from the city mains in that the temperature of the condensing water varies directly with the outdoor temperature and, as pointed out, the refrigeration load also varies with this temperature. Certain economies are possible when a cooling tower is used which cannot be achieved by the use of condensing water from city mains, even where the city water temperature is extremely low. Normally, the lowest city water temperature met during the summer months is from 65 to 70 F. This temperature range takes place for the entire cooling period, regardless of the outdoor temperatures. With the cooling tower, the temperature of the condensing water may rise to 80 or 85 F under maximum conditions, but under less than maximum conditions the temperature of the water leaving the cooling tower drops considerably, and it has been established that 50 per cent of the time the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water, for the same periods, varies from 65 to 75 F. When the outdoor wet-bulb temperature drops below 60 F, which occurs approximately 30 per cent of the time, the condensing water temperature is still lower. The cost of water used for condensing is negligible, as the only water required is that used to make up the loss by evaporation in the cooling tower itself. Refer to the section on cooling towers in Chapter 25.

Evaporative

Due to the high cost of city water for condenser purposes and due to ordinances in some localities prohibiting the discharge of large quantities of such water into the sewage systems, there has been developed a condenser which uses a minimum amount of water on a finned surface, cooling it to approximately the wet-bulb temperature of the surrounding atmosphere.

A diagram of a typical small evaporative condenser is shown in Fig. 8 which includes a fan that forces the air over a finned tube condenser coil to the outside atmosphere through a connecting duct. A fine spray of water keeps the coil surface wetted. The hot refrigerant gases enter the top of the condenser coil and the liquid collects in the receiver below

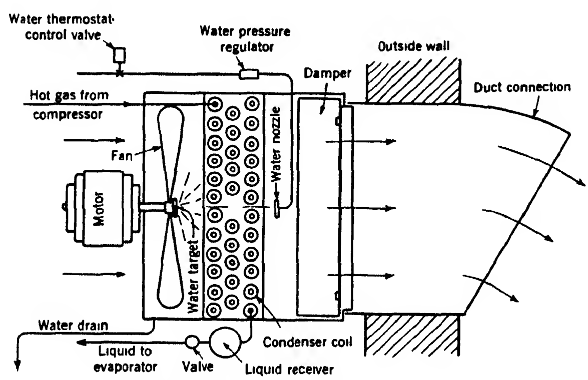

FIG. 8. DIAGRAM OF EVAPORATIVE CONDENSER

the condenser unit. A thermostatic control valve and pressure regulator are arranged on the condenser water supply for regulating and adjusting the water flow to the nozzle. It is desirable that a condenser of this type be located near the compressor.

Units of this design are available in sizes ranging from 2 tons of refrigeration up to about 6 tons. About 10 per cent as much water is required for this unit as is normally used in a shell and tube type condenser. For larger capacities evaporative condensers are usually equipped with a pump for recirculating the water. Such units are available in capacities ranging from 5 to 40 tons of refrigeration. They require no more water than a cooling tower, i.e., from 3 to 5 per cent as much water as required where all water is taken from the city main.

EVAPORATORS AND COOLERS

The types of coolers used in connection with air conditioning work fall into three general groups. The *first*, is the direct cooling of water; the *second*, direct cooling of air; and the *third*, cooling of brine for circulation in a closed system, which can cool either water or air. One method of the direct cooling of water is to install direct expansion coils in the spray chamber so that the water sprayed into the air comes in direct contact

with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray water cooler is the shell and tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor the suction temperature or the temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigeration increases due to the higher compression ratio, but there are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used, due to fire or other risks especially in densely populated areas, the brine can be cooled in an isolated room or building and then be circulated through the air conditioning equipment in perfect safety because it is used to cool the water or air, without any possibility of direct contact between the air and refrigerant.

REFRIGERANT PIPE SIZES

The selection of proper pipe sizes and frictional pressure losses varies with the installation and the capacity of the system. Generally the suction piping should be selected so that the pressure loss is between 2 and 3 lb per square inch. The pressure drop in liquid lines should be maintained so as to permit no vaporization in the pipes with limiting pressure drops not to exceed 5 lb per square inch. Hot or discharge gas

lines should be limited to approximately 4 lb per square inch pressure drop. All pressure drops mentioned are total system losses and not only include the piping losses, but also the pressure losses in the valves, fittings and coils.

For installations involving piping connections between compressors and evaporative or other remote condensers, pressure drops for discharge or hot gas lines may be referred to in Table 1. Pressure losses in liquid refrigerant lines of various sizes and capacities are given in Table 2. Pressure drops of suction refrigerant pipe lines at varying capacities and refrigerant temperatures may be referred to in Table 3. All tables are for 100 ft of pipe, including an average number of fittings, and for other lengths the losses are proportionate. Allowances should be considered for drops through control and regulating valves which must be added to the other pipe losses to determine the total drop. All copper pipe referred to in these tables are of type *L* wall thickness and are designated by outside diameter.

OPERATING METHODS

There are various methods of designing and operating air conditioning systems to obtain economical results. Peak outside conditions seldom exist for periods of greater than 3 hr. On many installations there is a peak internal load which may or may not coincide with the peak outside conditions. Thus, each application must be carefully analyzed by the engineer, and the proper equipment installed to satisfy the requirements. Adequate automatic controls should be installed for any system selected.

Where there are a number of small rooms to be conditioned, as for example, a group of hotel bedrooms where the load varies with occupancy and exposure, it may be best to employ individual room units, each with its own control. These individual units may be of the self-contained type (condensing unit, evaporator, fan and controls all in one cabinet) or of the remote type with the condensing units located outside of the room. In some cases it is good practice to use one large condensing unit to serve a group of room evaporator units. Where this is done, the condensing unit must have some type of control which will prevent freezing evaporator temperatures when only a portion of the evaporators are in use. This can be accomplished by means of a back pressure regulating valve which maintains the evaporator pressure at a safe limit, but allows the crank case pressure to fall. Other methods of accomplishing the same result are the use of a variable speed compressor or the use of a partial by-pass from the high side to the low side of the compressor. Any of these three methods of lowering the condensing unit capacity drop the operating cost at the reduced loads, but the operating cost per ton is higher.

Central Distribution Systems

On air conditioning systems using duct distribution, the same general types of control are employed to meet the varying load conditions, i.e. (1) the system may consist of several condensing units and evaporators which are cut in or out, depending upon the demand, or (2) condensing

TABLE 1. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE DISCHARGE OR HOT GAS LINES[a]

Capacity BTU per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[b]									
	Line Sizes, Inches									
	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8
10,000	2 3	1 0	0.6							
15,000	4.9	2.0	1.0							
20,000	8 5	3.4	1.7	0 6						
25,000		5 3	2 6	0 9						
30,000		7 5	3 6	1.2	0 5					
40,000			6 4	2.1	0.7					
50,000			9.8	3 1	1.0	0 5				
60,000				4 4	1.3	0 7				
70,000				6 0	1 9	0 9				
80,000				8.0	2 5	1 1				
90,000				10.2	3.1	1 4				
100,000					3 8	1.7	0 5			
125,000					6 0	2.6	0 7			
150,000					8 5	3 8	1 0			
175,000					11 6	5.1	1 3			
200,000						6.7	1 7	0 6		
250,000						10.4	2 6	0.9		
300,000							3 7	1.2	0 5	
400,000							6 7	2.2	0 9	
500,000							10 5	3 5	1.5	0 7
600,000								5 0	2.1	1 0
800,000								9.0	3 8	1 8
1,000,000									5 8	2.9
1,250,000									9 5	4.4
1,500,000										6 4
2,000,000										11 3

[a]Soft annealed copper tubing up to and including 3/4 in. outside diameter. Hard copper pipe 7/8 in outside diameter and larger.

[b]Length of tubing includes the average number of fittings.

unit capacity may be reduced by using back pressure regulating valves, by-pass valves, or variable speed compressors.

Another method of providing for economy of operation is to have storage capacity which can be utilized during the peak period. The refrigerating system can be operated for a longer period at maximum efficiency with tanks to store cold water or brine for supplementing the actual output of the refrigerating equipment. However, storage tanks require space and extra apparatus, which increase the cost of the entire system, and further, it is difficult to determine the exact size of the compressor because of the other variables which enter the problem. Depending upon the availability of storage space, the compressor may be designed for any reasonable percentage of the maximum load. On this basis of selection, the smaller the compressor, the larger the storage space, and vice versa.

TABLE 2. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE LIQUID REFRIGERANT LINES

CAPACITY BTU PER HOUR	PRESSURE DROP IN POUNDS PER SQUARE INCH PER 100 FT[a]			
	PIPE SIZES, INCHES			
	7/8	1 1/8	1 3/8	1 5/8
100,000	0.6			
125,000	0.9			
150,000	1.3			
175,000	1.8			
200,000	2.3	0.6		
225,000	2.9	0.8		
250,000	3.6	1.0		
275,000	4.3	1.2		
300,000	5.1	1.4		
325,000	5.9	1.6		
350,000	6.9	1.8		
375,000	7.9	2.1		
400,000	9.0	2.3	0.8	
450,000		2.9	1.0	
500,000		3.5	1.3	
550,000		4.3	1.5	0.7
600,000		5.0	1.8	0.8
700,000		6.7	2.4	1.1
800,000		8.7	3.1	1.4
900,000			3.9	1.7
1,000,000			4.7	2.1
1,200,000			6.7	3.0
1,400,000			9.0	4.0
1,600,000				5.1
1,800,000				6.3
2,000,000				7.9
2,200,000				9.2

[a]Length of tubing includes the average number of fittings

There is a further method of controlling the compressor output which is particularly adaptable to the centrifugal type of machine. This is accomplished by varying the amount of condensing water used with the fluctuation in load demand. Because of the characteristics of the centrifugal type of apparatus, as the condensing water quantity is reduced and the condensing temperature consequently raised, the discharge pressure of the centrifugal machine rises correspondingly and the horsepower input to the machine drops proportionately. While this reduces the total power input to the machine, it does not necessarily reduce the power input per ton of refrigeration developed, as the power input does not drop with a rising discharge pressure as fast as the refrigerating effect is reduced.

TABLE 3. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES

Copper Pipe Actual O D Inches	Capacity BTU per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
¾	2,000	0.3	0.3	0.2	0.2	0.2	0.1	0.1
	4,000	1.3	1.0	0.8	0.7	0.6	0.5	0.4
	6,000	2.8	2.2	1.8	1.5	1.2	1.0	0.9
	8,000	4.8	3.8	3.1	2.6	2.1	1.8	1.5
	10,000	7.4	5.8	4.8	3.9	3.3	2.8	2.3
	12,000	10.5	8.4	6.8	5.6	4.7	4.0	3.3
	14,000	14.0	11.0	9.1	7.6	6.4	5.4	4.5
	16,000		14.5	12.0	9.8	8.3	7.0	5.8
	18,000			15.0	12.3	10.4	8.7	7.2
	20,000				15.0	12.7	10.7	8.9
1⅛	7,000	0.4	0.3	0.3	0.2	0.2	0.2	0.1
	10,000	1.0	0.7	0.5	0.5	0.4	0.3	0.3
	15,000	1.9	1.5	1.2	1.0	0.8	0.7	0.6
	20,000	3.3	2.6	2.1	1.7	1.4	1.2	1.0
	25,000	5.0	4.0	3.2	2.7	2.2	1.9	1.6
	35,000	9.7	7.7	6.2	5.1	4.3	3.6	3.0
	45,000	15.8	12.6	10.0	8.4	7.0	5.9	4.9
	60,000				14.8	12.2	10.2	8.6
	70,000						14.0	11.7
1⅜	10,000	0.3	0.2	0.2	0.2	0.1	0.1	0.1
	15,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	20,000	1.2	0.9	0.7	0.6	0.5	0.4	0.4
	30,000	2.6	2.1	1.6	1.3	1.1	0.9	0.8
	40,000	4.6	3.6	2.8	2.3	1.9	1.6	1.4
	50,000	7.0	5.5	4.4	3.5	2.9	2.5	2.1
	60,000	10.0	7.8	6.2	5.0	4.2	3.5	3.0
	80,000		14.0	11.0	8.7	7.3	6.2	5.2
	100,000				13.5	11.3	9.5	8.2
1⅝	30,000	1.6	1.3	1.0	0.8	0.7	0.6	0.5
	40,000	2.7	2.1	1.7	1.4	1.1	0.9	0.8
	50,000	4.2	3.2	2.5	2.1	1.7	1.4	1.2
	60,000	6.1	4.5	3.6	2.9	2.4	2.0	1.7
	70,000	8.7	6.3	4.8	3.8	3.1	2.6	2.2
	80,000		8.4	6.3	4.9	4.0	3.3	2.8
	90,000			8.0	6.2	4.9	4.1	3.5
	100,000			10.0	7.6	6.1	5.0	4.2
	120,000					8.6	7.0	5.9
	140,000						9.5	7.9

[a]Length of tubing includes the average number of fittings

Table 3. Pressure Losses in Dichlorodifluoromethane Suction Refrigerant Lines (Continued)

Copper Pipe Actual O D Inches	Capacity BTU per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a] — Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
2⅛	50,000	0 7	0 5	0 4	0 3	0 3	0 2	0 2
	100,000	2 6	1 8	1 4	1.1	0 9	0 8	0 7
	150,000	5 6	3.9	3 0	2.4	2 0	1 6	1 4
	200,000	9 8	6.7	5 2	4 1	3 4	2.8	2.4
	250,000	14.8	10 3	8 0	6 3	5 1	4.2	3 6
	300,000		14 5	11 3	9 0	7 2	6 0	5.0
	350,000		19 5	15 3	12 0	9 7	7.8	6 7
	400,000			19 6	15 3	12 5	10.0	8.5
2⅝	50,000	0 2	0 2	0 1	0 1	0 1	0 1	0.1
	100,000	0 7	0 6	0 5	0 4	0 3	0.2	0 2
	150,000	1 6	1 2	1 0	0 8	0 6	0 5	0 4
	200,000	2 8	2 1	1 7	1 4	1.1	0 9	0.7
	250,000	4 3	3 4	2.6	2 1	1 7	1 3	1 1
	300,000	6 1	4 5	3 7	3 0	2 4	1 9	1 5
	350,000	8 2	6 0	5 0	4 0	3.2	2 5	2 0
	400,000		7.8	6 5	5 1	4.2	3 3	2 7
	450,000			7 7	6 4	5.3	4 0	3.5
	500,000				7 8	6 4	5 0	4 2
	550,000					7.7	6 2	5 1
	600,000						7 4	6 2
3⅛	200,000	1 2	1 0	0.8	0 6	0 5	0 4	0 4
	300,000	2 6	2 0	1 6	1 3	1 0	0 8	0 7
	400,000	4 5	3 4	2 6	2.1	1 7	1 4	1 3
	500,000	7 3	5.4	4 1	3.3	2 7	2 2	1 9
	600,000		8.1	6 0	4.7	3 8	3 1	2 7
	700,000			8 4	6 5	5 2	4 2	3 5
	800,000				8 6	6 8	5 5	4 6
	900,000					8 7	7 0	5 9
	1,000,000						8 9	7 3
3⅝	300,000	1.2	0 9	0 7	0 6	0 5	0 4	0 3
	400,000	2 0	1 6	1.3	1 0	0 8	0 7	0 6
	500,000	3.2	2 5	1 9	1 6	1 3	1 0	0 9
	600,000	4 6	3 6	2.8	2 2	1 8	1 5	1 3
	700,000	6 4	4 9	3 8	3 0	2 5	2 0	1 7
	800,000	8.7	6 4	4 9	3 9	3 2	2 5	2 2
	900,000		8 2	6 2	4 9	3 9	3 2	2 7
	1,000,000			7 7	6 1	4 9	4 0	3 3
	1,100,000			9 4	7 3	5 8	4 8	4 0
	1,200,000				8 7	6 9	5 6	4 8
	1,300,000					8 0	6 6	5 6
	1,400,000					9 3	7 6	6 4

[a]Length of tubing includes the average number of fittings.

TABLE 3. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES (CONCLUDED)

Copper Pipe Actual O D Inches	Capacity BTU per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a] Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
4⅛	400,000	1 0	0.8	0.6	0.4	0.4	0.3	0.3
	600,000	2.4	1.8	1.4	1.1	0.9	0 7	0 6
	800,000	4.1	3.1	2.4	2.0	1.6	1.3	1.1
	1,000,000	6.6	4 8	3.7	3 0	2 5	2 0	1 6
	1,200,000	10 0	7 1	5.4	4 4	3 5	2 9	2 4
	1,400,000		10 0	7.5	5 9	4 8	3 9	3 3
	1,600,000			10 0	7.7	6 2	5 1	4 2
	1,800,000				10 0	7.9	6 4	5.3
	2,000,000					9.7	7.9	6 6
	2,200,000						9 5	7 9

[a]Length of tubing includes the average number of fittings.

ADSORPTION SYSTEMS

A diagrammatic representation of an open solid material adsorption system is shown in Fig. 9. Two or more beds of the adsorbent are used so that one bed may be used as an adsorber while another is being reactivated. Most adsorption systems use some internal means of heating the adsorbent bed before activation, and cooling it after activation. Thus, the use of relatively high room temperatures and comparatively large amounts of outside air are desirable in connection with these systems. In order to offset the effect of high air temperature, some effort is made to keep the humidity lower than usual.

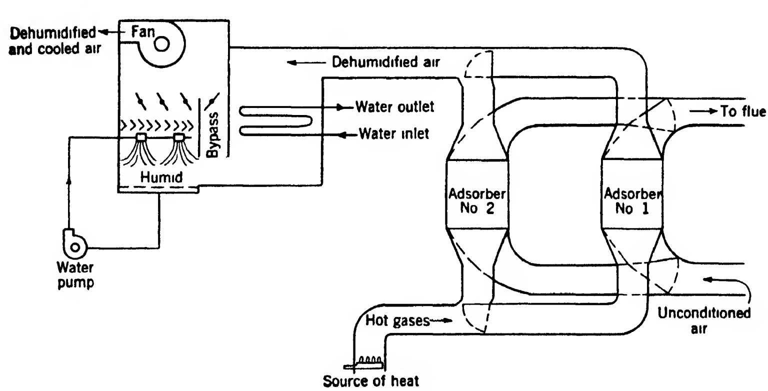

FIG 9. OPEN SOLID MATERIAL ADSORPTION SYSTEM

Silica Gel

Silica gel has two applications when used to replace refrigeration. In the one principally used, the air from which moisture is to be extracted is taken through silica gel beds by suction or pressure fans, and by means of this process the moisture becomes adsorbed by the silica gel and the air leaves at a lower dew-point and a higher sensible temperature than those at which it entered. If this air is passed over surface coolers in which tap water or another cooling medium is flowing through tubes, a certain amount of sensible heat will be removed. The air leaves the surface cooler or interchanger with the same dew-point with which it emerged from the silica gel beds, but with a lower dry-bulb temperature, although the dry-bulb temperature may be higher than the temperature of the air entering the silica gel beds.

In another method, the first two of the steps outlined are duplicated, and in addition the air is carried through a spray type washer. Because the air enters the washer with a low wet-bulb, and because adiabatic saturation will take place at a temperature close to the entering wet-bulb, considerable cooling of the air can be accomplished; but this can be done only with a consequent increase of the dew-point.

It is necessary to reactivate the silica gel after it has adsorbed about 25 per cent of its own weight in the form of moisture. As reactivation requires a high temperature and since silica gel is only active at low temperatures, cooling of the beds must also be completed before they can be used again. This necessitates three stages in the silica gel containers and requires either three beds of silica gel or one bed divided and automatically put in position. The reactivation is usually done by means of gas or oil fires and the cooling of the beds by means of indirect water cooling or by means of small quantities of dehydrated air taken from the system beyond the interchanger.

Activated Alumina

The application is quite similar to that employed for silica gel; that is, the material is exposed to the air flow and after reaching about 75 per cent saturation is reactivated by removing the moisture adsorbed by means of applied heat. The actual scheme generally followed in the use of this material for continuous service varies somewhat from silica gel inasmuch as the material is placed in three units which are used consecutively for the different steps. These steps permit each unit to operate as follows: (1) in series with the preceding unit, (2) alone, and (3) in series with the following unit. This plan allows for adsorption, reactivation, and cooling, in a manner similar to that used with silica gel.

Taking a single unit, when it is in the first step and operating with the preceding unit, the alumina adsorbs approximately 25 per cent of the moisture in the air and takes up about 1.3 per cent of its weight of water. During the second step when it is operating alone, it takes up 100 per cent of the moisture in the air until the weight of the water adsorbed is brought up to about 6.7 per cent. During the third step when the unit is operating with the succeeding unit, it extracts about 75 per cent of the moisture in the air until the water weight adsorbed comes up to about 10 per cent of

the weight of the adsorber. The time allowable for reactivating is equal to the time occupied by the second unit adsorbing alone, plus the time when the second and third units are adsorbing in series, plus the time when the third unit is adsorbing alone, at the expiration of which time the first unit will be again required.

The temperature of air used for alumina reactivation is usually between 300 and 700 F and the air flow rate will have to be higher with the low temperature air than it will be with reactivating air of higher temperature. For example, air at 400 F for reactivating will, at 10 cu ft per hour per pound of alumina, require about 6 hr for reactivation. In the three unit system, after reactivation the cooling of the activated alumina may be carried out with considerable rapidity by using dry air from the adsorption unit for circulation through the unit which has just completed reacti-

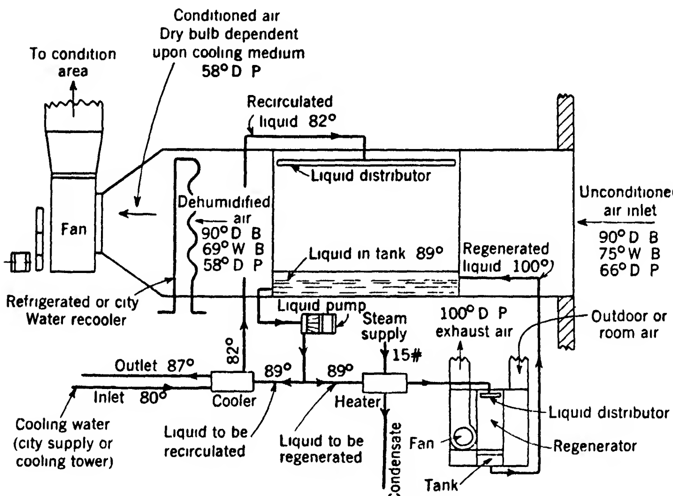

FIG. 10. DIAGRAM OF LITHIUM CHLORIDE ABSORPTION SYSTEM

vation. The final temperature of the unit before it goes back into service should be not over 200 F. As a basis for computing the amount of cooling air required for reactivation, each cubic foot of cooling air has been found capable of removing 2.2 Btu when heated from 85 to 200 F and of providing a sufficient margin of safety in operation.

OPEN ABSORPTION SYSTEMS

The cycle of liquid absorbents are fundamentally the same. A diagram of a lithium chloride absorption system is shown in Fig. 10. The liquid absorbent is brought in contact with air having a certain vapor pressure due to its contained water vapor. The absorbent having a lower vapor pressure, absorbs moisture in the form of water from the water vapor that is in the contacting air. A change of state takes place because there is a rise in temperature in the liquid absorbing the moisture, which is a function of the amount of water vapor condensed from the air stream.

Absorption of moisture by the liquid weakens the concentration of the liquid so that its absorbing capacity is reduced and regeneration, or the driving off of the excess moisture in the liquid, must be performed. A constant density can be maintained by continuously withdrawing a small portion of the total liquid for intensive concentration without varying the vapor pressure of the total mass.

There are two methods of regeneration. One is to boil off the excess moisture by raising the temperature of the solution above the boiling point of the particular concentration. As the salt in the solution does not vaporize, it is not carried off in the boiling process. In the small amounts of liquid diverted to the regenerator for concentration, care should be taken that too much moisture is not driven off, which may cause freezing or solidification of the salts.

The second method of regeneration is to raise the temperature of the solution with ordinary steam coil interchangers to about 225 F, and then passing the solution at this temperature over various types of scrubbers, through which untreated air is circulated. The increase in temperature of the liquid raises its vapor pressure to such an extent that there is an exchange of vapor between the liquid and the air, as well as an equalization of temperature between the air and the liquid absorbent. The air is then capable of taking up part of the moisture from the liquid and carries this excess moisture into the atmosphere with the leaving air.

After this vaporization has taken place, the highly concentrated, hot brine is circulated through an interchanger through which the water used in cooling the main solution can be re-used to reduce the temperature of the concentrated solution to a point where it may be introduced into the main solution tank at only a slightly higher temperature than the main body of the solution.

There are two places in the operation where there is a tendency to raise the temperature of the liquid. One is the absorption of vapor from the air, which changes the latent heat of the vapor absorbed to sensible heat, thus raising the temperature of the liquid and consequently, the temperature of the air. The other is the heat added to the regenerator liquid in order to re-evaporate and carry off the excess moisture which has been condensed in the first stage. This type of system is not limited to lithium chloride, as calcium, zinc or barium chloride and various combinations of halogen salts may also be used.

CLOSED ABSORPTION SYSTEMS

The fundamental rule governing the absorption (in a closed system) of a gas by a liquid is Raoult's Law, which states that at any given temperature the ratio of the partial pressure of a volatile component in a solution to the vapor pressure of the pure component at the same temperature is equal to its *mol* fraction in the solution. The mol fraction, in turn, is equal to the number of mols of substance divided by the total number of mols present. The number of mols in a given weight of a compound is equal to the weight divided by the molecular weight.

This law applies strictly, only to what is known as an ideal solution, that is, one in which the intermolecular forces between the substances

present in the solution are equal. Actually, no such solutions exist, so that deviations from Raoult's Law are always found in practice. The deviation is called positive when the observed pressure is greater than that calculated from Raoult's Law, while the term negative deviation refers to the opposite case. Negative deviations are found wherever chemical attraction exists between the solvent and the solute. Positive deviation occurs when there is a difference in the internal pressure of the components, chemical attraction between them being absent.

In order to make an effective absorption machine, large negative deviations from Raoult's Law must be shown by solutions of the refrigerant in the liquid absorbent, because, the larger the negative deviation, the greater is the amount of refrigerant that can be cycled, using a given weight of absorbent. Cycling a large amount of refrigerant for a given weight of absorbent is important because of the heat required to raise the temperature of the mixture and disassociate the refrigerant

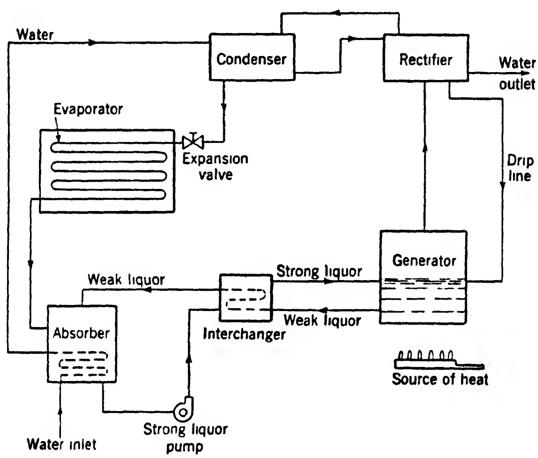

FIG 11. CLOSED ABSORPTION SYSTEM

and the absorbent. Only the latent heat of the refrigerant can be recovered for useful work.

Many refrigerant-absorbent combinations have been proposed and quite a number have been tested. Fig. 11 is a diagrammatic representation of a typical closed absorption system. In this system a mixture of refrigerant and absorbent is evaporated in the generator, passes to an analyzer and rectifier where it is purified, and then to a condenser where the refrigerant and remaining absorbent is condensed. It then passes through an expansion valve to an evaporator, where heat is absorbed from a cooling load. From the evaporator the vapor and residual absorbent passes to an absorber where it meets absorbent which is initially low (weak) in refrigerant concentration. The absorbent absorbs the vapor, and the strong absorbent liquor is transferred to the generator through an interchanger with the weak liquor returning from the generator.

A cooling medium, ordinarily water, is used in the absorber to remove

the heat of absorption and maintain the absorptive power of the absorber at a maximum.

Like the steam ejector system, the absorption system compares most favorably when a cheap source of cooling water and steam or other heat source is available. Unlike the ejector system, the comparative performance is usually best with a wide range of temperature between the evaporator and absorber, since with a good refrigerant-absorbent combination, the amount of heat and water required for a given refrigerating effect increases slowly with an increase of evaporator-condenser temperature range.

At the present time the most used refrigerant-absorbent combinations are: (1) water and ammonia and (2) monofluorodichloromethane and dimethyl ether of tetraethylene glycol. With the latter combination the boiling points of the refrigerant and absorbent are sufficiently wide apart that almost pure refrigerant is obtained without the use of a rectifier.

EVAPORATIVE COOLING

Evaporative cooling is accomplished by passing air through a water spray in which the water is being continually recirculated. The air, entering in an unsaturated condition, evaporates a part of the water at the expense of the sensible heat. As this is an adiabatic transfer, the total heat content of the air remains constant, while the dew-point rises and the dry-bulb falls until the air is saturated.

The reduction in dry-bulb temperature is a direct function of the wet-bulb depression of the air entering the spray chamber and the resulting air temperature is governed entirely by the entering wet-bulb temperature of the outside air and the efficiency of the spray.

Evaporative cooling is being used advantageously in many parts of the country. By using all outside air in sufficient volume to increase the air motion in an occupied space and to limit the temperature rise of the air to approximately 8 F an entering wet-bulb temperature as high as 70 F will result in effective evaporative cooling.

THE REVERSE CYCLE

The idea of heating by the reverse refrigeration cycle has captured the imagination of many people and has been much discussed. In principle, heat is absorbed in an evaporator from some available source of heat, pumped to a higher temperature and delivered to a condenser. The heat from the condenser is used for heating purposes. The compressor acts as a heat pump whose fundamental function is to raise the potential of the heat. The theoretical work of compression in relation to the heat delivered is:

$$\frac{T_2}{T_1 - T_2} \qquad (9)$$

where

T_1 = absolute temperature of evaporator
T_2 = absolute temperature of condenser.

Thus, with a small spread of temperature between the evaporator and the condenser, 6 or 8 times as much heat may be obtained theoretically

and 4 or 5 times practically, as the work put in. There are a number of limitations, however, the most serious of which is the lack of ready availability of a practical source of heat.

1. Well water is the most desirable since its temperature is high even in the winter and thus a large amount of heat may be removed in relation to the weight of water handled.

2. Air may be used but its specific heat is low and its temperature uncertain. When the most heat is needed, the temperature of the air is lowest, thus resulting in the least favorable temperature combination.

3. It has been proposed to obtain heat by freezing water, but this is still in the theoretical stage.

Some of the other factors which act as limitations are the large temperature spread when using air as a source of heat and when attempting to cool with even moderately low outside temperatures, the frequent disparity between the size of the cooling load and heating load requiring extra equipment for a complete heating load, and the relatively high initial cost of equipment at present available for the reverse cycle in comparison with that available for heating by conventional means.

Because of these limitations, the present application of the system is largely limited to temperate climates, such as Florida and Southern California, or to heating only for intermediate seasons, or to other localities which have peculiar advantages as, for instance, the ready availability of well water. In these locations it is frequently possible to do all of the heating necessary with the refrigeration equipment so that the extra cost is only that of reversing the functions of the condenser and evaporator.

ICE SYSTEMS

The use of ice for air conditioning is becoming more adaptable particularly for the smaller commercial installations where first cost is paramount and where the operating period is limited to only a few hours per day. Restaurants serving only one or two meals per day, tea rooms, meeting halls and other comparable installations have found ice to be advantageous in comparison with other methods. Local ice manufacturers will furnish complete data for this type of system.

PROBLEMS IN PRACTICE

1 ● Electrically driven dichlorodifluoromethane condensing units are to be used in an air conditioning system, requiring 20 tons refrigerating capacity for conditions of maximum load. An overall analysis of the seasons operating conditions shows an average load factor of 62.5 per cent, and allowing for variable time intervals of operation of refrigeration units installed, three-quarters of the operating season, or 750 hr, would require operation of the equipment at one-half load, and one-quarter of the operating season or 250 hr full load capacity of the refrigeration equipment would be required.

The increased first cost of 2–10 hp, 10 ton condensing units over 1–20 hp, 20 ton condensing unit is, $830.00 installed price, to the customer.

The increased first cost of a 2-speed compressor motor of 20 hp size over a constant speed 20 hp size motor including increased starter cost is $210.00. The efficiency of the 2-speed motor above is 83 per cent at full load speed, and 79 per cent for full load at ½ speed. At ½ speed, full load is ½ total bhp of full load speed.

Discuss the considerations involved in making a decision as to whether a single unit with a 20 hp motor of the 2 speed type would be used in preference to 2–10 hp constant speed units.

The cost of 2–10 hp 10 ton units in excess of 1–20 hp, 20 ton unit with 2-speed motor, is \$830.00 − \$210.00 or \$620.00, increased first cost. At 15 per cent fixed charges, this represents an increased annual cost of \$93.00 for 2 compressors over one compressor. The advantage of 2 compressors instead of one compressor on an installation of this type, is in the breakdown service provided in the event one compressor is shut down for repairs the system could be operated at one-half capacity utilizing the duplicate machine. The motor efficiency of the constant speed unit would be higher at full load than would be the efficiency of the 2-speed motor at low speed. Offsetting this latter advantage however, is the fact that the condenser on the condensing unit would provide a lower refrigerant condensing temperature for ½ load operation with the same final condensing water temperature than would be the case with duplicate units each furnished with its own compressor and condenser. Operation at a lower condensing temperature would provide for a power saving compensating for the lower efficiency of the 2-speed motor when operated at slow speeds It is, in a case of this kind, purely a question as to whether or not the purchaser would deem an investment of \$620 00 more and an increased fixed charge of \$93 00 a year, advisable to get breakdown service through the installation of duplicate units. In most cases, this increased first cost would not be warranted because of the fact that satisfactory indoor conditions could not be obtained at full load if only one-half the refrigeration capacity were available.

2 ● For condensing purposes, an air conditioning system uses city water which has an average 70 F supply temperature. The following table lists the number of hours per year during which definite wet-bulb temperatures and corresponding refrigeration rates pertain.

Wet-Bulb Temperature F	No. of Hours per Year	Refrigeration Required Tons
80	6	284
79 – 75	100	233
74 – 70	277	183
69 – 65	330	157
64 – 60	277	144
59 – 55	158	79
54 – 50	52	37
	Total 1200 hours	

If the power requirements of a dichlorodifluoromethane refrigeration system are in accordance with the following data on partial load operation, determine the seasonal power cost at 2 cents per kwhr:

Tons of Refrigeration	284	233	183	157	144	79	37
Kw per ton	0.89	0.89	0.87	0.86	0.86	0.93	0.97

Seasonal power cost:

Wet-Bulb Temperature F	Ton-Hours	Kwhr
80	6 × 284 = 1,704	1,704 × 0.89 = 1,517
79 – 75	100 × 233 = 23,300	23,300 × 0.89 = 20,750
74 – 70	277 × 183 = 50,700	50,700 × 0.87 = 44,100
69 – 65	330 × 157 = 51,800	51,800 × 0 86 = 44,500
64 – 60	277 × 144 = 39,900	39,900 × 0 86 = 34,300
59 – 55	158 × 79 = 12,500	12,500 × 0 93 = 11,600
54 – 50	52 × 37 = 1,920	1,920 × 0 97 = 1,860
Totals	181,824 ton-hours	158,627 kwhr

The 158,627 kwhr at 2 cents per kwhr will cost $3,173

The average consumption will be $\frac{158,627 \text{ kwhr}}{181,824 \text{ ton-hours}} = 0.873$ kw per ton.

3 ● Using the data from Question 2, if city water costs 20 cents per thousand gallons, and if 1.25 gallons are used per minute per ton, estimate the annual water cost.

$60 \times 1.25 = 75$ gal per ton-hour.
181,824 ton-hours $\times$ 75 = 13,620,000 gal per year.
$\frac{13,620\ 000 \times \$0.20}{1000} = \$2,724$ the yearly cooling water cost.

4 ● Using the data of Question 2, if a cooling tower were installed for re-using the condensing water, estimate the annual compressor power cost of a dichlorodifluoromethane refrigeration system if the final temperatures of the water leaving the cooling tower and the kilowatt input per ton are the following:

Tons	**284**	**233**	**183**	**157**	**144**	**79**	**37**
Temperature of water leaving tower, F	**86.7**	**81.8**	**76.5**	**72.1**	**66.4**	**61.3**	**55.6**
Kw input per ton	**1.10**	**0.94**	**0.85**	**0.80**	**0.74**	**0.59**	**0.62**

Wet-Bulb Temperature F	Ton-Hours		Kw per Ton		Kwhr
80	1,704	×	1.10	=	1,875
79 – 75	23,300	×	0.94	=	21,900
74 – 70	50,700	×	0.85	=	43,300
69 – 65	51,800	×	0.80	=	41,400
64 – 60	39,900	×	0.74	=	29,500
59 – 55	12,500	×	0.59	=	7,370
54 – 50	1,920	×	0.62	=	1,200
Totals	181,824 ton-hours				146,545 kwhr

The 146,545 kwhr at 2 cents per kwhr will cost $2,931.

The average consumption will be $\frac{146,545 \text{ kwhr}}{181,824 \text{ ton-hours}} = 0\ 805$ kw per ton

5 ● If a steam ejector system were used to secure the refrigeration for the air conditioning system of Question 2, compute the annual steam cost if steam is sold for 53 cents per thousand pounds and if there is an average steam consumption of 20 lb of steam per hour per ton when used with a cooling tower system.

181,824 tons $\times$ 20 lb of steam per ton = 3,636,480 lb of steam.
The 3,636,480 lb at 53 cents per thousand pounds will cost $1,929.

6 ● Discuss the difference in results obtained in cooling and dehumidifying in an air washer from those obtained in a surface cooling coil.

Air leaves a dehumidifying air washer in a saturated condition at a dew-point temperature which can be easily maintained at a constant level by controlling the spray water temperature. This saturated air may then be reheated to proper delivery temperature by reheating coils or by mixing with by-passed air.

For a set air velocity and a set mean refrigerant temperature, a given cooling coil is capable of absorbing a definite amount of heat. Whether the air leaving the coil is saturated or not depends then on the entering dry- and wet-bulb temperatures From practical operating standpoint, the easiest way to control the output of the cooling coil is by means of the dry-bulb temperature of the conditioned space. This means then that the final dew-point will vary somewhat depending on entering air conditions

Summarizing then, the air washers permit close control over both final dry-bulb and final dew-point temperatures, while the surface coolers permit close control over the final dry-bulb only

Chapter 24

HEAT TRANSFER SURFACE COILS

Coil Applications, Construction and Arrangement, Steam Coils, Water Coils, Direct-Expansion Coils, Flow Arrangements, Applications, Calculation of Heat Transfer, Air Flow Resistance, Coil Performance, Selection

THE coils described in this chapter are used in air conditioning systems for heating or cooling an air stream under forced convection. The surface coil equipment may be made up of a number of banks assembled in the field, or the entire assembly may be factory constructed. The applications of each type of coil are limited to the field within which it is rated. Other limitations are imposed by code regulations, by proper choice of materials for the refrigerants used and the condition of the air handled, or by an economic analysis of the possible alternates on each installation.

For heating service, these coils are used as preheaters, reheaters or booster heaters, (see Chapters 21 and 22). The function of the coils is air heating only, but the apparatus assembly may include means for humidification and air cleaning. Steam or hot water are the usual heating media, although others are used in special cases such as, reheating by means of discharge gas from a refrigerating system.

Coils are used for air cooling with or without accompanying dehumidification. Examples of cooling applications without dehumidification are precooling coils using well water or other relatively high temperature water to reduce the load on the refrigerating machinery, or water cooled coils to remove sensible heat in connection with chemical moisture-absorption apparatus. By proper coil selection it is possible to handle both sensible cooling and dehumidification together as further explained later. The apparatus assembly usually includes an air cleaning means to protect the coil from accumulation of dirt and to keep dust and foreign matter out of the conditioned space. Although cooling and dehumidification are the usual functions, there are cases of cooling coils purposely wetted as an aid to air cleaning and odor absorption.

The usual cooling media used in surface coils are cold water and volatile refrigerants such as dichlorodifluoromethane and methyl chloride, but others are used in special cases. Brines are seldom required for the range of applications covered by this chapter, although there are cases where low entering air temperatures with large latent heat loads require a refrigerant temperature so low that water becomes impractical. Some-

times, also, brine from an industrial system already installed, is the only convenient source of refrigeration.

For combined cooling and dehumidifying, surface coils present an alternate to spray dehumidifiers. For many applications it is possible, by proper selection of apparatus, choice of air velocities, refrigerant temperatures, etc., to perform the same duty with either. In a few cases both sprays and coils are used. The coils may then be installed within the spray chamber, either in series with the sprays or below them. In making the selection between spray and surface dehumidifiers, certain advantages of each should be considered. The fact that a spray dehumidifier is usually designed to deliver saturated or nearly saturated air, tends to simplify the control problem. In this case the dry-bulb temperature is also the the dew-point, and hence a dew-point control can be arranged by using a simple duct thermostat. Spray dehumidifiers have the advantage over unwetted coils of a certain degree of air cleaning and odor absorption. On the other hand, coils make possible a closed and balanced cooling water circuit, obviating the unbalanced pumping head, the complication of water level control, and danger from possible floods incidental to multiple-spray dehumidifiers, especially if located on different levels. The use of coils often makes it possible for the same surface to serve for summer cooling and winter heating by circulating cold water in the one season and hot water in the other, with consequent saving in apparatus and piping. Surface-coil dehumidifiers seldom deliver saturated air and wet bulb depression of 0.5 to 4 F (or more) is usual. Another advantage is that where the surface coil system can be used with direct expansion of refrigerant, it is comparatively low in initial and operating costs. Of course the safety of the occupant must be kept in mind in comfort conditioning applications. Some localities have refrigeration codes which restrict the use of direct-expansion coils in the air stream, and hence local codes should be consulted by the engineer before a system employing direct expansion methods is designed. The choice between spray dehumidifiers and coils depends upon the necessities and the economic aspects of each case and no general rule can be given. There are many installations in which either can be used.

COIL CONSTRUCTION AND ARRANGEMENT

Coils are basically of two types, those consisting of bare tubes or pipe and those of *extended* surface construction. The former are little used for the applications covered by this chapter, but are often employed where conditions cause frost accumulation, and for cooling surface within spray dehumidifiers.

The heat transmission from air passing over a tube to a refrigerant flowing within it, is impeded by three resistances. The same is true when the air is being heated by steam or hot water in the tube. The first resistance is from the air to the surface of the tube, usually called the outside surface resistance or air-film resistance. Second is the resistance to the flow of heat by conduction through the metal itself. Finally there is another surface or film resistance to the flow of heat between the inside surface of the metal and the fluid in the tube. For the applications under consideration both the resistance of the metal wall to heat conduction,

and the inside surface or film resistance are usually low as compared with the air-side surface resistance. This is especially the case where sensible heating or cooling only is accomplished. Where dehumidification accompanies sensible cooling, or where the external surface of the tube is sprayed with large quantities of water, the resistance to heat flow between the tube and the air flowing over it is much decreased. In the case of the water spray, the surface resistance depends on the amount and the method of application of the water. Economy in space, weight and cost make it advantageous to decrease the external surface resistance, where it is proportionately large, to approach that of the tube wall, and that from tube to refrigerant. This is accomplished in heating coils by increasing the external surface by means of fins. For cooling or dehumidification without external water spray, fin surface is also commonly used. With water spray the external resistance is already low, and the fins are less useful for increasing the overall heat transfer. Sometimes water spray is

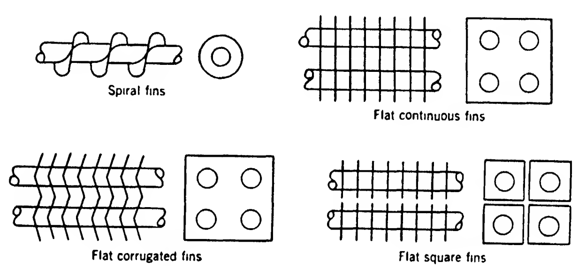

FIG. 1. TYPES OF FIN COIL ARRANGEMENT

applied to the same type surface as would have been used without it The overall heat transfer is not necessarily increased much by such an arrangement, but the water spray may serve other purposes than to increase the flow of heat, such as air and coil cleaning.

In fin or *extended* surface coils the external surface of the tubes is known as *primary* and the fin surface is called *secondary*. The primary surface consists generally of round tubes or pipes. In some cases these are staggered and in others in line with respect to the air flow. The staggered arrangement gives a somewhat higher heat transfer value but also a higher resistance to air flow and in some cases makes the header and return bend arrangement more complicated. A number of types of fin arrangement are used, the most common of which are spiral, flat and flat-crinkled or corrugated, all as shown in Fig. 1. While the spiral fin surrounds each tube individually in all cases, the flat types may be continuous (including several rows of tubes), or they may be round or square, with individual fins for each tube. All of these, as well as other less common types are in use, the selection for a particular installation being based on economic considerations, space requirements and resistances of individual designs of coils. A most important factor in the performance of extended surface coils is the bond between the fin and the tube. An intimate contact is assured in a number of ways. The assembled coil may be coated with tin, zinc, etc., after fabrication. The spiral type fin may be knurled into a shallow groove on the exterior of the tube The tube

may be expanded after the fins are assembled, or the tube hole flanges of a flat or corrugated fin may be made to override those in the preceding fin and so compress them upon the tube. There are also types of construction where the fin is formed out of the material of the tube itself. In any case the successful performance of a fin surface depends upon the bond between fin and tube being secure and remaining so in service.

For heating coils the materials most generally used are copper, steel and aluminum. Sometimes aluminum or brass fins are used on copper tubes. Steel is uncommon except in special cases. Some types of heating coils are made of cast-iron. There are sufficient practical installations of each of these to demonstrate that they can all give good service. The

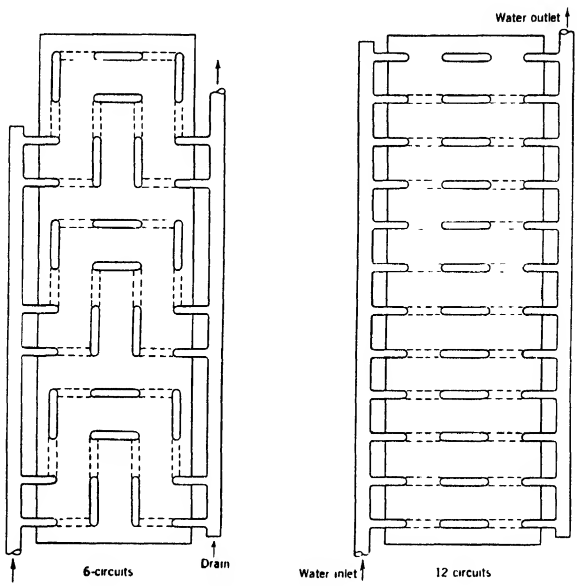

FIG. 2. VARIOUS WATER CIRCUIT ARRANGEMENTS

copper coils are frequently tin-dipped and steel coils galvanized to protect them from corrosion and to assure a bond between fin and tube. For heating applications copper tubes and fins are the most commonly used.

Cooling coils for water or for volatile refrigerants are most frequently of copper, both fin and tube. The coil may or may not be tin-dipped, depending on the type of bond between tube and fin. Aluminum fins on copper tubes are also used. For brines such as sodium or calcium chloride and for ammonia, steel fins and tubes are common.

Although there are many variations for special cases, tube and fin sizes and spacings for air conditioning coils, both heating and cooling, fall within fairly narrow limits. The tubes are usually $3/8$, $1/2$, $5/8$, or $3/4$ in. OD, and the fins spaced from 4 to 8 per inch, 6 per inch being a common design. The tube spacing generally varies from about $1\frac{1}{8}$ to 2 in. on centers. Small tube size and close fin spacing give large capacity with

small space demand, but the resistance, both over the surface and through the tubes, is higher than with larger tubes and more widely spaced fins. Moreover, too close a fin spacing may result in trouble from dirt accumulation, especially on dehumidifying coils, and may also cause trouble from water *hold-up* between the fins, particularly with air flow vertically upward. This condition increases the air resistance and decreases the capacity of the coil. Water hold-up sometimes causes flooding trouble in vertical air flow units by accumulating too much water for the drain to handle all at once when the fan is stopped.

Steam Coils

For proper performance of steam heating coils, condensate and air must be continually eliminated and the steam must be evenly distributed to the individual tubes. This distribution is usually accomplished by individual orifices in the tubes, by distributing plates and orifice in the steam header, or by perforated internal steam-distributing pipes extending

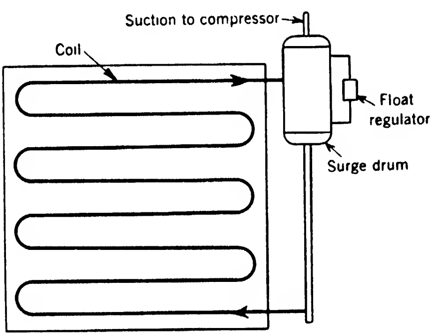

FIG. 3 DIRECT-EXPANSION COIL WITH FLOODED SYSTEM

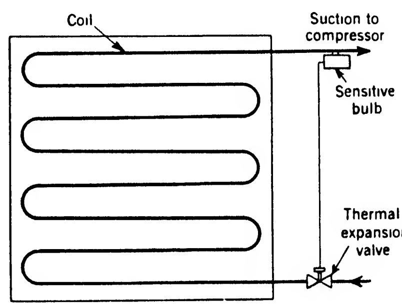

FIG. 4. DIRECT-EXPANSION COIL WITH THERMAL VALVE SYSTEM

into the individual tubes. The latter arrangement has the advantage of distributing the steam throughout the length of each tube, and is conducive to uniform delivered air temperatures. The tendency for freezing of condensate at the bottom of the coil with cold entering air and light heating loads is also minimized. This is especially valuable for outside, air preheaters. Methods of air and condensate elimination are discussed in detail in Chapters 15, 16 and 22.

Water Coils

The performance of water coils, for heating or cooling, depends on the elimination of air from the system and proper distribution of water. Air elimination is taken care of in the system piping as described in Chapter 17. To assure a pressure drop sufficient for adequate distribution but at the same time to provide against excessive pumping head where large water quantities are handled, water coils are provided with various water circuit arrangements. For instance, a typical coil 18 tubes high and 6 tubes deep in the direction of air flow can be arranged for 6, 9, 18 or 36 parallel water circuits as conditions may require. Orifices in individual tubes are occasionally employed but are usually unnecessary as the

resistance of individual water circuits is generally sufficient to effect a satisfactory distribution. In cases such as well water precooling coils, where there may be considerable sand and other foreign matter in the water, provision for cleaning of individual tubes is of advantage. It is important to arrange water coils for drainage if located where they will be exposed to freezing. For this reason the circuits should be so laid out that there are no pockets to hold water. Fig. 2 shows such construction. The drains may be provided in the water piping although they are often arranged in the coil headers.

Direct-Expansion Coils

Coils for volatile refrigerants present more complex problems of fluid distribution than do water, brine or steam. It is desirable that the coil be effectively and uniformly cooled throughout, and necessary that the

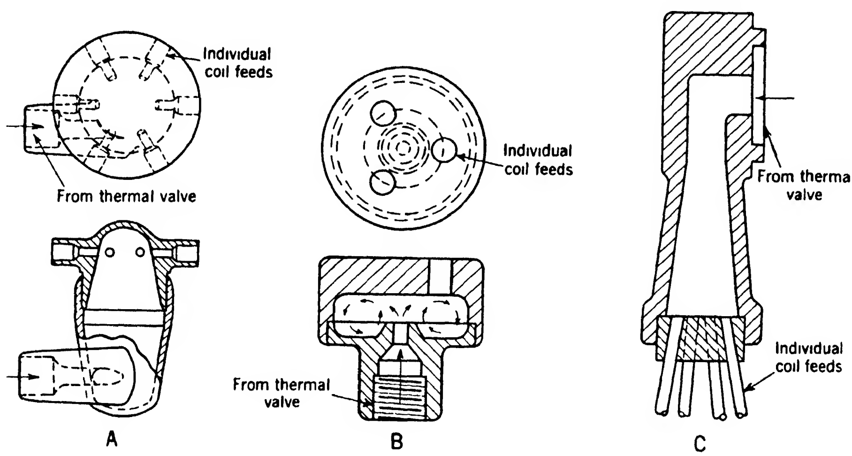

FIG. 5. TYPES OF REFRIGERANT FEED DISTRIBUTING HEADS

compressor be protected from entrained, unevaporated refrigerant. There are two types; namely, flooded systems, and thermal expansion valve systems, as shown in Figs. 3 and 4. With flooded control the coils are supplied with liquid by the same type of circulation that exists in a water tube boiler, while the level in the surge drum is maintained by the action of the float regulator, or by properly charging the plant in the case of the high pressure float drainer. The thermal expansion valve system depends upon the thermal valve automatically feeding just as much liquid to the coils as is required to maintain the superheat at the coil suction outlet within predetermined limits which vary from about 6 to 20 F. The thermal valve arrangement is in common use for the type of coils covered by this chapter, while the flooded system is comparatively rare.

With the flooded system the refrigerant distribution through the tubes depends on properly selecting the length of the feeds and the head of liquid imposed upon the liquid inlets. No auxiliary distributing devices are required. With the thermal valve system there are two factors to consider. There must be, generally, more than one refrigerant feed

through the coil per thermal valve to keep the pressure drop through the refrigerant circuit within practical limits and to reduce the corresponding penalty in increased evaporating temperature. At the same time the coil must be so arranged that the required suction superheat can be attained with a minimum sacrifice in the performance of the coil as a whole. It is general practice to attain this superheat within the coil itself and not by the use of external heat exchangers or other auxiliary devices.

With thermal expansion valves it is advantageous to keep the pressure drop through the refrigerant feeds as low as possible. The feeds are laid out to expose each to the same mean temperature difference so that it handles the same refrigerating load. A distributing means is imposed

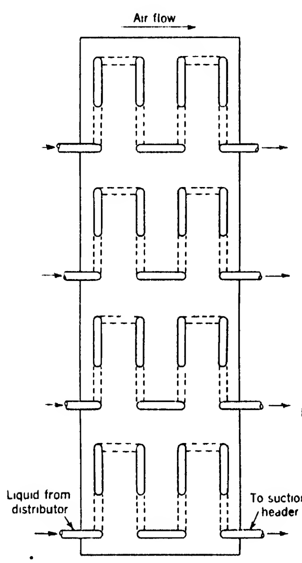

FIG. 6 ARRANGEMENT FOR SUPERHEATING AT AIR OUTLET

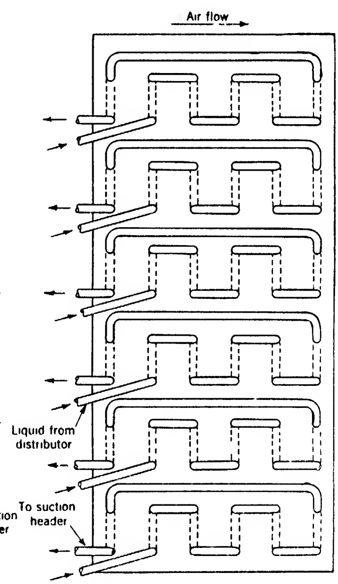

FIG 7. ARRANGEMENT FOR SUPERHEATING AT AIR INLET

between valve and coil liquid inlets to divide the refrigerant equally among the feeds. Such a distributor shall be effective for distributing both liquid and vapor, since the entering refrigerant is a mixture of the two. Fig. 5 shows three typical types of distributors. In distributor *A* the liquid and gas mixture from the thermal valve is led tangentially into a chamber. The coil feed connections extend outward radially at the top of this chamber. In distributor *B* the refrigerant is discharged at a high velocity through a central jet against the end plate, forming a uniform mixture of gas and liquid within the distributor, from which individual connections are led as shown. In type *C* the refrigerant enters at high velocity from the thermal valve and is discharged against the end plug in which the individual liquid feeds are closely arranged. These distributors can be used in either vertical or horizontal position. Although there

are other forms of distributors the above are typical examples. The individual liquid connections from the distributor to the coil inlet are commonly made of small diameter tubing and are all of the same length and diameter in order to impose the same friction between the distributor and the coil. Since the thermal valves act in response to the superheat at the coil outlet, this superheat should be produced with the least possible sacrifice of active evaporating surface. Where conditions require a coil which is thin in the d.rection of air flow, with large quantities of air per ton of refrigeration, the temperature difference between leaving air and refrigerant is large and it is, therefore, practical to feed the coil in the direction of the air flow as shown in Fig. 6. If the refrigerant temperature

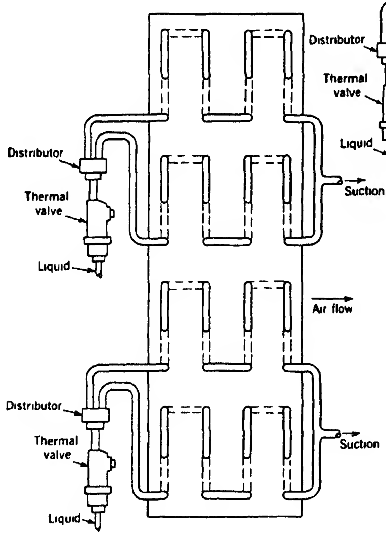

FIG. 8. ARRANGEMENT FOR FACE CONTROL

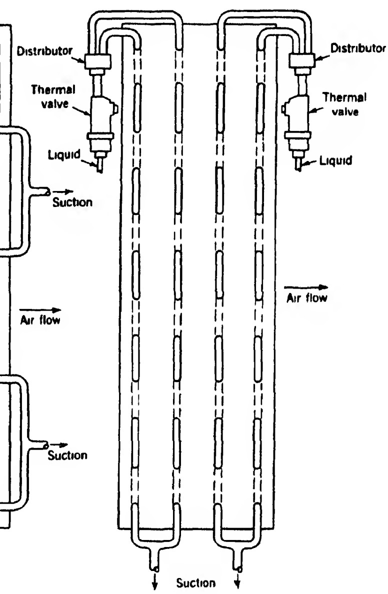

FIG 9. ARRANGEMENT FOR DEPTH CONTROL

is high and the air quantity per ton low so that the leaving temperature difference becomes small, it is expedient to arrange the circuits so that the refrigerant is brought back to the air inlet side of the coil to take advantage of the large temperature difference at that point for superheating, as in Fig. 7. Sometimes a single thermal valve is used per coil. In other cases multiple valves are used, with the coil divided across the air flow or parallel to it as shown in Figs. 8 and 9. The arrangement of Fig. 9 has the disadvantage of unequal load on the two parallel circuits. The choice between face or thickness division is dictated by the type of automatic temperature and humidity control used.

Flow Arrangements

The relative direction of flow of the air outside the tubes and the media within them influences the performance of the surface. There are three

types of relative flow in common use. Fig. 10A shows *parallel flow* in which the air and the medium in the tubes proceed through the coil in the same direction. Fig. 10B shows *counter-flow* in which the media in the tubes proceed in a direction opposite to the flow of air. Fig. 10C shows *cross-flow* in which the air and heating or cooling media pass at right angles to each other. *Parallel-flow* is often used in coils fed with volatile refrigerant as it is characterized by a fall of refrigerant temperature in direction of air flow due to the pressure drop through the refrigerant circuit. *Cross-flow* is common in steam heating coils, the temperature within the tubes being substantially uniform, and the mean temperature difference the same whatever the direction of flow, relative to the air. *Cross-flow* is also used in very thin coils with brine or water or volatile refrigerants, it being impractical to arrange these coils any other way. The *counter-flow* arrangement or some modifications of it is used almost universally in

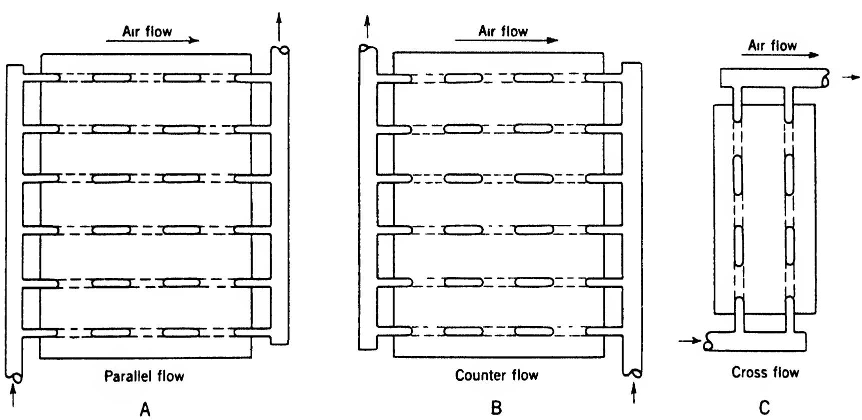

FIG. 10. FLOW OF MEDIA IN TUBES IN RELATION TO AIR FLOW

brine or water coils which are deep in the direction of air flow, to take advantage of the highest possible mean temperature difference for given entering water and air temperatures. *Counter-flow* is frequently used in coils fed with volatile refrigerant to take advantage of the higher air temperature for superheating the leaving gas. This arrangement permits complete evaporation of the refrigerant and proper operation of the thermal expansion valve.

Applications

Heating coils in field assembled banks are used for a number of purposes as described in Chapter 21. They may be arranged with the air flow vertical or horizontal, although the latter is more common. For steam heating the coils may be set with the tubes vertical or horizontal. In the latter case the coil should be sloped to provide for condensate drainage. Because of the multi-circuit feed arrangement and the necessity for avoiding air and water pockets, water heating coils are generally arranged with the tubes horizontal. Certain precautions must be taken against freezing. Where steam coils are used with entering air below

freezing temperature, throttling the steam supply may result in freezing the condensate in the bottom of the coil if the tubes are of the variety not provided with internal distributing pipes, or an equivalent arrangement. If these are used, there is little danger of freezing the condensate as long as the leaving air temperature is not allowed to fall below about 40 F. As an added precaution with both steam and water coils the outside air inlet dampers are often closed automatically when the fan is stopped to avoid trouble caused by very cold outside air drifting in during *off* periods.

A typical arrangement of water cooling coils is shown in Fig. 11. Some means should be provided to filter all the entering air to keep dirt and foreign matter from accumulating on the coils. The assembly is provided with a drip-pan to catch the condensate during summer dehumidifying duty and to collect the non-evaporated water from the humidifying sprays

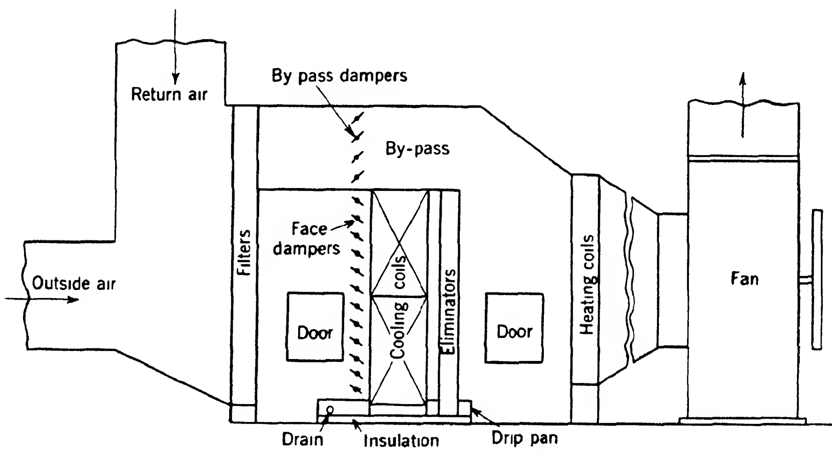

FIG. 11. TYPICAL ARRANGEMENT OF COOLING COILS IN A CENTRAL SYSTEM

in winter. The drip connection should be made ample in size and liberally provided with plugged tees and crosses for cleaning. It should not be exposed to freezing temperatures in winter if the apparatus is used on winter humidifying duty. Access doors should be provided for servicing filters, humidifying nozzles, and fan bearings and for cleaning the coils With certain designs of coils when used for dehumidifying, eliminators must be used beyond the coil to catch any water which may be blown into the air stream. It is customary to include these eliminators when the air velocity exceeds about 450 fpm with the individual fins and about 600 fpm for the continuous flat fin type. Where a number of coil sections are stacked one upon another, and where the velocities are low, so that eliminators need not be used, occasional trouble results when water splashes down from one coil to the next and blows out into the air stream. In such cases drip troughs as shown in Fig. 12, are used to collect this water and conduct it to the condensate pan.

Sometimes finned surface coils on summer cooling and dehumidifying duty are provided with water sprays. These sprays are of two types.

In the first type a set of spray nozzles is arranged for intermittent cleaning. The operator can wash the coils off as frequently as necessary. These sprays are not operative when the system is in use and no recirculating pump is provided. The second arrangement requires a collecting tank and a recirculating pump. The water is in circulation whenever the apparatus is in operation, and assists in keeping the coil clean and in absorbing odors. Fig. 13 illustrates such an arrangement. Wherever air by-passes are used around a coil on summer duty for control purposes, it is of advantage to direct only return air through the by-pass rather than a mixture of return and outside air. The casing should be arranged accordingly. To maintain the air quantity handled by the fan reasonably constant, and to assure the required design quantity of by-passed air

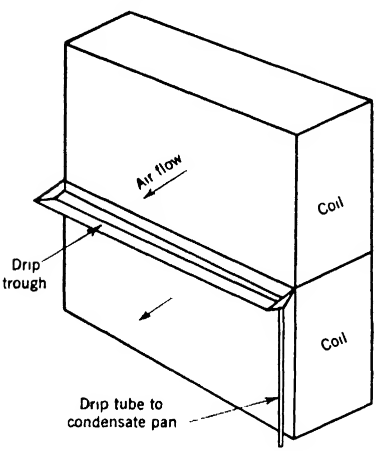

FIG. 12 COIL ARRANGED WITH DRIP TROUGH

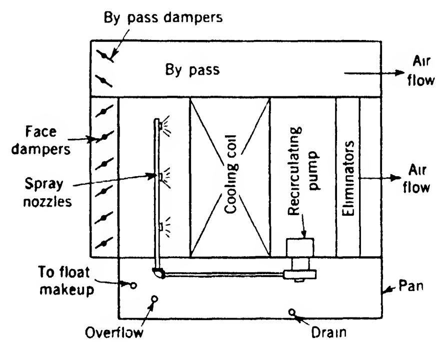

FIG 13 RECIRCULATING SPRAY SYSTEM FOR CLEANING COILS

when the by-pass damper is open, cooling coil banks are frequently furnished with both face and by-pass dampers as shown in Fig. 11.

Although both heating and cooling coils are made of sufficient strength to take up expansion and contraction arising within themselves, care should be taken to avoid imposing strains from the piping on to the coil connections. (See Chapters 16 and 17).

HEAT TRANSFER AND AIR FLOW RESISTANCE

The transfer of heat between the heating or cooling medium and the air stream is influenced by several variables:

1. The magnitude of the driving force, i e , the temperature difference
2. The design and surface arrangement of the coil.
3. The velocity and character of the air stream.
4. The velocity and character of the medium in the tubes

The driving force is usually taken as the logarithmic mean temperature difference for heating or cooling without dehumidification. For combined

cooling and dehumidification, a special measure of the propelling force is used as described later. Logarithmic differences are generally employed in practice although there are special flow relationships used, such as cross-flow, where they do not strictly apply. With volatile refrigerants there is often an appreciable pressure drop and corresponding change in evaporating temperature through the refrigerant circuit. The problem is further complicated by the fact that the refrigerant is evaporating in part of the circuit and superheating in the remainder. In spite of this, heat transfers and ratings for coils using volatile refrigerants are usually based in practice on a refrigerant temperature corresponding to the pressure at the coil outlet.

The design and surface arrangement of the coil includes such items as materials, type, thickness, height and spacing of the fins, and the ratio of this surface to that of the tube, the use of the staggered or in-line tube arrangement, and provisions to increase the air turbulence such as the use of corrugated as against flat fins. Staggered tubes increase the total heat transfer as against the in-line arrangement and corrugated fins are more effective than flat. Of especial importance is the bond between fin and tube.

The velocity of the air usually considered is the coil face velocity This bears a varied relation to the actual velocity over the surface, depending upon the individual coil design. As long as a fixed design of coil is under consideration face velocities may be used, but they may be unsatisfactory in comparing different designs, as it is the actual surface velocity that is significant. The air volume is often based on standard air at 70 F and a barometric pressure of 29.92 in. Hg. The use of air volume in coil rating information may be misleading. The significant value is mass velocity in pounds per minute and not cubic feet per minute, because for a fixed volume the corresponding weight may vary widely, depending upon the air density, temperature and barometric pressure under consideration.

At the same mass air velocity, varying performance can be obtained depending upon the turbulence of the air flow into the coil and upon the uniformity of distribution of air over the coil face. The latter is very important in obtaining reliable test ratings and in realizing rated performance in practical installations. The resistance through the coils will assist in properly distributing the air, but where the inlet duct connections are brought in at sharp angles to the coil face, the effect is frequently bad and there may even be reverse air currents through the coils. This reduces the capacity, but can be largely avoided by proper layout or by the use of directing baffles.

The heat transfer depends also upon the velocity of the medium in the tubes and upon its character, whether flowing water, condensing steam or evaporating volatile refrigerant. In the latter case, the effect is complex because of the combination of evaporation and superheating, and because of the influence of pressure drop in the refrigerant circuit. Heat transfer rates expressed as Btu per square foot of internal surface per degree logarithmic mean effective temperature difference between the fluid and tube wall are, for example, about 150 to 300 for evaporating dichlorodifluoromethane, about 350 to 1200 for water at 2 and 6 fps and about

1200 for condensing steam. The influence of the medium in the tubes on the overall heat transfer rate is, therefore, apparent.

Because of these variables, reliable rating and performance information for any design of coil must be based on actual tests on that coil under the expected conditions of operation. A comparison between the performance of two designs, unless based on such tests on each, may lead to entirely erroneous conclusions.

Heating and Dry Cooling Coils

To find the surface requirements for heating or dry cooling coils, the heat transfer coefficient U and the dry-bulb mean temperature difference must be known. It has been found that U can be expressed as an exponential function of the mass air velocity. If U_1 is known at a given velocity V_1, its value at another velocity V_2, can then be found by the Equation 1.

$$U_1/U_2 = (V_1/V_2)^n \tag{1}$$

This relationship can also be represented on logarithmic paper as a straight line of slope n. The value n must be determined by test as it is dependent upon the coil design. For coils in common use, n ranges from about 0.4 to 0.8. For a given design, n will depend upon the coefficient of heat transfer from the fluid in the tubes to the tube wall, so that n will be higher as the water velocity in the tube increases, and will be higher for condensing steam than for water at low velocities. A coil that is several rows deep will have a higher n than one of the same design with fewer rows. For heat transfer information as well as values of n for typical coils, see Table 1.

To avoid the labor of coil selection by use of heat transfer coefficients and mean temperature differences, the ratings of heating and dry cooling coils are frequently set up in tables from which, for known conditions, a coil can be selected directly.

Dehumidifying Coils

When air passes through a cooling coil, the temperature of which is lower than the dew-point of the air, there is a removal of both sensible and latent heat. The sensible heat transfer process is exactly the same as in heating and non-dehumidifying cooling coils. The moisture passes from the air to the cold surface by diffusion.

It is evident that the coil may be wet throughout or, because of temperature gradient through fins or temperature range in the refrigerant, may be partially dry and partially wet. In this case, part of the coil acts as dry and part as dehumidifying surface. Various approximations are used for this condition as the percentages of wet and dry surface and the temperatures applying to each are difficult to ascertain. One approximation is to assume that for ratios of total to sensible heat of 1.10 or more the coil is to be treated entirely as wet, and for ratios less than 1.10 entirely dry.

Although much research has been already conducted and more is in progress, there is no general agreement as to the most satisfactory and convenient manner in which to rate dehumidifying coils. A large number of methods are now in use, most of them combinations of theory and

TABLE 1. HEAT TRANSFER INFORMATION ON TYPICAL COPPER CIRCULAR FIN COILS

Dipped Metal Bond or Integral Fins

TEST NO.	DATA SOURCE	WORKING FLUID	NO OF ROWS	IN-LINE OR STAGGERED	TUBE DIAM IN	O D FINS IN	FINS PER IN	PER CENT FACE AREA	U AT 500 FPM FACE	n
	Air Heating Service									
1	Reported	Steam 2-100 lb	4	Stag	0 375	0 87	6	57	13 0	0.62
2	Research Tests	Steam 2 lb gage	2	In-line	0 625	1 37	7	42	10 8	0 61
3	Commercial	Steam 5-150 lb	1-6	Stag	0 625	1 44	8	50	10 7	0 52
4	Commercial	Steam 5-150 lb	1-6	Stag	0 625	1 37	7	50	10 5	0.57
5	Commercial	Steam 5-100 lb	1-6	...	0 625	...	7	...	10 5	0 60
6	Research Tests	Steam 5 lb gage	4	Stag	0 625	1 47	7	53	10 5	0 65
7	Commercial	Steam 5-150 lb	1	Cast-Iron Sec	...	...	..	44	10 2	0 62
8	Research Tests	Steam 2 lb gage	2	In-line	0 625	1 12	6	37	10 1	0 55
9	Research Tests	Steam 5 lb gage	1	...	0 625	1 47	7	52	9 3	0 52
10	Research Tests	Steam 5 lb gage	1	...	0 625	1 37	7	56	9.1	0 49
11	Commercial	Steam 5-150 lb	6	Cast-Iron Sec	...	...	..	44	9 1	0 62
12	Research Tests	Steam 2 lb gage	1	...	0 625	1 37	7	49	9 0	0 48
13	Research Tests	Steam 2 lb gage	1	...	0 625	1 12	6	37	9 0	0 46
14	Commercial	Steam 5 lb gage	1	Cast-Iron Sec	...	...	..	...	7 0	0 50
15	Commercial	Steam 5 lb gage	6	Cast-Iron Sec	...	...	..	...	6 5	0 60
	Air Cooling Without Dehumidification									
16	Commercial	Water 2 fps	1-6	Stag	0 625	1 37	7	50	8 9	0 47
17	Commercial	Water 2 fps	1-6	Stag	0 625	1 44	8	50	8 6	0 45
18	Reported	Water 2 fps	...	...	0 625	1 44	8	50	8 5	...
19	Research Tests	Water 2 fps	2	Stag	0 75	1 75	7	55	7 2	0 45
20	Research Tests	Water 2 fps	6	Stag	0 75	1 75	7	55	7 1	0 64
21	Commercial	Direct Expansion	1	...	0 75	1 62	4-6	...	5 8	0 62
22	Commercial	Direct Expansion	6	...	0 75	1 62	4-6	...	4 7	0 61
23	Research Tests	Direct Expansion	4	Stag	0 75	1 75	7	44	6 2	0 51

empirical presentations of test data. Information is given, usually in the form of tables or curves to determine the total cooling capacity, with supplementary devices to ascertain the proportions of sensible and latent. Sometimes the presentation of data is simplified, often at the expense of flexibility and coverage, by setting up tables covering limited and specific air and refrigerant conditions, in which total, sensible and latent capacities are set forth directly. Some of the methods now used require trial and error solutions and others are of very questionable accuracy, although the error may be small when applied within narrow limits of the several variables. It is hoped that investigations now in progress will lead to a convenient and reasonably accurate method of rating which can be widely adopted.

In rating dehumidifying coils, there are two requirements. The total capacity must be determined and the proportion of sensible and latent heat transfer ascertained. These determinations often involve an average coil surface temperature. This may be determined experimentally by the use of thermocouples or calculated theoretically from other test data. Sometimes, a fictitious temperature is used, determined by the point of intersection between the saturation curve on the psychrometric chart and a straight line drawn through points representing the entering and leaving air conditions.

The total cooling capacity is determined in a variety of ways, of which the following are the most usual:

1. The use of surface or overall heat transfer coefficients in conjunction with dry-bulb mean temperature differences. The result is corrected for dehumidification by means of functions for. (1) the temperatures differences, (2) the expected total to sensible load ratio, and (3) empirical factors determined from test

2. The use of surface or overall coefficients for combined sensible and latent heat removal with a wet coil, using as the driving force the difference between heat content of the entering air and that of saturated air at either the surface or the refrigerant temperature.

3 The calculation of sensible and latent capacities separately. The sensible is based on dry-bulb mean temperature difference and heat transfer while the latent is determined using a dew-point mean difference and a corresponding latent heat transfer value.

4. The use of a *contact factor* or ratio of heat removed to heat removable This factor is a function of coil depth and air velocity, is experimentally determined for each design and used in conjunction with a so-called *surface temperature.*

The total capacity is influenced by the factors already enumerated and, as with dry cooling or heating coils, the air velocity correction is an exponential function of the mass air velocity.

Unless the sensible and latent capacities have been separately calculated as in item 3 above, various means are used to determine these.

The slope of a straight line on the psychrometric chart connecting points representing the entering and leaving air conditions gives a direct means of fixing the proportions of sensible and latent heat removal. Such a line is termed the *load-ratio* line. For given entering air conditions and a required proportion of sensible and latent cooling, the load ratio line may therefore be drawn, its slope being determined by the temperature and moisture content coordinates of the psychrometric chart used.

To satisfy the performance requirements, the point representing the leaving air conditions for a coil chosen to give the necessary total cooling capacity must also fall on this line.

Some of the common methods of coil rating are based on the assumption that the load ratio line for a given coil under given operating conditions not only passes through points representing the entering and leaving air conditions, but also intersects the saturation curve at a temperature fictitiously referred to as the coil *surface temperature*, see Fig. 14. This *surface temperature* is higher than the refrigerant temperature by an amount proportional to the total cooling load, and determined by tests for each design of coil. For a selected coil, therefore, the load ratio line can be determined after such tests have been made. The chosen coil will not satisfy the required performance unless the coil load ratio line coincides with that of the required duty, a condition requiring a trial and error process. This entire method is not suitable when the proportion of latent heat load is high, because the true load ratio line fails to intersect the saturation curve.

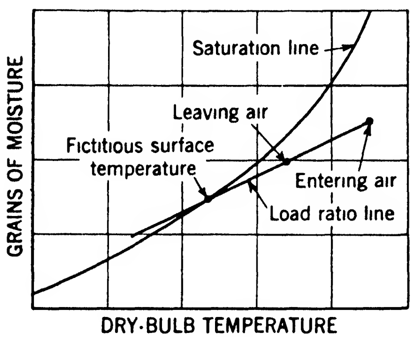

FIG. 14. LOAD RATIO LINE ON PSYCHROMETRIC CHART

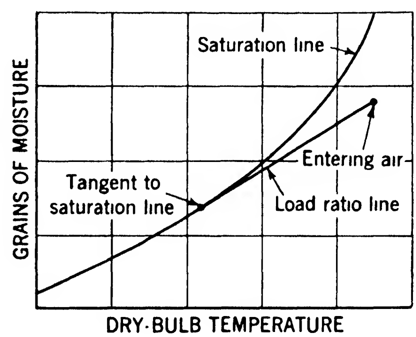

FIG. 15. APPROXIMATE MAXIMUM LOAD RATIO OR LATFNT HEAT LOAD

Having selected a coil for which the load ratio line and total capacity meet the estimated load requirements, the condition of the leaving air can be located on the load ratio line by separately determining from rating tests and psychrometric information, any quantity which will give an intersection with the load ratio line, such as:

1. The dry-bulb temperature of exit air.
2. The wet-bulb temperature of exit air.
3. The dew-point temperature of exit air.
4. The relative humidity of exit air.
5. The wet-bulb depression of exit air.
6. The ratio: $\frac{\text{Length of line from inlet to exit conditions}}{\text{Length of line from inlet to saturation curve}}$.

(This is frequently called the *ratio of heat removed to heat removable*).

As it is evidently impossible to dehumidify with a cooling coil, with no removal of sensible heat, there must be a limit to the possible ratio of total to sensible heat removal. A common approximation is to limit this ratio to that corresponding to a load ratio line drawn through the entering air condition and tangent either to the saturation curve or to an arbitrarily selected relative humidity line not far from saturation. (See Fig. 15).

The resistance to air flow for a given coil is usually greater for a wetted than for a dry coil, the difference being dependent upon the coil design and the facility with which the water of dehumidification is removed. The effect of water is greater with close than with wide fin spacing and for some designs is greater with upward vertical than with horizontal air flow. For typical coils, the air resistance of the wet coil is about 30 per cent greater than the resistance of the dry coil, although this depends also on the heat load ratio. Air resistance is usually taken as proportional to the square of the mass air velocity, although the 1.8 power is sometimes used instead of the square.

COIL PERFORMANCE AND SELECTION

In the selection of a coil it is necessary to consider several factors:

1. The duty required—heating, cooling, dehumidifying
2. Temperature of entering air—dry-bulb only if there is no dehumidification, dry- and wet-bulb if moisture is to be removed.
3. Available heating and cooling media
4. Space and dimensional limitations
5. Air quantity limitations.
6. Allowable resistances in air circuit and through tubes
7. Peculiarities of individual designs of coils.
8. Individual installation requirements, such, for example, as type of automatic control to be used.

The duties required may be determined from information in Chapters 5, 6, 7 and 8. There may or may not be a choice of cooling and heating media, as well as temperatures available, depending upon whether the installation is new or is in combination with present sources of heating or cooling. Space limitations are dictated by the requirements of individual cases. The air quantity is influenced by a number of considerations. The air quantity through heating coils is often made the same as that necessary to handle the summer cooling load The air handled may be fixed by the use of old ventilating ducts as an air distribution system for new air conditioning apparatus, or may be dictated by requirements of satisfactory air distribution or ventilation. The resistance through the air circuit influences the fan horsepower and speed. This resistance may be limited to allow the use of a given size of fan motor, or to keep the operating expense low, or it may be limited by the maximum fan peripheral velocity which requirements of quietness may permit. The friction through the water or brine circuit may be dictated by the head available from a given size of pump and pump motor. As the fan and pump motor inputs represent a refrigerating load on cooling installations, it is economical to keep them low.

Proper performance of a surface heating or cooling coil depends upon correct choice of the original equipment and upon certain other factors. The usual coil ratings are based on a uniform face velocity or air. If the air is brought in at odd angles or if the fan is located so as to block part of the air flow, the performance as given in the Manufacturer's Ratings cannot usually be obtained. To obtain this performance it is necessary also that the air quantity be adjusted on the job to that used in determining the coil selection, and must also be kept at this value. The most common causes for a reduction of air quantity are the fouling of the filters

and collection of dirt in the coils. These difficulties can be avoided by proper design and proper servicing. There are a number of ways in which coils may be cleaned. A common method is to wash them off with water. They can sometimes be brushed and cleaned with a vacuum cleaner. In bad cases of neglect, especially on restaurant jobs where grease and dirt have accumulated, it is sometimes necessary to remove the coils and wash off the accumulation with steam, compressed air and water, or hot water. The most satisfactory solution, however, is to keep the filters serviced, and thus make the cleaning of the coils unnecessary.

The proper selection of coils requires an understanding of the necessities of each case and should be based on an economic analysis of the plant design as a whole. No general rule can, therefore, be laid down for the selection of heating or cooling coils. It is possible, however, to point out the limits of usual practice and to indicate the influence of the variables involved in the coil selection.

Heating Coils

Steam and hot water heating coils are usually rated within these limits:

Air Face Velocity—200 to 1200 fpm, sometimes up to 1500 fpm.

Steam Pressure—2 to 200 lb, sometimes up to 350 lb per square inch.

Hot Water Temperature—150 to 225 F.

Water Velocity—2 to 6 fps.

Individual cases may deviate widely, but the tabulation given herewith will serve as a guide to usual heating practice:

Air Face Velocity—500 to 800 fpm face, 500 being a common figure

Delivered Air Temperature—varies from about 72 F for ventilation only to about 150 F for complete heating

Steam Pressure—2 to 10 lb, 5 lb being common.

Hot Water Temperature—150 to 225 F.

Water Velocity—2 to 6 fps.

Water Quantity—Based on about 20 F temperature drop through a hot-water coil.

Air Resistance—The total resistance through heating coils is usually limited to from 3/8 to 5/8 in of water gage for public buildings, to about 1 in for factories.

The selection of heating coils is relatively simple as it involves dry-bulb temperatures and sensible heat only, without the complication of simultaneous latent heat loads, as in cooling coils. For a given duty, entering air temperature, and steam pressure it is possible to select several arrangements of the same design of coil depending upon the relative importance of space, cross-sectional area, and air resistance. Table 2 shows an example.

TABLE 2. SEVERAL HEATING COIL ARRANGEMENTS

SELECTION	1	2	3
Steam pressure, lb per sq in	5	5	5
Temperature of air entering coil, deg F	40	40	40
Temperature of air leaving coil, deg F	129	129	129
Air quantity, cfm	10,000	10,000	10,000
Coil face area, sq ft	33.3	12.5	7.50
Coil rows deep	2	3	4
Face velocity, fpm	300	800	1330
Air friction, in. water	0.044	0.396	1 077

Cooling Coils

The usual range of ratings for cooling and dehumidifying coils are enumerated herewith:

Entering Air Dry-Bulb—60 to 100 F.

Entering Air Wet-Bulb—50 to 80 F.

Air Face Velocities—300 to 800 fpm, (sometimes as low as 200 and as high as 1200).

Volatile Refrigerant Temperatures—25 to 55 F, at coil suction outlet.

Water Temperatures—40 to 65 F.

Water Quantities—2 to 6 gpm per ton, or equivalent to a water temperature range of from 4 to 12 F.

Water Velocity—2 to 6 fps.

The ratio of total to sensible heat removed varies in practice from 1.00 to about 1.65, i.e., sensible heat is from 60 to 100 per cent of total, depending on the application. (See Chapter 21, Tables 1 and 2). Required ratios may demand wide variations in air velocities, refrigerant temperatures, and coil depth, so that general rules as to these values may be misleading. On usual comfort installations air face velocities between 400 and 600 fpm are frequent, 500 being a common value. Refrigerant temperatures will ordinarily vary between 40 and 50 F where cooling is accompanied with dehumidification. Water velocities will range from 2 to about 6 fps.

When no dehumidification is desired, for which condition the dew-point of the entering air will be equal to or lower than the cooling coil temperature, the coil selection is made on the basis of dry-bulb temperatures and sensible heat transfers only, the same as with heating coils. It is possible also to choose various arrangements of face area, depth, air velocity, etc , for the same duty, as illustrated in Table 2 for a steam coil.

Dehumidifying Coils

The selection of coils for combined cooling and dehumidifying duty is more involved than for heating or sensible cooling and requires consideration of both dry- and wet-bulb air temperatures. It is further complicated by the fact that the proportional amount of dehumidification required is also highly variable. The methods outlined previously under Heat Transfer and Resistance may be used to determine whether it is possible for a coil to perform the duty required. If entering and leaving

TABLE 3. VARIOUS COOLING COIL ARRANGEMENTS

SELECTION	1	2	3	4
Total cooling capacity, tons	100	100	100	100
Sensible cooling capacity, tons	69	69	69	69
Latent cooling capacity, tons	31	31	31	31
Ratio total to sensible heat	1 45	1.45	1.45	1.45
Air quantity, cfm	47,800	41,700	37,100	46,800
Cfm per total ton	478	417	371	468
Face velocity, fpm	325	423	500	600
Resistance, in. water	0 11	0.27	0 51	0 37
Coil face area, sq ft	147	99.0	74.2	78.1
Coil rows deep	4	6	8	4
Coil evaporator temp, deg F	45	45	45	38

air conditions are arbitrarily specified, the corresponding duty sometimes cannot be obtained at all without the use of reheat. As with heating and sensible cooling coils, there are combinations of face areas, depth, air velocity and refrigerant temperatures which will give the required performance. This is illustrated in Table 3.

It is possible as shown in Table 3 to perform approximately the same duty at a given refrigerant temperature with small face area and large thickness or vice versa. The large face area coil will give low air velocity and resistance but high air quantities per ton. The coil of small face area and great depth will require small air quantities per ton of refrigeration, high resistance and high air velocities. As shown also in Table 3 the same sensible, latent and total cooling capacity may be obtained with various refrigerant temperatures by proper choice of coil. This makes it possible

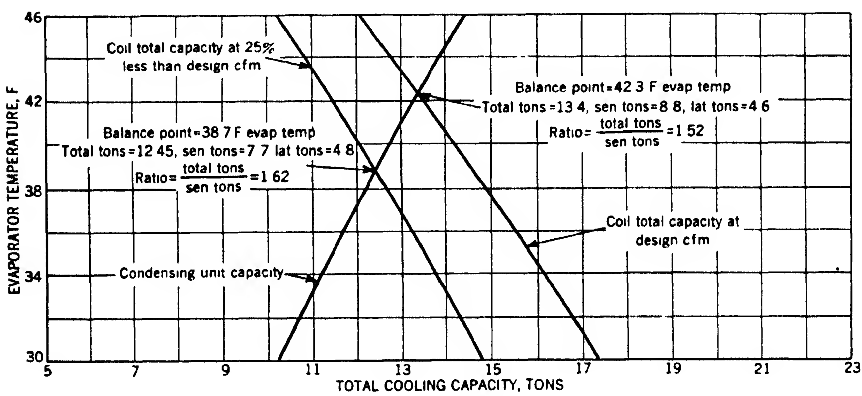

FIG 16. BALANCE OF EVAPORATOR TEMPERATURE CONDITIONS WITH CONDENSING UNIT CAPACITY

to keep the evaporating temperature high enough to carry the load with a chosen size of condensing unit. High evaporating temperatures with correspondingly small compressor operating expense can be attained but at the expense of coil surface, air quantity or both. The choice will be determined by the necessities of individual installations.

For a given quantity and condition of entering air the evaporating temperature of a volatile refrigerant coil will be determined by a balance between the condensing unit and the coil. The total, sensible and latent cooling capacity can then be determined from the coil rating information. Fig. 16 shows typical balances. If the condensing unit and cooling coil have been properly balanced for the required load and, due to miscalculated duct resistance or improper choice of fan speed, the air quantity is reduced, the total cooling capacity will also be reduced. The decrease is generally in the sensible capacity. This is the effect also when the air by-pass or volume control is used.

It is necessary that not only the total capacity but also the sensible and latent cooling requirements both be met. The installation of an excess of coil will result in an increase in total capacity, but not a proportional gain

in latent heat capacity. On installations controlled from dry-bulb temperature the operating time will be shortened because of the added sensible cooling capacity. The result will be less moisture pick-up than calculated, and higher relative humidity. If an oversize condensing unit is installed the opposite situation will take place. The relative humidity will be lower than estimated. This is not generally a disadvantage except that it results in a greater load from outside air than calculated, as well as in increased power consumption. Balances to illustrate these cases appear in Fig. 17. If oversize equipment is furnished, a balance should be made to assure that the ratio of total to sensible capacity is the same as in the estimated load.

Sometimes arbitrary air quantities are specified for ventilation or other reasons independent of the selection of the cooling coil. As shown in

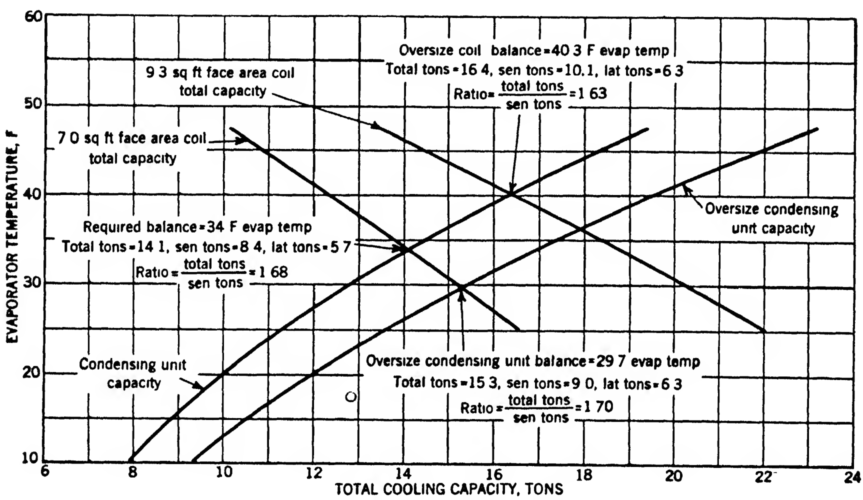

FIG 17. BALANCING CONDITIONS FOR EXCESS COIL AND CONDENSING UNIT CAPACITIES

Table 3, the coil selection can be altered to take care of various air quantities for the same duty.

Where coil and condensing unit are selected for the peak load condition, and the sensible load partially disappears due to fall of outside temperature or other cause, the condensing unit and coil rebalance. This may result in more sensible capacity than required at the light load condition and less latent in proportion, with an increased relative humidity in the conditioned space. Such a condition, for a typical installation, is shown in Table 4. If approximately 40 per cent of the total air is by-passed, the condition will be improved as indicated. The situation could be entirely avoided by using reheat. With sufficient reheat, it is possible to handle any ratio of sensible and latent loads and maintain the design temperature and humidity.

Care should be taken to avoid freezing at light loads. In general, freezing occurs when the coil surface temperature falls to 32 F. With

TABLE 4. CAPACITY BALANCES FOR MAXIMUM AND MINIMUM LOAD CONDITIONS

Conditions	Capacity in Tons			Ratio Total/Sensible
	Total	Sensible	Latent	
Required at peak load conditions	10 90	7.90	3 00	1 38
Required at minimum load conditions	6 62	3 36	3 26	1 98
Peak load equipment balance	10 90	7 90	3.00	1.38
Same equipment balanced at minimum load conditions	9 85	6 58	3 26	1.50
Same equipment balanced at maximum load conditions with 40 per cent by-pass	8 38	5 05	3 33	1.66
Same equipment balanced at minimum load conditions with 38,800 Btu per hour reheat	6 62	3 36	3 26	1 98

usual coils for comfort installations, this will not occur unless the evaporating temperature at the coil outlet is about 20 to 25 F. The exact value depends on the design of coil and the amount of loading. Although it is not customary to choose coil and condensing units to balance at low temperatures at peak loads, there is danger of this occurring when the load decreases. This is further aggravated if a by-pass is used so that less air is passed through the coil at light loads. It may be even worse if the control is arranged for decrease of inside temperature with fall of that outside. Freezing can be avoided by making the full load balance a high evaporating temperature and checking the balance at the minimum load condition.

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PROBLEMS IN PRACTICE

1 ● Why are extended surface coils used in preference to plain tube coils and where is the advantage greatest and least?

Extended surface is used to increase the flow of heat between an air stream and the external surface of a tube, where the resistance through the tube wall and from tube wall

to medium flowing in the tube are proportionately low. Where the latter resistances are very low as compared to the external resistance, as with dry air flowing over tube containing water at high velocity, it is economical to use large ratios of extended to tube surface. Where the external resistance is already low, as where large quantities of water are sprayed over the surface, lower ratios are used, sometimes even plain tube coils.

2 ● On what one feature of construction does the performance of extended surface coils largely depend?

An intimate and permanent bond between tube and fin

3 ● What is the maximum number of fins per inch and why?

Eight fins per inch is the usual practical maximum. A closer fin spacing causes trouble with dirt accumulation and by holding up water between the fins when on dehumidifying duty.

4 ● What is the purpose of the liquid distributor in volatile refrigerant coils?

The distributor is used to apportion the incoming liquid and gas mixture as evenly as possible between the individual refrigerant circuits through the coil, at the same time making it practical to arrange these circuits for a low refrigerant pressure drop.

5 ● What is the usual practice as to the relative direction of flow of air over the tubes and the liquid or vapor in the tubes?

With water and brine, counter-flow is used wherever possible. Volatile refrigerant coils are generally arranged for parallel-flow although cross-flow and combination arrangements are also used. Steam heating coils are usually arranged for cross-flow.

6 ● What factors influence the transfer of heat from the air to fluid in the tubes?

The magnitude of the propelling force or temperature difference. The design and surface arrangement of the coil The velocity and character of the air stream. The velocity and character of the fluid in the tubes.

7 ● What effect has air velocity on the heat transfer?

It has been found that the relation between total heat transfer coefficient and mass air velocity can be represented as a straight line on logarithmic paper. The slope varies with the coil design, number of rows deep and the fluid in the tubes.

8 ● Why do practical coil applications sometimes fall short of rated capacity?

To attain rated capacity it is necessary that the incoming air be uniformly distributed over the coil face. Sometimes this is not attained due to poor layout of apparatus and air connections.

9 ● What are the three general conditions of operation to which cooling coils are subjected in practice?

The coil may be entirely dry, part dry and part wet, or wet throughout, the conditions being based upon whether or not any, part or all of the coil surface is below the dew-point temperature of the air passing over it.

10 ● What is the load-ratio line for a cooling and dehumidifying coil?

It is a straight line drawn on a non-logarithmic psychrometric chart through points representing the entering and leaving air conditions. The slope of this line gives directly the relationship between the sensible and latent cooling corresponding to the selected entering and leaving air conditions

11 ● Upon what does the selection of a heating or cooling coil depend?

An economic analysis of each case should be made together with the relative importance of space, air resistance and quantity, and available or desirable temperatures of heating and cooling media. For a specific design of coil there are a number of possible arrangements for the same duty.

12 ● Can a cooling and dehumidifying coil be selected for any arbitrarily specified conditions of entering and leaving air?

No. Although sensible and latent cooling effects may be obtained simultaneously, and in varying ratios, there is a limiting ratio of total to sensible cooling effect, for any given entering air conditions which cannot be exceeded without the use of reheat regardless of the coil design or refrigerant temperature.

13 ● What precautions must be taken in selecting a condensing unit and cooling coil to operate together?

The total refrigerating capacity and the ratio of total to sensible heat load will be determined by the evaporating temperature at which the coil and condensing unit balance It is possible that, unless a check is made at this balance point, the resultant ratio of total to sensible may differ considerably from that required. An oversized coil, for instance, is likely to result in the desired dry-bulb temperature but excessive relative humidity of the air discharged

14 ● What precautions as to the air circuit should be taken to assure the proper performance of a coil?

The air conditions should be adjusted to obtain those upon which the selection was based, the filters should be kept clean, and the incoming air should be distributed evenly over the coil face.

15 ● What are the usual causes of freezing in the case of a volatile refrigerant coil?

Freezing is caused by a reduction or stoppage of air flow due to dirty filters, clogged coil or sticking of automatic dampers. Or the selection of equipment may have been made for so low an evaporating temperature at peak load that the balance between condensing unit and coil at light loads causes the coil temperature to fall below 32 F

Chapter 25

SPRAY EQUIPMENT FOR HUMIDIFICATION AND DEHUMIDIFICATION

Air Washers, Apparatus for Direct Humidification, Spray Generation and Distribution, Self-contained Humidifiers, Atmospheric Water Cooling Equipment, Design Wet-bulb Temperatures, Cooling Ponds, Spray Cooling Towers, Natural Draft Deck Type Towers, Mechanical Draft Towers, Winter Freezing

AIR humidification is effected by the vaporization of water which always requires heat from some source. This heat may be added to the water prior to the time vaporization occurs or it may be secured by a transformation of sensible heat of the air being humidified to latent heat as the vapor is added to the air. The thermodynamics of the process are discussed in Chapter 1. Dehumidification consists of the removal of moisture from air and may or may not involve the removal of heat from the air-vapor mixture. With spray equipment dehumidification of air necessitates the removal of heat.

AIR WASHERS

Air washers may be used as either humidifiers or dehumidifiers depending upon the method of their operation and the temperature of the spray water. The functions of an air washer are to regulate the moisture and heat content of air passing through it and to remove dust and dirt from the air. As cleaning devices air washers are not as effective as air filters in the removal of dust and dirt.

The construction of commercial air washers is indicated in Figs. 1 and 2. Any air washer consists essentially of a chamber through which the air passes and comes in intimate contact with water. This chamber may be built of either wood, stone, or sheet metal; the latter being the almost universal material of construction. The lower portion of the washer chamber serves as a sump for the water passing to its bottom.

Contact between the air and the washer water is secured: (1) by breaking the water into a very fine mist, (2) by passing the air over surfaces which are continuously wetted by water, or (3) by a combination of water sprays and wetted plates. Scrubber-plate types of washers are

used largely to wash heavy reclaimable products from the air, and are generally composed of one to three eliminator-type baffle scrubber plates across the air stream. Water is generally supplied at the tops of the scrubber plates by flooding nozzles placed across the top of the washer. Spray washers have one or more banks of water atomizing nozzles placed in the air stream above the level of the water in the sump. The direction of the water sprays may be against the air stream, with the air stream, or with one bank spraying with the air stream and one bank of nozzles spraying against it. The number of nozzles required depends upon their design, the quantity of air handled, and the arrangement of the nozzles.

Scrubbers generally consist of eliminator-type baffle plates placed in the air stream to cause several reversals of the direction of air flow. The scrubber plates are more effective as air cleaners than as humidifiers.

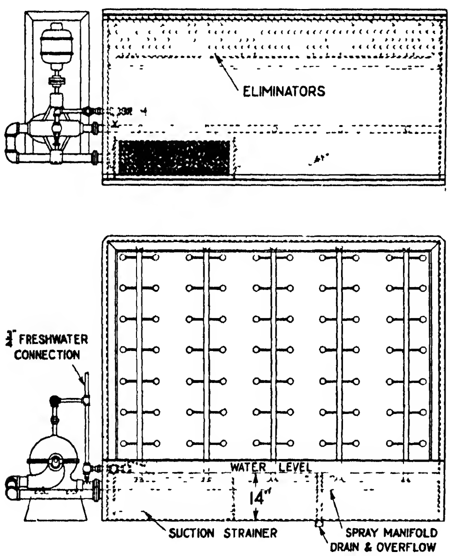

FIG. 1. TYPICAL SINGLE BANK AIR WASHER

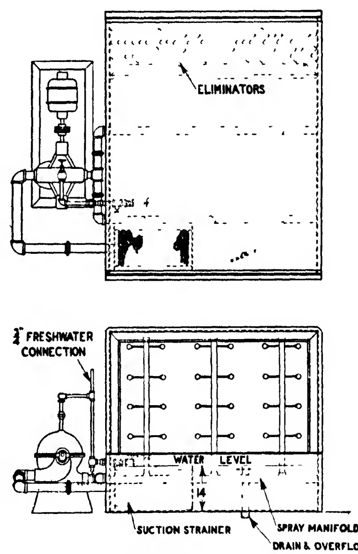

FIG. 2. TYPICAL TWO BANK AIR WASHER

All washer chambers should have inlet diffuser plates to aid in producing more uniform velocities of air flow through the washer spray chamber. These inlet vanes also aid in preventing spray water from being thrown into the air duct ahead of the washer. At the outlet end of the washer suitable flooded eliminator plates, which will cause from 4 to 6 reversals of the direction of air flow, should be installed for the purpose of removing drops of unvaporized water from the leaving air. When the air carries sulphur and gases mixed with it the spray water may become acidulated and special consideration must be given to the selection of eliminator plates to reduce the corrosive action.

Essential items in air washer operation are: uniform distribution of the air across the chamber section above the level of the water in the

sump; moderate velocities of air flow, 300 to 600 fpm in the spray chamber; an adequate amount of spray water broken up into a fine mist throughout the air stream; sufficient length of air travel through the water spray and over thoroughly wetted surfaces, and the elimination of free moisture from the air as it leaves the unit.

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the overall efficiency of heat transfer between the air and the heating or cooling medium. A multi-stage washer is equivalent to a number of washers in a series arrangement. Each stage is in effect a separate washer.

Usually the catalog capacity of a washer is expressed in cubic feet of air per minute and is based upon an air velocity of 500 fpm through the gross cross-sectional area of the unit above the tank. At this rating spray type washers handle about 2½ gpm of water per bank per square foot of area, that is, about 5 gpm per bank per 1000 cfm. These proportions of air, water, area, and velocity may be departed from to meet the needs of some particular job, but certain limiting relationships should be observed.

For a single stage air washer, a 15 F drop in dry-bulb temperature of the air passing through the washer is about the maximum that should be anticipated. For greater decrease in dry-bulb temperature, multi-stage washers should be utilized. A rise of 6 F should be the calculated maximum for the spray water.

The area of a washer may be dictated by space limitations outside the washer, such as headroom, or by the inside space requirements, such as face area needed by a bank of cooling coils. The length of a washer is determined by the number of spray banks, or scrubber plates, and if cooling coils are installed in the unit, by the number of banks of coils. Roughly, a spray space of about 2 ft 6 in. in length is required for each bank of sprays, (the *leaving* eliminators require about 1 ft 6 in., *entering* eliminators about 1 ft).

The resistance to air flow through an air washer varies with the type of eliminators, number of banks of sprays, direction of spray, air velocity, type of scrubber plates, size and type of cooling coils if located in the washer. Manufacturers should be consulted to obtain the resistance for a particular installation.

HUMIDIFICATION WITH AIR WASHER

Air humidification can be accomplished in three ways with an air washer. These are: (1) use of recirculated spray water without prior treatment of the air, (2) preheating the air and washing it with recirculated spray water, and (3) using heated spray water. In any problem of air washing the air should not enter the washer with a dry-bulb temperature less than 35 F so that there will be no danger of freezing the spray water.

When method 1 is used the principles of adiabatic saturation described in Chapter 1 are involved. The process is one of evaporative cooling as the dry-bulb temperature of the air is reduced and the total heat of the air and water-vapor mixture is unchanged. Moisture is added

to the air and a part of the sensible heat of the initial mixture is transformed to latent heat as evaporation of some of the spray water takes place. Theoretically the spray water and the dry- and wet-bulb temperatures of the air should come to the wet-bulb temperature of the air entering the washer and the air should leave the washer adiabatically saturated at the entering wet-bulb temperature. Due to limitations of air washer construction and operation air is not generally completely adiabatically saturated. This introduces an item into the calculations which is known as humidifying or saturating efficiency. This efficiency is the ratio of the actual reduction of dry-bulb temperature to the reduction of dry-bulb temperature theoretically possible. Expressed as a percentage humidifying efficiency is:

$$e_h = \frac{(t_1 - t_2)\,100}{t_1 - t'} \tag{1}$$

where

e_h = humidifying efficiency, per cent.
t_1 = initial dry-bulb temperature, degrees Fahrenheit.
t_2 = final dry-bulb temperature, degrees Fahrenheit.
t' = initial wet-bulb temperature of the entering air, degrees Fahrenheit

The humidifying or saturating efficiency of a washer is dependent upon the number of spray banks and nozzles, the effectiveness of the nozzles in breaking an adequate quantity of water into a fine spray, the velocity of air flow through the water sprays, and the time of the contact of the air with the spray water. Other conditions being the same, low velocities of air flow are more conducive to higher humidifying efficiencies than high velocities of air flow. The following may be taken as representative humidifying or saturating efficiencies of air washers for the conditions stated:

1 bank—downstream	60–70 per cent
1 bank—upstream	65–75 per cent
2 banks—downstream	85–90 per cent
2 banks—1 upstream and 1 downstream	90–95 per cent
2 banks—upstream	90–95 per cent

The air leaving the washer may require the use of a reheater coil to produce the required dry-bulb temperature and relative humidity.

When air of a given specific humidity has a low initial dry-bulb temperature it may be preheated before it enters a washer using recirculated spray water. The preheating of the air increases both the dry- and wet-bulb temperatures and lowers the relative humidity, but not the specific humidity of the air. With an increased wet-bulb temperature, the air is capable of accumulating more moisture by the process of adiabatic saturation and the final specific humidity and the final dry-bulb temperature of the air as it leaves the washer will be higher. An addition of sensible heat by the preheater takes place prior to the air entry into the washer. In the case of method 2 the process of humidification within the washer is similar to method 1. The final desired conditions are secured by adjusting the wet-bulb temperature of the entering air and the use of a reheater when such is necessary.

Method 3 involves heating the spray water to a temperature equal

to the dew-point temperature of the air at the final desired conditions. The water heater may be located either in the washer sump or external to it as in Fig. 3. The air tends to become saturated as it comes in contact with the heated spray water and if the humidifying efficiency of the washer is 100 per cent, the air will leave the washer saturated at the spray-water temperature. Reheating of the air will give the necessary final dry-bulb temperature and relative humidity of the air if the spray water has been maintained at the proper temperature. In this process both heat and moisture are added to the air as it passes through the washer and the dry-bulb temperature of the leaving air is greater than its entering dry-bulb temperature.

Example 1. Air is to be maintained at 70 F with a relative humidity of 40 per cent when the outside air is at 0 F and 70 per cent relative humidity and a barometric pressure of 29.92 in. of Hg. Find the required temperature of the spray water, weight of water vapor to be added, and the heat added in the process

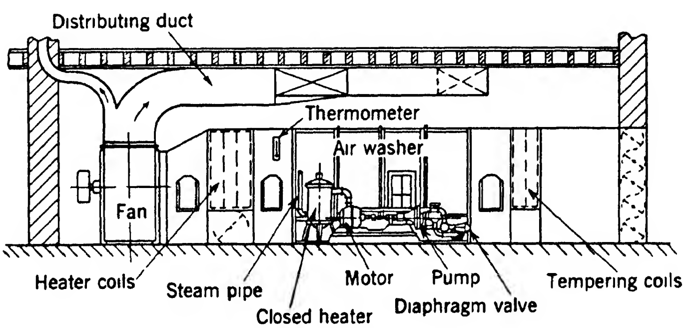

FIG. 3. AIR WASHER WITH SPRAY WATER HEATING ARRANGEMENT

Solution. From Example 2, Chapter 1, the final dew-point temperature is 44.5 F, W_1 = 0.000548, W_2 = 0.00618, and $W_2 - W_1$ = 0 005632 lb per pound of dry air. Therefore, the spray water temperature should be maintained at 44.5 F and the moisture addition per pound of dry air is 0.005632 lb. Assuming that the air after being preheated enters the washer at 40 F and that it leaves the washer saturated at 44 5 F the heat added per pound of dry air is:

Preheater:	[0.24 (40 − 0)] + [0 45 × 0.000548 (40 − 0)] =	9.61
Washer:	[0.24 (44.5 − 40)] + [0.45 × 0.000548 (44.5 − 40)]	
	+ (0 005632 × 1079.2) =	7.16
Reheater:	[0.24 (70 − 44 5)] + [0.45 × 0.00618 (70 − 44.5)] =	6.19
		22.96

If the make-up spray water enters the heater at 40 F its initial heat of the liquid per pound of dry air is 0.005632 × 4.5 = 0 03 Btu making the total heat per pound of dry air, excluding pipe losses, chargeable to the heaters equal to 22 96 − 0.03 = 22.93 Btu The chargeable heat for humidification alone is (0.005632 × 1079 2) + [0 45 × 0.005632 (70 − 44.5)] − (0 005632 × 4.5) = 6.11 Btu per pound of dry air.

APPARATUS FOR DIRECT HUMIDIFICATION

Humidifiers may be divided into the following general types, according to the method of operation: (1) *Direct*, which spray into the room; (2) *Indirect*, which introduce moistened air; and (3) *Combined* direct and indirect.

As in the cases of humidification by use of an air washer the heat necessary for the vaporization of the moisture added to the air is secured either from heat stored in the spray water or by a transformation of sensible to latent heat in the air humidified. In the latter case the total heat of the air remains constant but the dry-bulb temperature of the air humidified is reduced.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where *hydraulic separation* is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers, all of which rely upon compressed air as the atomizing and distributing agency, similar to the familiar method used in ordinary nasal atomizers. Compressed air (ordinarily about 30 lb per square inch) is supplied from a centrally-located air compressor through pipe lines to the atomizing units. The air lines are usually horizontal and parallel to water lines which supply water by gravity from a float tank. The water in the tank is maintained at a constant level slightly lower than the outlets of the atomizers themselves and is drawn constantly to the atomizer by aspiration when compressed air is supplied. This aspiration ceases and the flow of water stops when the air supply is cut off. The water should not be supplied under pressure to atomizers because of the possibility of leakage, drip, or coarse spray which cannot be permitted when water is supplied by aspiration.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located

pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled to the wet-bulb temperature. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is quickly evaporated and the resulting vapor is rapidly and thoroughly diffused. This effective distribution of fine spray over the maximum possible area insures complete and extremely rapid vaporization even at the highest humidities.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

The spray and high-duty types of humidifiers have many features in common but the latter, because of its finer spray and greater capacity, is often considered better adapted for producing high humidities.

Self-Contained Humidifiers

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

Where large quantities of power are generated in a limited space and where a comparatively high relative humidity is required, it is often feasible and economical to use a combination of direct and indirect humidification. The indirect humidification provides the desired quantity of ventilation and cooling, and the additional direct humidification provides for increase in humidity without interfering with the ventilation or the cooling effected by the indirect system.

In general, it may be stated that direct humidification is most satisfactory where high humidities are desired but where little cooling, ventilation or air motion is required. Therefore, the indirect system is most applicable where either low or high relative humidities are desired with maximum cooling and ventilation effect. For conditions that require an unusually large amount of heat to be absorbed by ventilation, together with the maintenance of high humidities, it is often preferable to make use of the combination system of indirect and direct humidification. If the indirect system alone were used it would mean an unusually large volume of air to be handled, which might interfere, due to air motion, with production, even though it would result in greater cooling effect. If direct humidification alone were used, no ventilation would be obtained, with consequently higher room temperatures.

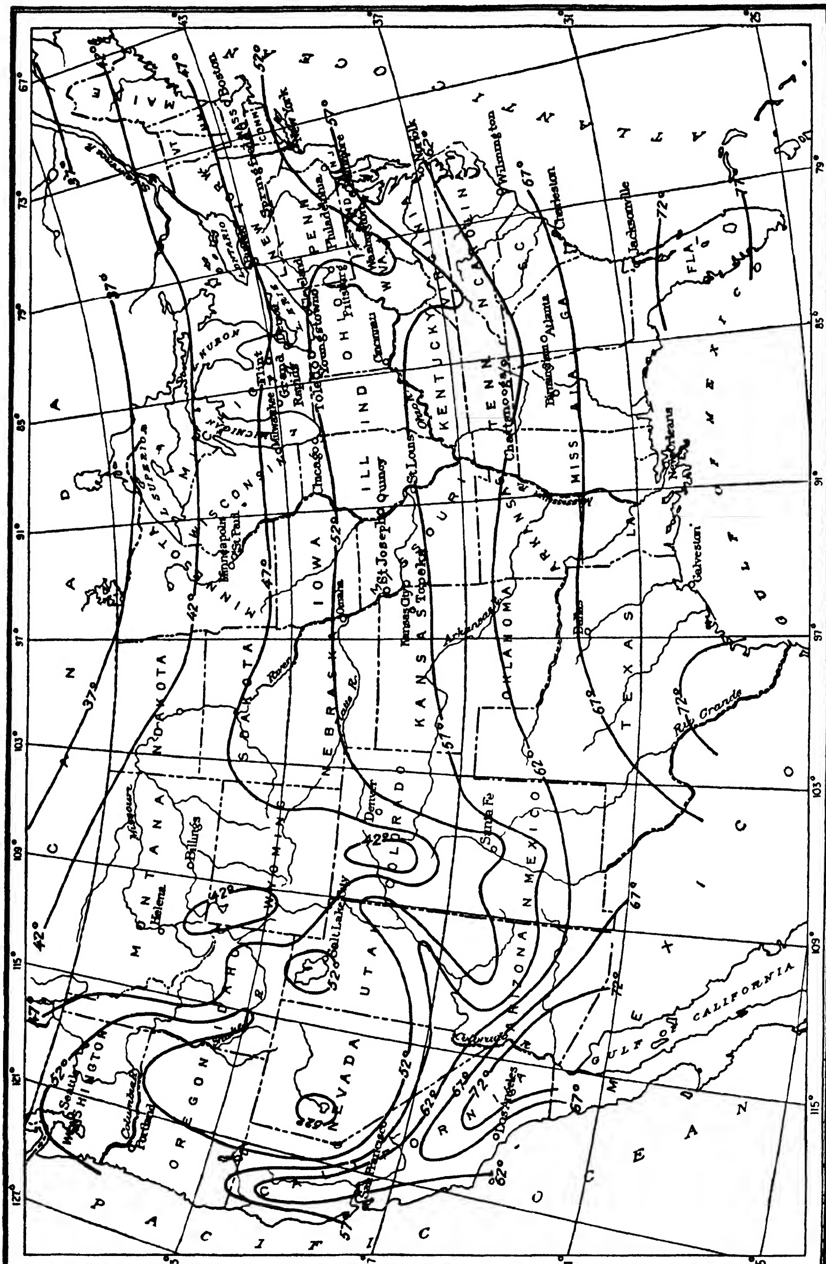

FIG. 4. APPROXIMATE WELL WATER TEMPERATURES AT DEPTHS OF 30 TO 60 FT

AIR DEHUMIDIFICATION WITH WASHERS

Moisture removal from an air-vapor mixture can be accomplished by use of an air washer so long as the temperature of the spray medium is less than the dew-point of the air passing through the unit. The final dry-bulb temperature and the percentage of the saturation of the air leaving a dehumidifier washer are dependent upon: the air velocity, the length of air travel through the sprays, the dry- and wet-bulb temperatures of the entering air, the spray temperature, the number of spray banks and nozzles, the quantity of spray medium handled, and the effectiveness of the nozzles in breaking the spray into a fine mist.

Both sensible and latent heat are removed in the process of dehumidification by cooling. Abstraction of sensible heat occurs during the entire time that the air is in contact with the spray medium. Latent heat removal takes place as condensation occurs. Therefore, the lower the spray temperature the greater the amount of moisture removal per pound of dry air all other conditions remaining the same. The spray temperature should be controlled to 1 or 2 F below the desired leaving dew-point temperature of the air. Washers with two or more banks of sprays are usually selected for comfort air conditioning installations. Such washers will cool the air to within 1 or 2 F of the spray temperature.

Where a limited supply of cold water is available multiple stage washers may be used to an advantage. The cool water is pumped through the multiple spray systems in series. By this arrangement the entering air is cooled first by the warmer water and finally by the cooler water which gives the maximum amount of cooling with the minimum amount of water. The approximate temperatures of water from non-thermal wells at depths of 30 to 60 ft are given in Fig. 4[1]. Frequently the temperature of the city water main supply is low enough during the summer to permit an appreciable cooling effect. Table 1 lists the maximum city water main temperatures for various localities in this country and Canada.

Air washers using refrigerated spray media generally have their own recirculating pumps. These pumps deliver to the washer sprays a mixture of water from the washer sump, which has not been re-cooled, and refrigerated water. The quantities of each of the portions of the spray medium are controlled by a three-way or mixing valve actuated by a dew-point thermostat located in the washer air outlet.

An illustration of a cooling and dehumidifying calculation is given in Example 3 of Chapter 1.

ATMOSPHERIC WATER COOLING EQUIPMENT

In the operation of a refrigerating plant or a condensing turbine, one of the main problems is the removal and dissipation of heat from the compressed refrigerant or the discharged steam. This is accomplished ordinarily by first transferring the heat of the gas to water in a heat exchanger, from which water it may then be dissipated in a number of ways. If the plant is situated on the banks of a river or lake, an intake

[1]Temperature of Water Available for Industrial Use in the United States, by W. D. Collins (*U. S Geological Survey*, Water Supply Paper No 520 F)

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURES[a]

State	City	Temp. F
Ala.	Birmingham	84
	Mobile	73
Ariz.	Phoenix	81
	Tucson	80
Calif.	Anaheim	60
	Berkeley	69
	Fresno	72
	Fullerton	75
	Glendale	68
	Los Angeles	75
	Oakland	69
	Ontario	70
	Pasadena	82
	Pomona	75
	Riverside	78
	Sacramento	72
	San Bernardino	65
	San Diego	82
	San Francisco	62
	Whittier	75
Colo.	Denver	75
Conn.	Bridgeport	66
	Hartford	73
	New Haven	76
	Waterbury	72
D. C.	Washington	84
Del.	Wilmington	83
Fla.	Jacksonville	80
	Miami	80
	Tampa	77
Ga.	Atlanta	87
	Macon	80
Ill.	Chicago	76
	Cicero	76
	Evanston	73
	Peoria	67
	Rockford	59
	Springfield	82
Ind.	Evansville	86
	Gary	75
	Indianapolis	80
	South Bend	61
	Terre Haute	82
Iowa	Cedar Rapids	78
	Des Moines	77
	Sioux City	62
Kans	Concordia	57
	Kansas City	86
	Topeka	88
	Wichita	72
Ky.	Louisville	85
La.	Baton Rouge	85
	New Orleans	85
Me.	Augusta	60
Md.	Baltimore	75

State	City	Temp. F
Mass.	Boston	80
	Cambridge	70
	Fall River	76
	Lowell	50
	Lynn	68
	New Bedford	70
	Salem	68
	Worcester	76
Mich	Detroit	77
	Flint	70
	Grand Rapids	84
	Highland Park	77
	Jackson	56
	Kalamazoo	53
	Lansing	64
	Saginaw	82
Minn	Duluth	55
	Minneapolis	80
	St. Paul	77
Mo	Jefferson City	82
	Kansas City	84
	St. Joseph	84
	St. Louis	85
	Springfield	70
Nebr.	Lincoln	87
	Omaha	87
Nev	Reno	61
N H.	Manchester	76
N. J.	Jersey City	63
	Newark	74
	Paterson	78
	Trenton	79
N. Y.	Albany	68
	Buffalo	75
	Jamaica	56
	Mt. Vernon	74
	New Rochelle	75
	New York	72
	Rochester	70
	Schenectady	60
	Syracuse	74
	Utica	69
	Yonkers	70
N. C.	Asheville	74
	Charlotte	85
	Winston-Salem	82
N. M.	Albuquerque	65
Ohio	Akron	76
	Canton	50
	Cincinnati	84
	Cleveland	74
	Columbus	82
	Dayton	60
	Lakewood	82
	Springfield	72
	Toledo	83

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURE[a] (Concluded)

STATE	CITY	TEMP F	STATE	CITY	TEMP F
Okla	Oklahoma City	82	Utah	Logan	44
	Tulsa	85		Salt Lake City	60
Oreg	Eugene	60	Va	Fredericksburg	75
	Portland	64		Lynchburg	73
Pa	Altoona	74		Norfolk	80
	Erie	75	Wash	Olympia	58
	Johnstown	74		Seattle	62
	McKeesport	82		Spokane	51
	Philadelphia	83		Tacoma	57
	Pittsburgh	67	W Va	Charleston	85
R. I	Providence	68		Huntington	78
S C	Charleston	80		Wheeling	78
	Greenville	81	Wis	LaCrosse	54
	Spartanburg	78		Madison	58
S. Dak	Rapid City	55		Milwaukee	70
Tenn	Chattanooga	84		Racine	68
	Knoxville	89			
	Memphis	70			
	Nashville	90	PROVINCE		
Texas	Amarillo	65			
	Austin	90			
	Beaumont	86			
	Dallas	86	Alta	Calgary	64
	Fort Worth	84	B C	Vancouver	60
	Galveston	90	Ont	London	50
	Houston	84		Toronto	63
	Port Arthur	83	P. E. I	Charlottetown	48
	San Antonio	76	Que	Montreal	78
	Wichita Falls	85		Quebec	68

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown

may be taken upstream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of cooling water is a city supply or a well, the discharge water may be run into the nearest sewer or open waterway. Lacking an unlimited water supply, or in cases where city water is too expensive or where the water available contains dissolved salts which would form scale on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-exchanger by exposure to air in an atmospheric water cooling apparatus.

Air has a capacity for absorbing heat from water when the wet-bulb temperature of the air is lower than the temperature of the water with which it is in contact. The rapidity with which this transfer of heat occurs depends upon (1) the area of water in contact with the air, (2) the relative velocity of the air and water, and (3) the difference between the wet-bulb temperature of the air and the temperature of the water. Because the changes in rate do not occur in direct proportion to changes in the governing factors, data on the performance of atmospheric water cooling equipment are largely empirical.

As the heat content of the air increases, its wet-bulb temperature rises. (See Chapter 1). Because it is impractical to leave the air in contact

with water for a long enough time to permit the wet-bulb temperature of the air and the temperature of the water to reach equilibrium, atmospheric water cooling equipment aims to circulate only enough air to cool the water to the desired temperature with the least possible expenditure of power.

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its moisture and temperature, and it is then sent to the place where it is to be used. In water cooling equipment the temperature of the water is reduced by air, and the cooled water is carried to its point of usage. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess quantity of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design, however, in that the size of the equipment is determined by the quantity of air that must be handled. With the air washer, the size of the equipment is fixed by the quantity of air to be conditioned, and the amount of conditioning is controlled by the quantity and temperature of the water supplied and its method of application. With water cooling apparatus, its size and the quantity of air required bear no direct relation to the quantity of water being cooled, but vary through a wide range for different services and conditions.

Sizes of Equipment

Assuming a definite quantity of water to be cooled, the size and design of atmospheric cooling equipment are affected by the following factors:

1. Temperature range through which the water must be cooled.
2. Number of degrees above the wet-bulb temperature of the entering air to which the water temperature must be reduced.
3. Temperature of the atmospheric wet-bulb at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air).
5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water.

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon:

1. Type of service (refrigerating, internal combustion engine and steam condensing)
2. Wet-bulb temperature at which the equipment must operate satisfactorily.
3. Type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 lb head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 lb and power consumption increases. To obtain this head pressure, the temperature of

the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 or 4 F of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 F difference.

Table 2 lists several gases with data as to the temperatures and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

TABLE 2 CONDENSER DESIGN DATA

Gas	Maximum Pressure Desired in Condenser	Gas Temperature in Condenser Deg F	Leaving Hot Water Temperature Deg F	
			Best Condenser Design	Average Condenser Design
Steam	28 in vacuum	101 2	97	93
Steam	27 in. vacuum	115.1	110	105
Steam	26 in. vacuum	125.9	120	114
Ammonia	185 lb gage head pressure	96.0	92	88
Carbon dioxide	1030 lb gage head pressure	86.0	83	81
Methyl chloride	102 lb gage head pressure	100.0	96	92
Dichlorodifluoromethane	117 lb gage head pressure	100.0	96	93

The temperature range, once the hot water temperature is approximately known, depends upon:

1. Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.
2. Efficiency of the atmospheric cooling equipment considered.

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1934, inclusive, there were but 8 hours per year when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September, inclusive) when the wet-bulb temperature was 68 F or above. As these

975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for refrigeration design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City), with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on a refrigerating plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 1, Chapter 8, shows design wet-bulb temperatures which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must

TABLE 3 EFFICIENCY OF ATMOSPHERIC WATER COOLING EQUIPMENT

Equipment	Cooling Efficiency—Per Cent		
	Minimum	Usual	Maximum
Spray Ponds	30	45 to 55	60
Spray Towers	40	45 to 55	60
Natural Draft Deck or Atmospheric Towers	35	50 to 70	90
Mechanical Draft	35	55 to 75	90

be chosen to place the requirement within the efficiency range of the type of atmospheric water cooling apparatus to be used. Efficiency of atmospheric water cooling apparatus is expressed as the percentage ratio of the actual cooling range to the possible cooling range. Since the wet-bulb temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

Percentage cooling efficiency of atmospheric water cooling equipment =

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}.$$

Efficiencies of various types of atmospheric water cooling apparatus vary through wide limits, depending upon air velocity, concentration of water per square foot of area, and the type of equipment. The commercial range of efficiencies is given in Table 3 although unusual designs may operate outside these ranges.

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various processes of the cooling equipment are:

Compressor refrigeration	220 to 270 Btu per minute per ton.
Condenser turbine	950 to 980 Btu per pound of steam.
Steam jet refrigerating apparatus	1030 to 1150 Btu per pound of steam.
Diesel engine	2800 to 4500 Btu per horsepower

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line. Natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.

Spray Cooling Ponds

The spray pond consists of a basin, above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist because the individual drops must be heavy enough to fall back into the basin and not drift away with the air movement. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in temperature between the water and the wet-bulb temperature of the air, the relative velocity of air and water, and the duration of contact of the air with the water, a much larger quantity of heat may be dissipated in a given area with the spray pond than with the cooling pond, because of (1) the speed with which the drops travel as they are propelled into the air and fall back into the water basin, (2) the increased wind velocity at a point above the surrounding structures or terrain, (3) the increased volume of air used, and (4) the vastly increased area of contact between air and water.

Spray pond efficiencies are increased by (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity nozzles, to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area. Usual practice is to locate the nozzles from 3 to 7 ft above the edge of the basin, to supply from 5 to 12 lb pressure at the nozzles, using nozzles spraying from 20 gpm to 60 gpm each and spacing them so the average water delivered to the surface of the pond is from 0.1 gpm per square foot in a small pond to 0.8 gpm per square foot in a large pond.

Increasing the pressure, spacing the nozzles farther apart, or increasing the elevation of the nozzles will increase the cross-section of spray cloud exposed to the air, and therefore increase the quantity of air coming in contact with the water. Best results are obtained by placing the nozzles in a long relatively narrow area located broadside to the wind.

Spray ponds may be located on the ground if they have an earthen or a concrete basin, or they may be placed on roofs having special waterproof roofing. To prevent excessive drift loss, or the carrying of entrained water beyond the edge of the pond by the air on the leeward side, louver fences are required for roof locations and for those ground locations where space is so restricted that the outer nozzles cannot be located at least 20 ft to 25 ft from the edge of the basin. Such fences usually are constructed of horizontal louvers overlapping so the air is forced to turn a

corner in passing through the fence, and the heavier drops of water are thrown back, owing to their inertia. The louvers also restrict the flow of air, particularly at the higher wind velocities, and thus further reduce the possibility of water being carried off. The height of an effective fence should be equal to the height of the spray cloud. Louver boards are preferably of red gulf cypress or California redwood supported on cast-iron, steel or wood posts. Where building ordinances forbid the use of combustible materials, sheet metal is customarily used.

Algae growths, during warm weather, in cooling towers and spray ponds may be eliminated while the plant is in operation by the use of potassium permanganate. This chemical can be dissolved at the rate of 1 lb in $1\frac{1}{4}$ to $1\frac{1}{2}$ gal of hot water. About 10 parts of permanganate should be used per million parts of cooling water.

The permanganate attacks the algae, forms a brown covering over it, and causes it to settle. Enough of the permanganate solution should be added periodically to cause the water to have a pink color for a period of from 15 to 20 min. Small additions of the permanganate daily do not give concentrations which are effective. The best results are obtained when sufficient quantities are added periodically at intervals of several weeks, the time intervals being dependent upon local operating conditions. The chemical is non-poisonous and is non-corrosive when used as directed.

Spray Cooling Towers

Where not more than 30,000 Btu per minute are to be dissipated, the spray cooling tower is a satisfactory apparatus. The word *tower* in this connection is somewhat of a misnomer as the apparatus is essentially a narrow spray pond with a high louver fence. As usually built, the nozzles spray down from the top of the structure and the distance from the center of the nozzle system to the fence on either side is not more than half the distance that the nozzles are elevated above the water basin. Heights range from 6 ft to 15 ft and the total width of a structure is not usually greater than its height. Spray cooling towers occupy less space on small jobs than spray ponds of equivalent capacities because the towers have a capacity of from 0.6 gpm to 1.5 gpm per square foot of tower area. The louvers are continually wet, and so add to the surface of water exposed to the cooling air.

Natural Draft Deck Type Towers

In past years most of the atmospheric water cooling on refrigeration work has been done with natural draft deck type towers, which are also referred to as *wind* or *atmospheric* towers. These towers consist of heavy wooden or steel framework from 15 to 80 ft high and from 6 to 30 ft wide, having open horizontal lattice-work platforms or decks at regular intervals from top to bottom, and a catch basin at the foot. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and it drips from deck to deck down to the basin. The object of the decks is to arrest the fall of the water so as to present efficient cooling surfaces to the air, which passes through the tower parallel to the decks. The decks also add to the area of water surface exposed to the air, but since they furnish a resistance to air flow, too many decks are a detriment.

To prevent the loss of water on the leeward side of the tower, wide splash boards are attached at regular intervals from top to bottom. These boards or louvers extend outward and upward, and in most designs the top edge of each louver extends above the bottom edge of the one above it.

Efficiency of a deck tower is improved, within limits, by increased height, increased length, or increased width. The first two increase the area of water exposed to the wind, and the latter increases the time of contact of the air with the water.

Wind Velocities on Natural Draft Equipment

Since natural air movement is the prime requirement for a deck type tower, spray cooling tower, or spray pond, the apparatus must be designed to produce the desired cooling on days when the wind velocity is below average when the wet-bulb temperature is at the maximum chosen for design, and when the plant is operating at full load. The apparatus must also, for best results, be located with its longest axis at right angles to the direction of the prevailing hot weather breeze. Table 1, Chapter 8, gives the average summer wind velocities and directions in representative cities. Natural draft cooling equipment should be designed to operate properly with *not more than one-half* of the *average* wind velocity, and in no case for a wind velocity of more than 5 mph. It is obvious that natural draft towers and other natural draft equipment must be so located that they are not obstructed by trees, buildings, or other wind deflectors.

Mechanical Draft Towers

Mechanical draft towers usually consist of vertical shells, constructed of wood, metal, or masonry, in which water is distributed uniformly at the top and falls to a collecting basin at the bottom. The inside of the tower may be filled with wood checker-work over which the water drips, or the water surface may be presented to the air by filling the entire inside of the structure with spray from nozzles. Air is circulated through the tower from bottom to top by forced or induced draft fans. Since the air flows counter to the water, the air is in contact with the hottest of the water just before leaving the top of the tower, and each unit of air picks up more heat than a similar unit would on natural draft equipment, so the mechanical draft tower cools water by using less air than the other types of equipment need. As movement of the air through the towers is obtained by power-consuming fans, it is essential that the air used be reduced to a minimum so as to secure the lowest possible operating cost.

The efficiency of a mechanical draft tower is increased by increasing height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water without affecting seriously the fan power required, but it increases the pumping power needed. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of lowered velocities it increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet-bulb temperature of the air is increased, and

this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air quantity. Air velocities through mechanical draft towers vary from 250 to 600 fpm over the gross area of the structure.

Mechanical draft water cooling equipment may be set up inside buildings, where it usually draws its air supply from the general space in which it is installed, and discharges its exhaust air through a duct to the outside. Indoor cooling towers may be either of the wood-filled or the spray-filled type. In many cases where little height but considerable area is available, water is cooled in a spray-filled structure similar to an air washer, with the air passing horizontally through the apparatus and being discharged

TABLE 4. COMPARISON OF VARIOUS TYPES OF ATMOSPHERIC WATER COOLING EQUIPMENT

Figures indicate order of desirability

	Cooling Pond	Spray Pond	Spray Tower	Deck Tower	Mechanical Draft	Indoor Tower
Cost	x	2	1	3	4	5
Area	5	4	3	2	1	x
Height	1	2	3	4–5	4–5	x
Weight per square foot	x	x	1	3	4	2
Independence of wind velocity	6	3	4	5	1–2	1–2
Drift nuisance	1	6	5	4	2–3	2–3
Make-up water required	1	6	5	4	2–3	2–3
Pumping head	1	2	3	4–5	4–5	6
Maintenance	2	1	3	4	5	6
Suitability for congested districts	x	5	4	3	1	2
Water quantity required for definite result	6	5	4	1–2	1–2	3

xNot comparable

through a duct to the outside. Such apparatus does not have the counterflow advantage of the vertical mechanical draft water cooling equipment, and therefore requires a much larger excess of air for proper operation. Air velocities and operating powers are considerably above those required by vertical mechanical draft water cooling equipment.

Make-Up Water

Since the atmospheric water cooling equipment performs its functions chiefly by evaporating a portion of the water in order to cool the remainder, there is a continual drain on the quantity of water in the system, and this loss must be replaced. Approximately 1 gal of water is lost for every 1000 gal of water cooled per degree of cooling range; so if 1000 gpm of water are cooled through a 10 F range, 10 gpm of water will be required to replace evaporated water. Replacement supply is usually regulated by a float control valve. Because the evaporation of the water leaves behind the salts which the water contained, high concentration of salts may make chemical treatment of the make-up water necessary to avoid excessive deposits in the condensers. An additional amount of make-up water must be added to replace windage, or *drift loss.* This additional amount of water varies from 0.1 to 3 per cent of the quantity of water being circulated, this percentage depending upon the type of equipment and the wind velocity.

Winter Freezing

If atmospheric water cooling equipment is operated in freezing weather, the water may be cooled below freezing temperature so ice forms and collects until its weight causes damage. To obviate freezing during continued operation, the efficiency of the apparatus may be lowered. This is done on the spray pond and the spray cooling tower by reducing the quantity of water fed to the apparatus, thereby lowering the pressure at the nozzles and increasing the size of the drops produced. On the deck tower the upper system may be shut off and a secondary distribution system put in service midway down the height of the tower. The water will be kept above freezing because it will have shorter contact with the air. The mechanical draft tower can be protected by reducing the air flow through the tower, by stopping or reducing the speed of the fans, or by partially closing dampers.

If the system is operated intermittently in freezing weather, water in the basin may freeze and the expansion of the ice may do harm. Freezing during intermittent operation can be prevented only by draining the water basin when it is out of service. On small roof installations, a tank large enough to hold all the water in the system is often installed inside the building and the basin is drained into this by gravity, the pump suction being taken from this inside tank.

A comparison of various types of water cooling equipment is given in Table 4.

PROBLEMS IN PRACTICE

1 ● What performance tests should be given air washers?

a. Capacity, *b* Resistance, *c.* Visible entrainment of free moisture, and *d.* Humidifying or dehumidifying efficiency.

2 ● What are different types of air washers?

a Spray, *b.* Wet scrubber, and *c* Combination spray and scrubber

3 ● Upon what air velocity are air washers usually rated?

500 fpm through the area above the tank

4 ● What is the difference between direct and indirect humidification?

Direct humidification signifies that the humidifiers are within the space to be humidified with distribution produced by the number of humidifiers. With direct humidification there is relatively little air movement.

Indirect humidification signifies that the air is drawn from the enclosure and passed through the humidifier (air washer) and distributed by means of a duct system.

5 ● Where is direct humidification desirable?

Direct humidification is desirable when high humidity is required accompanied with cooling, ventilation, or air motion.

6 ● Where is indirect humidification desirable?

Indirect humidification is desirable when high humidity is required with simultaneous removal of heat by ventilation.

7 ● Why do cooling towers give best results when the humidity of the air is low?

The cooling of the water by dropping it through the air depends mostly upon the evaporation of the water. If the relative humidity of the air is low, the water vapor will be readily absorbed and carried away, while if the relative humidity is high, its capacity to pick up water vapor is less and the water is cooled less with the same exposure to the air.

8 ● What are some of the advantages and disadvantageous of a forced draft cooling tower compared with a natural draft wind tower?

Advantages: a Does not depend on wind, *b.* Less space required, and *c.* Less drift loss and less make-up.

Disadvantages: a. Higher first cost, and *b.* Higher maintenance cost.

9 ● What wet-bulb temperature for outside air is usually selected in air conditioning design when cooling is to be accomplished?

One which is not exceeded more than 5 to 8 per cent of the time in the locality where the plant is situated.

10 ● Where should the suction connection be placed in a cooling pond?

As far below the surface as possible and as far away from the discharge as practicable.

11 ● What chemical is used to kill algae formation in spray ponds?

Potassium permanganate.

12 ● What is the usual amount of spray water delivered to a cooling pond per square foot of area?

From 0.1 gpm on small sizes to 0.8 gpm on large sizes.

13 ● About how much water is lost by evaporation in atmospheric cooling?

About 1 gal per 1000 gal for each degree of cooling range.

14 ● How is freezing obviated in cooling pond sprays?

The pressure and quantity of water is lowered so that the drops become larger in size and do not freeze so readily.

Chapter 26

AIR CLEANING DEVICES

Air Cleaner Requirements, Classifications, Viscous Type Filters, Unit Filters, Automatic Filters, Dry Air Filters, Air Washers, Methods of Installation, Stack Gases, Settling Chambers, Centrifugal Separators, Industrial Filters, Electrical Precipitators, Exhaust Systems, Air Scrubbers

THE removal of impurities from air brought into a building, or from air recirculated in a building for ventilating or air conditioning purposes is the function of any air cleaning or filtering device. These impurities include carbon (soot) from the incomplete combustion of fuels burned in furnaces and automobile engines, particles of earth, sand, ash, automobile tires, leather, animal excretion, stone, wood, rust and paper, threads of cotton, wool and silk, bits of animal and vegetable matter, bacteria and pollen. Microscopic examination shows that the character of the impurities varies with the locality, but as a rule carbon forms the greater part of them while the total is somewhat proportional to the state of industrial activity and the wind intensity. Additional information on sources of air pollution and the particle sizes of atmospheric impurities will be found in Chapter 4.

AIR CLEANER REQUIREMENTS

To fulfill the essential requirements of clean air, an air cleaner should:

1. Be efficient in the removal of harmful and objectionable impurities in the air, such as dust, dirt, pollens, bacteria.
2. Be efficient over a considerable range of air velocities.
3. Have a low frictional resistance to air flow; that is, the pressure drop across the filter should be as low as possible.
4. Have a large dust-holding capacity without excessive increase of resistance, or have ability to operate so as to keep the resistance constant automatically.
5. Be easy to clean and handle, cleans itself automatically, or else be inexpensive enough to replace when dirty.
6. Leave the air free from entrained moisture or charging liquids used in the cleaner.

The SOCIETY has developed a code[1] which explains how such devices are rated by (1) capacity in cubic feet of air handled per minute, (2) resistance in inches of water at rated capacity, (3) dust arrestance, the

[1]A S H V E Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A S H V.E. TRANSACTIONS, Vol 39, 1933, p 225)

percentage relationship expressing dust removal efficiency at rated capacity, (4) reconditioning power, the energy necessary to operate the mechanism of an automatic air cleaning device, and (5) dust-holding capacity, the amount by weight of standard dust which a non-automatic air cleaning device will retain before reconditioning is necessary.

CLASSIFICATION OF AIR CLEANERS

According to the Code, the following four classifications are given the devices:

Class A. Automatic Type: In general all air cleaning devices which use power to automatically recondition the filter medium and maintain a non-varying resistance to air flow.

Class B. Low Resistance Non-Automatic Type: Air cleaning devices for warm-air furnaces, unit ventilating machines and similar apparatus and installations in which a maximum of not more than 0.18 in. water gage is available to move air through the air cleaning device.

Class C. Medium Resistance Non-Automatic Type: Air cleaning devices for systems in which a maximum of not more than 0 5 in. water gage is available to move air through the air cleaning device.

Class D. High Resistance Non-Automatic Type: Air cleaning devices for the air intake of compressors, internal combustion engines, and the like, where a pressure of 1.0 in. or more water gage is available to move air through the air cleaning device.

Air cleaners may also be classified as follows:

1. According to principle of air cleaning.
   a. Viscous air filters.
      (1) Unit type.
      (2) Automatic type.
   b. Dry air filters.
   c. Air washers.
   d. Electrical precipitators.
2. According to application.
   a. For central fan systems of ventilation and air conditioning. Filters of the automatic or semi-automatic type, as well as the non-automatic viscous unit or dry type are usually recommended and are installed in a central plenum chamber.
   b. For unit ventilators. Filters of viscous unit or dry type, installed at inlet of individual units.
   c. For window installations. Self-contained units consisting of fan and filter, usually dry or viscous type, adapted to be placed in the ordinary window.
   d. For warm-air furnaces. Unit type viscous or dry filters placed in small plenum chamber of warm-air house heating systems.
   e. For compressors and Diesel engines. Unit or automatic type viscous or dry filters, installed at air intake of compressors and Diesel engines.
   f. For compressed air lines. Unit type viscous or dry filters.
   g. For stack gases. Settling chambers, dynamic or electrical precipitators.
   h. For exhaust systems. All types.

Air cleaners may be classified further as follows:

1. For general air conditioning. With the growing congestion of large cities and an industrial growth throughout the entire country, the percentages of foreign material in the air, such as soot or carbon, which are unaffected by an air washer type of air cleaner, have increased. This has brought about the development of

the viscous and dry type air filters which are part of many ventilating and air conditioning systems.

2. For removal of dusts, smokes and fumes from stack gases. Prevention of atmospheric pollution from this source is of ever increasing importance, sometimes forced legally and frequently used in order to obtain increased efficiency.
3. For removal and collection of industrial dusts from the point of their production through exhaust systems.

VISCOUS TYPE FILTERS

The principle of air cleaning used in viscous filters is that of *adhesive impingement.* Dust and dirt in the air, especially soot and carbons, are trapped and retained by successive impingements on coated surfaces. While the arrangements of filtering media and the kind of materials used are almost unlimited, there are certain rather definite requirements for a practical commercial filter.

Investigations in this country and abroad demonstrate that the first impingement of dust laden air on a viscous coated surface removes about 60 per cent of the dust, the next impingement takes 60 per cent of what then remains—that is, 24 per cent—and the next impingement removes 9.6 per cent. To secure maximum efficiency, it is necessary to divide the air into innumerable fine streams, as the more intimately and freely the air is brought into contact with the viscous-coated media the better will be the cleaning.

The binding liquid used with viscous filters should have the following properties:

1. Its surface tension should be such as to produce a homogeneous film-like coating on the filter medium.
2. The viscosity should vary only slightly with normal changes of temperature.
3. It should be germicidal in its action to prevent the development of mold spores and bacteria on the filter media.
4. The liquid should have a high affinity for dust at low temperatures.
5. The liquid should have high capillarity, or ability to wet and retain the dust.
6. Evaporation should not exceed 1 per cent.
7. It should be fireproof.
8. It should be odorless.

Viscous Unit Filters

In the unit type viscous filter, the filtering media are arranged in units of convenient size to facilitate installation, maintenance, and cleaning. Each unit consists of an interchangeable cell or replaceable filter pad and a substantial frame which may be bolted to the frames of other like units to form a partition between the source of dusty air and the fan inlet. Where necessary reconditioning equipment should be installed near each group of unit filters, with hot water and sewer connections provided.

To secure greater dust holding capacity and a practically constant resistance and air volume, the filter media are usually placed in the direction of air flow, with progressively finer filter densities determined by the percentage of dust impinged. This arrangement provides relatively large spaces for the collection of dirt in the front of the filter where the bulk of the dust is taken out without undue increase in resistance, while at the back of the filter the openings are smaller to secure high efficiency in the removal of the finer dust particles.

The resistance of a well-designed unit filter of the adhesive impingement type usually depends upon the velocity at which the air is handled and upon whether the unit is clean or dirty. The cleaning efficiency of the unit is usually highest after it has accumulated a certain portion of its maximum load of dirt because some dust collected in the cell acts as an efficient medium for the further seizing of solids from the air. By periodically cleaning a predetermined number of cells, the resistance and capacity

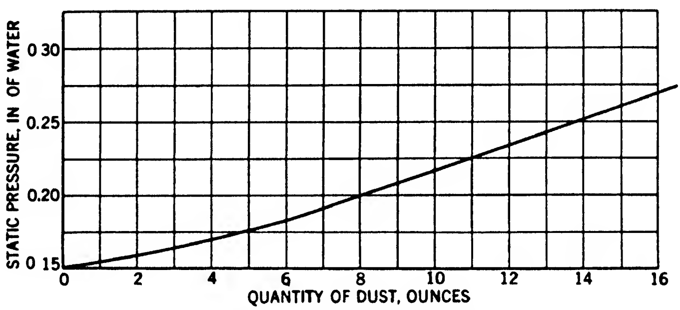

FIG. 1. CHART SHOWING CHANGE IN RESISTANCE DUE TO DUST ACCUMULATION

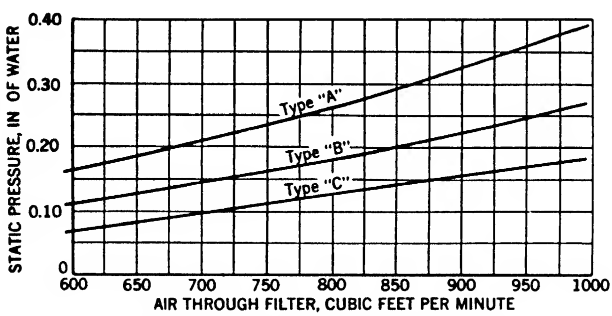

FIG. 2. RESISTANCE TO AIR-FLOW OF TYPICAL UNIT AIR FILTERS

of a built-up filter may be held at any desired figure. The frequency of cleaning any unit filter installation depends upon the dust concentration of air being cleaned, and on the amount of dirt which can be accumulated in the filter medium without causing excessive resistance. (Figs. 1, 2 and 3).

It is difficult to satisfactorily compare the cleaning efficiencies of various filter types unless the efficiency ratings are determined under laboratory conditions in accordance with some definite test procedure such as that developed by the SOCIETY.[2] Efficiency tests made in the field with *atmospheric dust* are subject to so many variables that consistent comparisons are difficult. Of course there is no *standard atmospheric dust*, as atmospheric dust varies widely in composition and concentrations in different

[2]Loc. Cit. Note 1.

localities. Wide variations are also found due to different seasons of the year as well as the time of day and the direction of the wind. A chart showing the increase in resistance of a unit filter of the viscous impingement type, when tested with the standard test dust described in the code[3], is given in Fig. 1. The resistance to air flow of three typical clean viscous impingement type filters having different media densities is shown in Fig. 2. Type A is a dense pack used in bacteria control; Type B is a medium pack used for general ventilation work, and Type C is a low resistance unit for use where low resistance is the important factor and maximum cleaning efficiencies are not essential. The operating characteristics which might be expected under various dust concentrations with air filters having different dust-holding capacities are illustrated in Fig. 3.

Filters consisting of inexpensive frames of cardboard or similar material filled with viscous-coated glass wool, steel wool or the like are available.

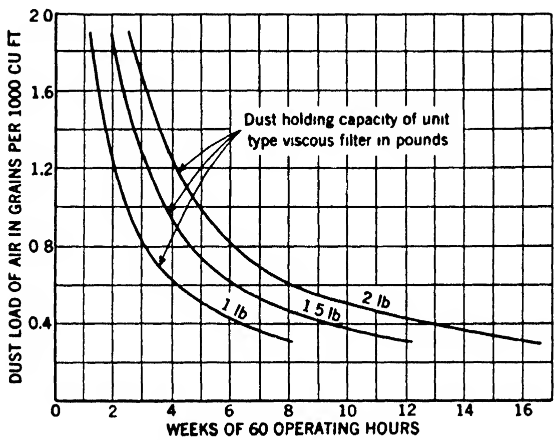

FIG. 3. MAINTENANCE CHART FOR UNIT TYPE VISCOUS FILTERS

Because of their construction these units may be discarded when dirty and replaced with new units at relatively little expense. They are used in general ventilation work and with warm-air furnaces and other installations where low first cost and low resistance to air flow are essential. The operating characteristics of these units conform in general with those of the rigid frame type.

Viscous Automatic Filters

The principle of air cleaning used in the viscous automatic filters is the same as in the unit filters. The removal of the accumulated dust, however, is done automatically instead of by hand. The automatic cleaning and recoating of these filters is based on the principle that the viscous fluid itself will perform the cleaning function, thereby eliminating a separate washing agent. The dust collected by the filter thus is deposited finally in the bottom of the viscous fluid reservoir from which it may be

[3]Loc. Cit. Note 1.

removed by different methods, depending on the design of the filter.

There are three general types of automatic filters. They are differentiated from each other according to the process of self-cleaning and renewing of the viscous coating used by each type, as follows:

1. The filter medium has the form of an endless curtain suspended vertically, with its lower portion submerged in a viscous fluid reservoir. The curtain rotates slowly through this bath, thus performing the cleaning and recoating of the filter medium.

2. The filter screen is arranged in the form of shelves or cylinders, and the viscous fluid is flushed through all parts of the medium in a direction opposite to the air flow.

3. The filter medium is arranged vertically and is stationary. The viscous fluid is flushed from above over the medium, while the air flow is stopped.

The washing and renewing process in automatic filters usually is intermittent. It is accomplished by an electric motor or by other motive power and is controlled by manual or by automatic timing devices. The operating cycle is of a predetermined frequency and should be so timed as to insure a constant static pressure drop across the filter. The customary resistance to air flow is $\frac{3}{8}$-in. water gage at an air velocity of 500 fpm, measured at the filter entrance. Automatic viscous filters are made up in units which are delivered either fully assembled or in parts to be assembled at the point of installation.

DRY AIR FILTERS

Dry air filters, in which dust is impinged upon or filtered through screens made of felt, cloth, or cellulose, are available in various types. These filters require no adhesive liquid, but depend on the straining or screening action of the filtering medium. Because of the close texture of the filtering media used in most of the dry filters, the surface velocity, or velocity of the air entering the media, ranges between 10 and 50 fpm, depending on the nature and texture of the fabric. This necessitates a relatively large screen surface, and the filter media are usually arranged in the form of pockets to bring the frontal area within customary space requirements.

As in viscous unit filters, an average constant resistance and air volume may be obtained by periodic reconditioning or renewal of the filter screens. Since some materials suitable for dry filtering media are affected considerably by moisture which tends to cause a rapid increase in resistance, they should be treated or processed to minimize the effect of changes in humidity.

Filters using felt and similar materials as filter media usually depend upon vacuum cleaning for reconditioning. A special nozzle, operated from a portable or stationary vacuum cleaner, is shaped to reach all parts of the filter pockets. Permanent filter media should be capable of withstanding repeated vacuum cleanings without loss in dust removal efficiency. While most dry filters are cleaned by replacing an inexpensive filter sheet, the useful life of these sheets often may be lengthened by vibrating or vacuum cleaning.

AIR WASHERS

Air washers have not been used extensively in the past in cleaning air for ventilating purposes because of their inability to remove fine dirt

particles. However, new types have been developed which appear to have possibilities for applications where the air to be cleaned is extremely dirty or where a higher degree of cleanliness is desired than can be obtained with a conventionally designed air washer. Information on air washers used in connection with humidifiers will be found in Chapter 25.

METHODS OF INSTALLATION

The published performance data for all air filters are based on *straight through* unrestricted air flow. Filters should be installed so that the face area is at right angles to the air flow whenever possible. Eddy currents and dead air spaces should be avoided and air should be distributed uniformly over the entire filter surface, using baffles or diffusers if necessary.

The most important requirements of a satisfactory and efficiently operating air filter installation are:

1. The filter must be of ample size for the amount of air it is expected to handle. An overload of 10 to 15 per cent is regarded as the maximum allowable. When air volume is subject to increase, a larger filter should be installed.

2. The filter must be suited to the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperatures, and maintenance facilities.

3. The filter type should be the most economical for the specific application. The first cost of the installation should be balanced against depreciation as well as expense and convenience of maintenance.

The following recommendations apply to filters and washers installed with central fan systems:

1. Duct connections to and from the filter should change size or shape gradually to insure even air distribution over the entire filter area.

2. Sufficient space should be provided in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum.

3. Access doors of convenient size should be provided in the sheet metal connections leading to and from the filters.

4. All doors on the clean air side should be lined with felt to prevent infiltration of unclean air. All connections and seams of the sheet metal ducts on the clean air side should be as air-tight as possible.

5. Electric lights should be installed in the chamber in front of and behind the air filter.

6. Air washers should, whenever possible, be installed between the tempering and heating coils to protect them from extreme cold in winter time.

7. Filters installed close to air inlet should be protected from the weather by suitable louvers, in front of which a large mesh wire screen should be provided.

8. Filters should have permanent indicators to give a warning when the filter resistance reaches too high a value.

STACK GASES

Solid particles discharged with stack gases, both domestic and industrial, contribute to the need for air cleaning in general ventilation. The common foreign matter includes the larger fly-ash and unburned carbon particles ranging up to 100 microns and larger, as well as the permanently suspended smokes. Usually it is economical to collect the coarser par-

ticles in separators, either gravitational or centrifugal, thus preventing clogging and overloading of the filters or precipitators used for the fines. Air cleaning devices for this purpose must meet the severe conditions of temperature and corrosion while handling large air volumes at low power and labor costs.

SEPARATORS

In addition to the air cleaning devices previously mentioned, the following are common types available for application to the removal of stack gases.

Gravitational Settling Chambers

The larger dust and gas particles will settle out from air if time and space are provided. Since the settling rate is constant, the required time of retention of the air in a gravitational settling chamber varies directly with the distance through which the particles must fall before reaching a retaining surface. Horizontal plates placed parallel with the air flow are effective and introduce negligible resistance. Air velocities should be selected so that the settled dust will not be redispersed, and for this reason baffles or constrictions producing increased velocity or turbulence should be avoided.

Relations between time of gas passage, distance of fall, and size of particles removed can be calculated from Fig. 1 in Chapter 4. With a forward air velocity of 50 fps passing between horizontal 14 ft shelves placed 3.3 in. apart vertically, particles of 100 microns, which settle at the rate of 59.2 fpm, will all have time to settle through the 3.3 in. vertical distance and reach the shelf while the air is passing along the shelf. Due to redispersion, actual operation would be much less favorable except at low air velocities.

Simple settling chambers consist of large spaces through which air velocities are decreased to one or two feet per second and in which dust particles fall into hoppers. In proper design, the inlets and outlets are placed and baffled so as to cause minimum turbulence, and the collected dust is protected from eddy currents.

Centrifugal Separators

The force causing settling can be increased many times that of gravitation by giving the air a whirling motion and introducing centrifugal force. The settling rate then becomes dependent upon the peripheral air velocity and the radius of curvature as well as upon the other factors.

In centrifugal and cyclone separators, air is introduced tangentially into a vertical cylinder and passes out from the center of the top. The gas velocity and curvature of the cylinder cause whirling which throws the particles to the surface. In the simple centrifugal type, the particles slide down the surface and are removed through a hopper in the cone bottom. In the cyclone type, they are thrown through slits in the periphery and collect in a second outer cylinder where the air is nearly static and there is little chance for redispersion.

Assumptions regarding streamline flow and turbulence make general calculations of centrifugal settling rate quite involved and rough. Their

range of usefulness is indicated in Fig. 1 of Chapter 4. They have wide application in connection with industrial operations such as grinding, screening, combustion, etc., but have little or no effect upon the finer particles.

Small diameters give smoother stream lines and larger centrifugal forces for the same power consumption, so that several small units in parallel are to be preferred to a larger one.

INDUSTRIAL FILTERS

In principle and practice the industrial dry filters are similar to those used for general ventilation, the latter being a development of the former. Bag filters up to 2.5 ft in diameter and 30 ft long, hung vertically, are fed through a header, allowing gas to pass out through the sides of the bag and retaining the dust particles on the inner surface. Depending on the nature of the cloth or mat filtering medium, retention of fines can be very high if gas velocity is low, about 0.5 to 3 cu ft per square foot per minute. The collected dust particles themselves aid in agglomerating and retaining others. Periodic shaking, with the fans off or reversed, at intervals of a few hours drops the excess dust into a lower header or hopper for removal.

Readily removed filters built in small sections in which filter media can be replaced are of distinct advantage where deterioration is rapid. Various styles of construction are available which combine quick interchangeability and large filtering area per square foot cross-sectional area. Use of several independent units in parallel is important for the reconditioning of each unit separately. Both continuous and intermittent shaking and sweeping devices remove excess dust and maintain a low resistance.

ELECTRICAL PRECIPITATORS

For removing fine dust or liquid particles which show no gravitational settling tendency, electrical precipitators are highly effective in air cleaning applications. In this system of air cleaning the particles are first ionized in a region where they acquire an electrostatic charge. The separation of the particles is then accomplished by passing the air between parallel plates where the dust particles are attracted to grounded collecting electrodes. The electric field holds them to this electrode unless a high critical redispersing gas velocity is exceeded. Particles are shaken down by either periodic mechanical or hand rapping.

The materials of construction for the apparatus may be selected to meet nearly any required conditions of temperature and corrosion. The discharge electrodes are usually of metal in the form of wires or edges placed equidistant between collecting electrodes either in the form of hollow pipes or plates. By properly choosing the type of electrodes the generation of oxides of nitrogen may be practically eliminated.

Voltage requirements depend primarily upon the electrode spacing and the gas conditions or particle nature. The maximum field intensity is limited by the arcing voltage for the particular conditions. The discharge electrode should be negative because with this charge higher voltages may be carried without arcing.

EXHAUST SYSTEMS

Quick removal of dust particles produced by such operations as grinding, screening, mixing, etc., is accomplished with exhaust systems. Their applications are numerous and varied, and not only prevent health hazards but eliminate product contamination. Mere discharge to the atmosphere outside of the building without collection is frequently of little effect, for incoming air redistributes the objectionable material. Information on the design of industrial exhaust systems will be found in Chapter 34. Any of the air cleaning devices previously described may be used with them depending upon the severity of the conditions and the nature and size of the material to be collected.

AIR SCRUBBERS

Air scrubbers are used extensively in exhaust systems since they provide removal of at least the coarser particles. The choice of scrubbing medium depends upon the character of the particles to be removed. The liquid medium should wet the particles, and the wetting is a surface tension phenomenon specific for each liquid-solid pair. Water effectively wets particles similar to silica, and oil wets particles similar to carbon. Combinations of oil and water, producing a froth, are effective for both and for materials of intermediate nature.

Intimate contact between the scrubbing liquid and the particles is essential, and many variations in constructions are available. Fine sprays, baffles, bubble caps, open and packed towers, and splash systems are used. Even the finest sprays are of low efficiency when used in an open chamber. Impact of the dust particles against a wetted surface is necessary for their retention, and this requires high gas velocities and well placed baffles or packing. Atomization of the liquid and air together is highly effective in removing the finest particles but makes for high power requirements.

Corrosion is frequently serious, particularly with high temperature gases containing soluble constituents. The collected material is removed as a thick sludge, and its wetted condition is a factor for consideration if it has possible recovery value.

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PROBLEMS IN PRACTICE

1 ● Assume a fan and duct system which handled 10,000 cfm through clean filters with a system resistance of 0.8 in. of water and that after the filters have become dirty the system resistance increases to 1.0 in. of water, and that the fan speed remains unchanged. Is there any way of predicting the volume of air delivered after the filter becomes dirty?

Yes If the performance curves for the particular make of fan are available, the new volume may be determined from the resistance pressure curve (Figs. 1, 2, 3 and 4, Chapter 27)

2 ● What are the advantages of viscous filters?

The principal advantage of the viscous filter is its large dust holding capacity. The dust accumulation is distributed through the depth of the filtering medium rather than upon the surface as in the dry types, which makes it possible for viscous filters to handle heavy dust concentrations without excessive resistance. Since its efficiency and resistance are based on maximum air velocities of from 300 to 500 fpm through the filter, the viscous filter consumes the minimum amount of space for a given air volume.

3 ● What are the advantages of dry filters?

Dry filters are more efficient in the removal of fine dust particles from the air, and some types will eliminate even as much as 60 per cent of the smoke particles. Dry filters also are easily and conveniently maintained by vacuum cleaning, vibrating, or renewing the filtering medium.

4 ● If an air washer is used for cooling and humidity control in an air conditioning system, is a filter needed?

An air filter is desirable in conjunction with an air washer because of the large amount of soot in the air which, due to its greasy and amorphous nature, is not readily trapped in an air washer. Filters should be placed between the washer and the air intake so that all the dirt will be collected at one point to simplify maintenance and to protect all the equipment in the system.

5 ● Is an air filter needed with an extended surface type heat exchanger?

An air filter is essential with an extended surface heat exchanger in order to maintain its efficiency, for without this protection dust particles will adhere to the exposed surfaces, and gradually build up a deposit to the point where the efficiency will be impaired and the resistance increased by restricting the air passage

6 ● What is the proper location of a filter in relation to the fan?

A filter will operate equally well whether placed on the suction or discharge side of the fan. It has become standard practice, however, to locate the filter on the fan inlet side because there it has: (1) simpler duct connections, (2) reduced static pressure losses, (3) more even air distribution over the entire filter area Where an exceptionally high efficiency in dust removal must be maintained, it is often advisable to place the filter on the discharge side of the fan so there can be no infiltration of unclean air.

7 ● What instruments and apparatus are required for determining the pollen concentration in air by means of the settling method?

A microscope with a field of known area and a glass slide coated with a viscous material.

8 ● Describe the procedure for determining the pollen concentration in air by means of the settling method.

A glass slide coated with a viscous material is placed for a period of 24 hours in a horizontal position in the atmosphere to be tested. The slide is then removed and placed under the microscope, and pollen counts are made of approximately 25 fields over the area of the glass slide. Having determined the count over a definite area, as for example, 1 sq cm, and finding the settling rate of the average particles from the chart, Fig. 1 in Chapter 4, the concentration in parts per cubic yard can be calculated.

9 ● The resistance to air flow of a unit air filter is found to be 0.4 in. of water. The volume of air passing through the filter is 1000 cfm at a velocity of 200 fpm. What would be the filter area required in order to reduce the pressure drop across the filter from 0.4 in. of water to 0.16 in. of water?

Referring to Fig. 2 The resistance is substantially proportional to the square of the velocity, or

$$\frac{R_1}{R_2} = \frac{V_1^2}{V_2^2}$$

$$\frac{0.4}{0.16} = \frac{200^2}{V_2^2}$$

$$V_2 = 126.5 \text{ fpm}$$

$$Q = A V$$

$$1000 = 126.5\, A$$

$$A = \frac{1000}{126.5} = 7.91 \text{ sq ft.}$$

The filter area would be increased from 5 sq ft to 7.91 sq ft

Chapter 27

FANS

Classification, Performance, Fan Efficiency, Characteristic Curves, System Characteristics, Selection of Fans, Volume Control, Fan Designations, Motive Power

IN heating and ventilating practice, fans are used to produce air flow except where positive displacement is required, in which case compressors or rotary blowers are used. Fans are classified according to the direction of air flow as (1) *axial flow* or *propeller* type if the flow is parallel with the axis, and (2) *radial flow* or *centrifugal* type if the flow is parallel with the radius of rotation.

Axial flow fans are made with various numbers of blades of a variety of forms. The blades may be of uniform thickness (sheet metal), either flat or cambered, or may be of varying thickness of so-called aerofoil section (airplane propeller type). Where an axial flow fan is intended for operation at comparatively high pressures the hub sometimes is enlarged in the form of a disc and the fan is known as a *disc fan*.

Radial flow or *centrifugal fans* include steel plate fans, pressure blowers, cone fans, and the so-called multiblade fans. All the foregoing types have variations which may be obtained by modification of the proportions or change in the curvature and angularity of the blades. The angularity of the blades determines the operating characteristics of a fan; a forward curved blade is found in a fan having slow speed operating characteristics, while a backward curved blade is found in a fan having high speed operating characteristics.

A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements must be met. In general, fans of all types in each general class can be made to perform the same duty, although mechanical difficulties, noise or lack of efficiency may limit the use to one or another type. The most common field of service for fans of the propeller type is in moving air against moderate resistances, especially where no long ducts or heavy friction must be overcome and where noise is not objectionable, whereas centrifugal fans are commonly employed for operation at the comparatively higher pressures and where extreme quietness is necessary.

FAN PERFORMANCE

Fans of all types follow certain laws of performance which are useful in determining the effect of changes in the conditions of operation. These

laws apply to installations comprising any type of fan, any given piping system and constant air density, and are as follows:

1. The air capacity varies directly as the fan speed.
2. The pressure (static, velocity, and total) varies as the square of the fan speed.
3. The power demand varies as the cube of the fan speed.

Example 1. A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

When the density of the air varies the following laws apply:

4. At constant speed and capacity the pressure and power vary directly as the density.

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.07492 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.06015 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.06015}{0.07492} = 0\ 80 \text{ in}$$

$$\text{Power} = 4 \times \frac{0.06015}{0.07492} = 3.20 \text{ hp}$$

5. At constant pressure the speed, capacity and power vary inversely as the square root of the density.

Example 3. If the speed of the fan of Example 2 is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.07492}{0.06015}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.07492}{0.06015}} = 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.07492}{0.06015}} = 4.46 \text{ hp}$$

6. For a constant weight of air:

(*a*) The speed, capacity, and pressure vary inversely as the density

(*b*) The horsepower varies inversely as the square of the density.

Example 4. If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0.07492}{0.06015} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.07492}{0.06015} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.07492}{0.06015} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.07492}{0.06015}\right)^2 = 6.20 \text{ hp}$$

FAN EFFICIENCY

The efficiency of a fan may be defined as the ratio of the horsepower output to the horsepower input.

The horsepower output is expressed by the formula:

$$\text{Air Horsepower}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356} \tag{1}$$

When the static pressure is used in the computation it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows:

$$\text{Static efficiency}^1 = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{2}$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total or dynamic pressure. This efficiency is variously known as the total, dynamic, or mechanical efficiency, and may be expressed as follows:

$$\text{Mechanical or Total efficiency}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{3}$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different designs the peak of efficiency occurs when the fans deliver different per-

[1]See Standard Test Code for Centrifugal and Axial Fans, Third Edition of 1938.

centages of their wide-open capacity. Variations in efficiency accompany variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans are determined by tests performed in accordance with the Standard Test Code for Centrifugal and Axial Fans[2] prepared jointly by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.* The results of tests are plotted in different ways: the abscissae may be the

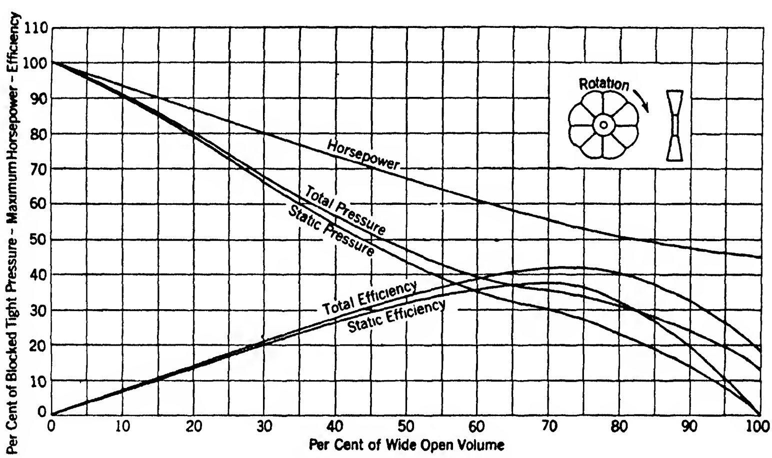


FIG. 1. OPERATING CHARACTERISTICS OF AN AXIAL FLOW FAN

ratio of delivery, assuming full open discharge as 100 per cent, and the ordinates may be static pressure, dynamic pressure, horsepower and efficiency. Pressures may be expressed in per cent of the maximum pressure in the manner shown in the illustrations in this chapter, but in engineering calculations they are sometimes expressed in proportion to the pressures due to the peripheral velocity.

It should be noted that characteristic curves of fan performance are plotted for a constant speed. Some variation in values of efficiency may occur at different speeds but such variation is usually slight within a wide range of speeds. Fans of similar design but of different size will also show some difference in efficiency. Figs. 1 to 4 show characteristic curves for different types of fans using blades of various shapes, but without reference to the design of housing employed. The efficiency curves are therefore not serviceable for making rigid comparisons of efficiencies obtainable with blades of the various shapes but are intended merely to show reasonable values and more particularly to show the manner in which variations occur with changes in fan capacity.

[2]A S H V E TRANSACTIONS, Vol 29, 1923, p 407 Amended in A S H V E TRANSACTIONS, Vol 37, 1931, p 363 Third Edition of 1938

Axial flow fan characteristics are indicated by Figs. 1 and 2. These fans, when properly designed, have a satisfactory efficiency at low resistance, comparing favorably in this respect with centrifugal fans. They are low in cost and economical in operation and occupy relatively little space. Although this type of fan can operate against considerable resistance, the noise often becomes objectionable, so that it does not always compare favorably with centrifugal fans for such service. With most of the designs which employ blades of uniform thickness the power increases rapidly with an increase in resistance.

The curves (Fig. 1) show the rapid reduction in capacity and increase in power as the resistance increases. The low efficiency when overcoming

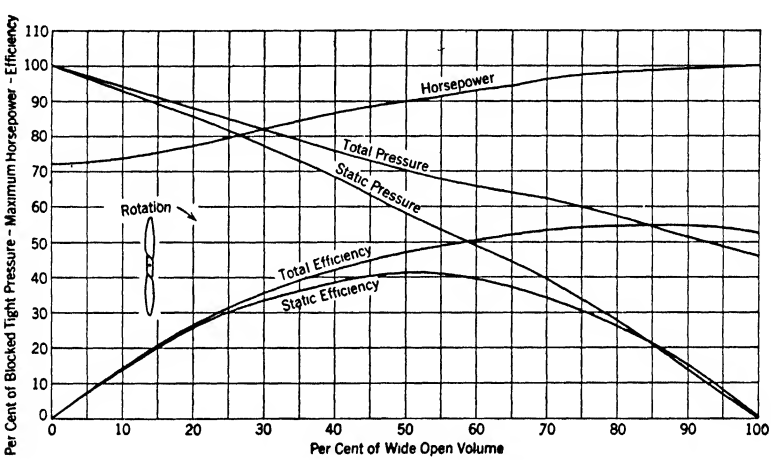


FIG. 2. OPERATING CHARACTERISTICS OF AN AIRPLANE PROPELLER FAN

heavy resistance is due to the low speed of the blades near the hub as compared to the relatively high peripheral or tip speed. The air driven by the blade area near the rim can pass back through the less effective blade area at the hub more easily than it can overcome the duct resistance.

Fig. 2 shows the performance of the *airplane propeller fan* in which the blades are similar in shape to those of an airplane propeller but of varying number according to the pressure to be developed. This fan usually operates at a higher speed than does the former type of propeller fan, and with a different power characteristic, the power remaining fairly constant throughout the range of pressures, being somewhat less at the higher than at the lower pressures. The flatness of the horsepower curve indicates the advantage of this type of fan in preventing overloading of motors where fluctuations in pressure occur. Variations in the diameter, width, pitch, camber, and the thickness of the blades provide a considerable degree of flexibility in design, so that the peak of total efficiency may be made to occur at wide-open volume or at various percentages of that volume.

Another advantage of this type of axial flow fan is its low resistance to air passage when standing still. There are some installations in which such a characteristic is desirable.

The straight blade (paddle-wheel) or partially backward curved blade type of fan is practically obsolete for ventilation. Its use is largely confined to such applications as conveyors for material, or for gases containing foreign material, fumes and vapors. The open construction and the few large flat blades of these wheels render them resistant to corrosion and tend to prevent material from collecting on the blades. This type of fan has a good efficiency, but the power steadily increases as the static pressure falls off, which requires that the motor be selected with a moderate reserve in power to take care of possible error in calculation of duct resistance.

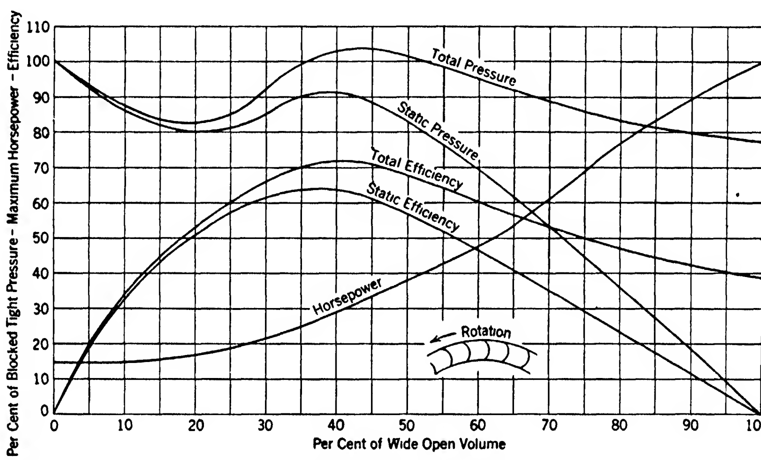


FIG. 3. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

The *forward curved multiblade fan* is the type most commonly used in heating and ventilating work, as it has a low peripheral speed, a large capacity, and is quiet in operation. The point of maximum efficiency for this fan occurs near the point of maximum static pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. The power curve rises continually from low to peak capacity, but if reasonable care is exercised in figuring resistance there is no danger of overloading the motor.

The outstanding characteristics of the *full backward curve multiblade type fan* are the steep pressure curves, the non-overloading power curve, and the high speed. (See Fig. 4.) This fan operates at a peripheral speed of approximately 250 per cent of the forward curve multiblade type for like results. The pressure curves begin to drop at very low capacity and continue to fall rapidly to full outlet opening. The steep pressure curves tend to produce constant capacity under changing pressures. Where wide fluctuations in demand occur, this type of fan is desirable to prevent overloading of motors. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capacities at a given speed. The high speed of this type

makes it adaptable for direct connected electric motor drives. The high speed may necessitate somewhat heavier construction and more operating attention or service. The dimensional bulk for a given duty often is 150 per cent of that of a forward curve multiblade type fan.

Between the extremes of the forward and the full backward curve blade type centrifugal fans a number of modified designs exist, differing in the angularity or in the shape of the blades. Common among these designs are the straight radial blade type, the radial tip type, and the double curve blade type with a forward angle at the heel and a slight backward angle at the tip of the blade. Characteristic curves of these types show

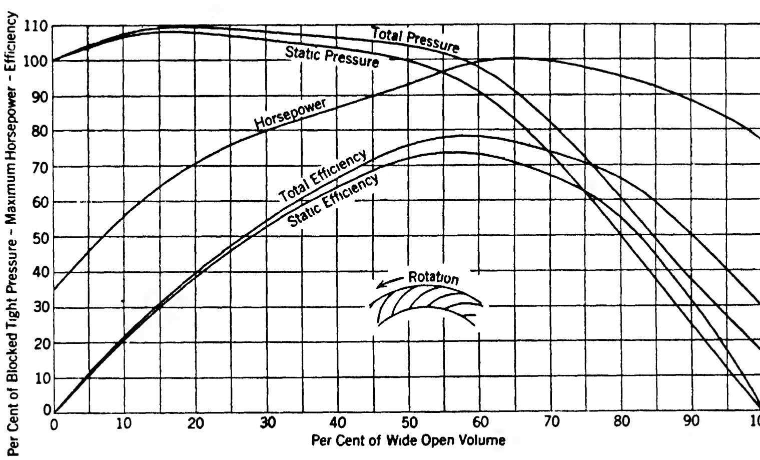


FIG. 4. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

varying degrees of resemblance to the curves of Figs. 3 and 4, according to the degree of similarity to one or the other of the two designs of fan considered.

SYSTEM CHARACTERISTICS

A given fan performs as determined by the real characteristic of the system to which it is attached. When a different performance of a fan is desired, it is necessary to either change the speed of the fan (as *A* to *B* or *C* to *D* in Fig. 5), or to change the system (as by moving a damper from *A* to *C* in Fig. 5). If the speed of the fan is changed, the new point of operation is the intersection of the constant speed static pressure—cubic feet per minute curve for the new speed with the system characteristic. If the system is changed, the new point of operation is the intersection of the constant speed static pressure, cubic feet per minute curve with the new system characteristic.

Heating and ventilating systems follow the simple parabolic law quite closely but other types of systems follow some other more or less complex relation. The more complex systems can be separated into their component parts whose individual characteristics are known and the summation of the characteristics of the several parts of a system will give the composite characteristic of the system

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Type of motive power available.
4. Whether fans are to operate singly or in parallel on any one duct.
5. What degree of noise is permissible.
6. Nature of the load, such as variable air quantities or pressures.

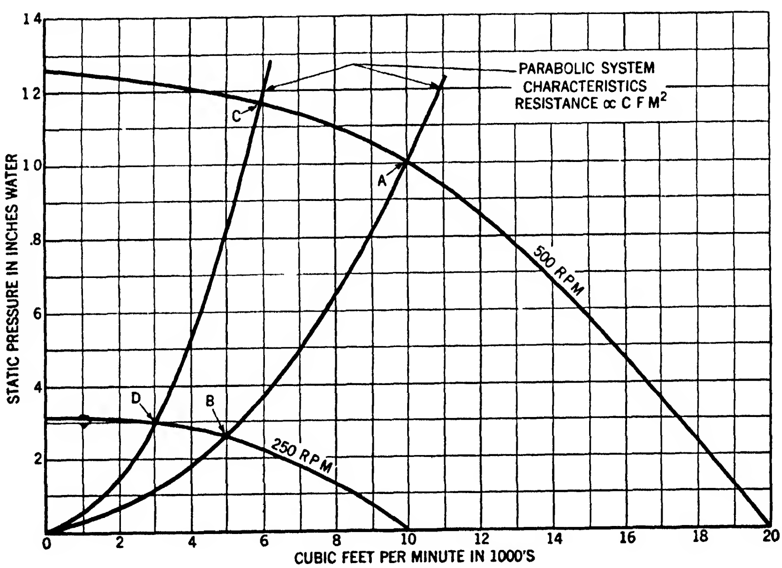


FIG. 5. ILLUSTRATION OF OPERATING POINTS OF A GIVEN FAN AT TWO SPEEDS ON THE SAME AND DIFFERENT SYSTEMS

Knowing the requirements of the system, the main points to be considered for fan selection are (1) efficiency, (2) speed, (3) noise, (4) size and weight, and (5) cost.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures:

1. Volume of air in cubic feet per minute (68 F, 50 per cent relative humidity, 0.07488 lb per cubic foot).
2. Outlet velocity.
3. Revolutions per minute.
4. Brake power.
5. Tip or peripheral speed.
6. Static pressure.

The most efficient operating point of the fan is usually shown by either bold-face or italicized figures in the capacity tables.

Fans for Ventilating and Air Conditioning Systems

Two important factors in selecting fans for ventilating systems are efficiency (which affects the cost of operation) and noise. First cost and space available are secondary. The fans should be selected to operate at maximum efficiency without noise. Because noise in a ventilating system is irritating and a cause for complaint, fans must be selected of proper size in order to reduce it to a minimum. Noise may be caused by other factors than the fan, namely, high velocity in the duct work, unsatisfactory location of the fan room, improper construction of floors and walls, and poor installation. Where noise is chargeable directly to the fan, it is caused either by excessive peripheral speeds, or the fan is of insufficient size. It should be remembered, however, that the tip speed required for a specified capacity and pressure varies with the type of blade, and that a tip speed which may be excessive for the forward curved type is not necessarily so for the backward or slightly backward type. A noisy fan usually is one which is operated at a point considerably beyond maximum efficiency.

TABLE 1. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR FORWARD CURVED MULTIBLADE VENTILATING FANS

Static Pressure Inches of Water	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute
1/4	1000–1100	1520–1700
3/8	1000–1100	1760–1900
1/2	1000–1200	1970–2150
5/8	1100–1300	2225–2450
3/4	1200–1400	2480–2700
7/8	1300–1600	2660–2910
1	1500–1800	2820–3120
1 1/4	1600–1900	3162–3450
1 1/2	1800–2100	3480–3810
1 3/4	1900–2200	3760–4205
2	2000–2400	4000–4500
2 1/4	2200–2600	4250–4740
2 1/2	2300–2600	4475–4970
3	2500–2800	4900–5365

For a given static pressure there is a corresponding outlet velocity and peripheral speed wherein maximum efficiency is obtained. If a fan is selected to operate at this point, the cost of operation and the noise can be held within control.

To aid in selecting fans as near as possible to the point of maximum efficiency, there are listed in Tables 1 and 2 for each static pressure corresponding outlet velocities and tip speeds which will give satisfactory results. The proper tip speed for a given static pressure varies with the design of wheel and with the number of blades or vanes in the wheel.

Lower outlet velocities than those listed in Table 1 may be employed, but care must be exercised to avoid selecting a fan for operation below its useful range. The useful range of the fans of Table 2 extends over the full length of the performance curve.

In exhaust ventilating systems where the air column moves toward the fan, noise due to the higher tip speeds and outlet velocities will not be so readily transmitted back through the air column to the building as when the air column is moving toward the rooms. Therefore higher outlet velocities may be used, but this will be at the expense of increased horsepower.

Amply large fans should always be used for both exhaust and supply systems, as there may be and usually is leakage despite the most careful workmanship, necessitating the delivery of more air at the fans than is exhausted from or supplied through the openings in the various rooms.

Long runs of distributing ducts, heaters, and air washers require definite increments of the total pressure which a supply fan in a ventilating system must overcome. These static pressures should be considered when selecting the fan characteristics, speed, and power.

TABLE 2. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR MULTIBLADE VENTILATING FANS WITH BACKWARD TIPPED AND DOUBLE CURVED BLADES

STATIC PRESSURE INCHES OF WATER	OUTLET VELOCITY FEET PER MINUTE	TIP SPEED FEET PER MINUTE
1/4	800–1100	2600–3100
3/8	800–1150	3000–3500
1/2	900–1300	3400–4000
5/8	1000–1500	3800–4500
3/4	1100–1650	4200–5000
7/8	1200–1750	4500–5300
1	1200–1900	4800–5750
1 1/4	1300–2100	5300–6350
1 1/2	1400–2300	5750–6950
1 3/4	1500–2500	6200–7550
2	1600–2700	6650–8050
2 1/4	1700–2800	7050–8550
2 1/2	1800–2950	7450–9000
3	2000–3200	8200–9850

Fans picked within the limits of Table 1 will operate close to the point of maximum efficiency. No attempt has been made to select these limits for quiet operation, since this is a relative term and varies with the type and location of the installation.

The connection of a fan to a metallic duct system should be made by canvas or a similar flexible material so as to prevent the transmission of fan vibration or noises. Where noise prevention is a factor the fan and its driver should have floating foundations.

Fans for Drying

Both axial flow and centrifugal types of fans are used for drying work. Propeller fans are well adapted to the removal of moisture-laden air when operating against low resistance and when handling air at low temperatures. Motors on these fans usually are of the fully-enclosed moisture-proof types so that saturated air or air containing foreign material will not injure the motors.

Unit heaters employing axial flow fans are widely used in the drying

field. In drying, these fans may be used with unit heaters where not too much duct work is required and where air is to be delivered against pressure, since the noise developed from the high peripheral speed of these fans is not ordinarily objectionable in process work.

Centrifugal fans of the multiblade type generally are selected to supply air for drying, as they are capable of delivering large volumes of air against all pressures likely to be encountered.

Belt driven fans usually are to be preferred to direct-connected fans since efficient motor speeds do not usually coincide with efficient fan speeds. Replacement of a standard motor is quick and easy if it is belted.

Wherever drying is done throughout the year and where air requirements change as the drying conditions change, the drying can be speeded up or reduced through control of the fan capacity. This may be done by changing the fan speed or by varying the outlet area with dampers. A throttled outlet reduces the volume and reduces the power.

Due to the low speeds of forward curved multiblade or paddle-wheel type fans, these can be direct-connected to reciprocating steam engines, and the exhaust steam from the engines may be used in the heating apparatus. In selecting engine driven fans for drying processes, where a large quantity of exhaust steam is used in the heaters, a smaller fan and greater power consumption may be used, because power economy is not essential under this condition.

Where static pressure in a dryer varies, and where several fans must operate in parallel, fans are to be preferred which have a continuously rising pressure characteristic, such as is given by backward-curved or double-curved blades. This type of fan is well adapted for direct-connected motors of the higher speeds. (See Chapter 35 on Drying Systems.)

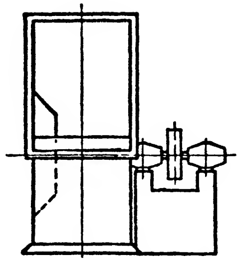
Fans for Dust Collecting and Conveying

The application of fans for handling refuse, dust, and fumes generated by machine equipment is covered in Chapter 34. Information is given regarding the methods for determining air quantities, the velocity required for carrying various materials and the method of determining maintained resistance or total static pressure at which the fan is to operate. The selection of a proper size fan is at times governed by the future requirements of the plant. In many instances, additional future capacity is anticipated and should be provided for.

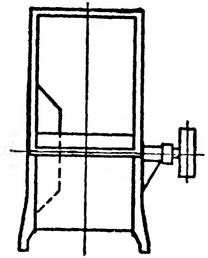
Having determined the necessary volume of air and the maintained resistance or static pressure required, the proper size fan may be selected from the fan manufacturers' performance charts or capacity tables. The fan chosen should be the size that will provide the required ultimate quantities with the minimum power consumption.

FAN VOLUME CONTROL

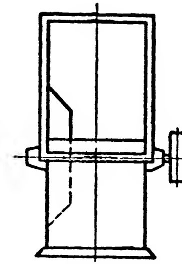
Some method of volume control of fans usually is desirable. This may be done by varying the peripheral velocity or by interposing resistance, as by throttling-dampers. Both methods, since they reduce the volume of air, reduce the power required. In many installations adjustments of volume are desirable during varying hours of the day. In others an



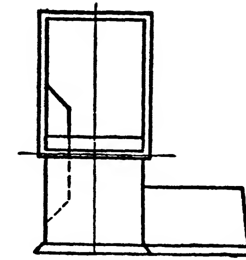
Arr. 1.
For belt drive.
Wheel overhung. Bearings on pedestal.



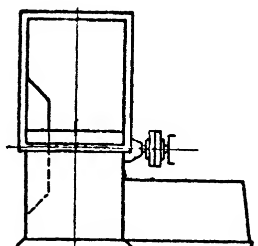
Arr. 2.
For belt drive.
Pulley and wheel overhung Bearings in bracket on fan housing Made only in smaller sizes for reversible discharge



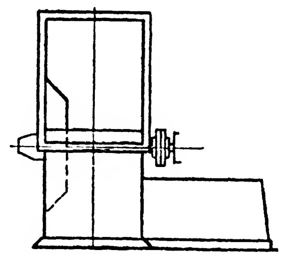
Arr. 3.
For belt drive.
Pulley overhung Bearings supported on fan housing.



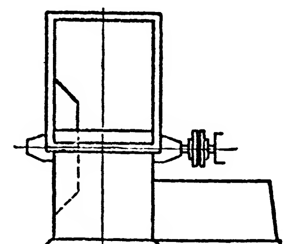
Arr. 4.
For direct drive.
Wheel overhung No bearings on fan. Wheel mounted on motor or engine shaft Pedestal for motor or engine.



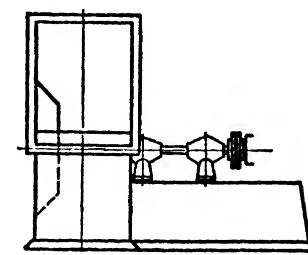
Arr. 5
For direct drive.
Wheel overhung. Includes housing, wheel, shaft, one intermediate bearing, flanged coupling and pedestal only for motor or engine



Arr. 6.
For direct drive.
Three-bearing arrangement with fan bearing at inlet side Includes housing, wheel, shaft, one bearing (in inlet), rigid coupling, and pedestal only for motor or engine



Arr. 7.
For direct drive.
Similar to Arr 6, but with two bearings on fan, and flexible instead of rigid coupling



Arr. 8.
For direct drive.
Similar to Arr. 5, but with two bearings on pedestal with motor, and flexible instead of rigid coupling

FIG. 6. ARRANGEMENT OF FAN DRIVES

increased supply of air in summer over that needed for winter is demanded. Experience is required in deciding whether speed-control or damper-control shall be used for specific cases. Where noise is a factor, it may be exceedingly desirable to reduce the speed at times, while on the other hand, any fan which has its normal speed reduced as much as 50 per cent without *change in resistance* will move only 50 per cent of the air.

FAN DESIGNATIONS

Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is clockwise, the fan, blower, or blast wheel will be designated as *clockwise*. If the proper direction of rotation is counter-clockwise, the designation will be *counter-clockwise*. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.)[1]

This method of designation will apply to all centrifugal fans, single or double width, and single or double inlet. Do not use the word "hand," but specify "clockwise" or "counter-clockwise."

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as follows:

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

Up blast: If the line of air discharge is vertically up.

Down blast: If the line of air discharge is vertically down.

All intermediate discharges will be indicated as angular discharge as follows:

Either top or bottom angular up discharge or top or bottom angular down discharge, the smallest angle made by the line of air discharge with the horizontal being specified.

In order to prevent misunderstandings, which cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 6 are suggested.

If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half. The *backward curved* and *double curved* types with backward tip operate satisfactorily in double or in parallel operation.

MOTIVE POWER

It is no easy matter to predetermine the exact resistance to be encountered by a fan or, having determined this resistance, to insure that no changes in construction or operation shall ensue which may increase air resistance, thus requiring more fan speed and power to deliver the required volume, or which may reduce air resistance, thus causing delivery of more air and a consequent increase of power even at constant speed.

It is recommended, therefore, for centrifugal type fans that the rated power to be supplied shall exceed the rated fan power by a liberal margin, when *forward curved* types are used. When *backward* or *double curved* blade types are used, motors with ratings very close to that of the fan horsepower demand can be employed, provided the fan has a limiting horsepower characteristic.

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[1]Recommendations adopted by the *National Association of Fan Manufacturers*

Justification for liberal power provision exists also in the possibility of varying demand due to changes in ventilation requirements, intensity of occupation, and weather conditions.

The motive power of fans should be determined in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.*

Fans may be driven by electric motors, steam engines (either horizontal or vertical), gasoline or oil engines, and turbines, but as previously stated the drive commonly used is the electric motor.

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PROBLEMS IN PRACTICE

1 ● What information must be supplied to the manufacturer when ordering a centrifugal fan?

a. Size of fan (catalog number).

b. Type of fan.

c. Width of fan (single or double).

d. Number of inlets (single or double).

e. Fan performance and kind of application.

f. Direction of rotation (clockwise or counter-clockwise).

g. Direction of discharge (top horizontal, down blast, etc.).

h. Drive arrangement (see Fig. 6).

i. Style of housing (full, three-quarters, etc.).

2 ● In selecting fans for quiet operation in public buildings:

a. Should the outlet velocity of the fan be limited?

b. Should the tip speed of the fan be limited?

a. Because all commercial fans operating at pressures suitable for this class of work would be considered noisy if the fan were to discharge directly into the room, and because the duct system on the fan discharge is depended upon to absorb a reasonable amount of fan noise, it is desirable to have a moderate run of duct work with some bends and elbows included as sound deadeners. Where this duct is of necessity very short, the outlet velocity must be kept down to the lower limits recommended in this chapter or else an efficient sound absorber must be used. The experience of the engineer must be his guide in determining the allowable outlet velocity in each individual case.

b. Tip speed should not ordinarily be limited, because different types of fan blades have entirely different allowable tip speeds for quiet operation. A fan having a backward blade at the tip can run at much higher tip speed than can a forward curved or a straight blade fan, with the same degree of quietness.

3 ● Is a direct connected or a belted fan preferable in public building work?

Where space is at a premium, direct connection is best. Next in space economy is the short V-belt drive. The flat belt drive fan requires the greatest floor space. In this class of work, pressures are usually so low that even with the high speed fans the motor cost is greater for direct connected units than for belt drive fans.

4 ● a. What type fans are used in industrial work?

b. What outlet velocity is suitable?

a. All of the centrifugal types are suitable; the disc and propeller types are suitable for low pressure work, or they are often used as exhausters.

b. The outlet velocities on fans for industrial work can be much higher than can those in public building work, where quietness is essential. Fans should be selected with outlet velocities as recommended in this chapter, using the upper limit of velocities.

5 ● Are direct connected or belted fans preferred in industrial work?

In industrial applications, fans are often advantageously direct connected to motors. The pressures are usually high enough to use standard motor speeds. The high speed types of fans have limiting horsepower characteristics so that little margin in power must be provided in the driving motor. Belted fans may be used, but where high power is required a special arrangement is often necessary for shaft and bearings on account of the weight of the sheave and the belt pull.

6 ● A forward curved multiblade fan which requires 5.4 bhp is delivering 22,800 cfm at 70 F against a resistance pressure of 1 in. of water at an outlet velocity of 1440 fpm:

a. What is the static efficiency?

b. What is the total efficiency?

a. 66.3 per cent (see Equation 2)

b. 74.5 per cent (see Equation 3).

7 ● If the above fan has a 54-in. diameter wheel and operates at 193 rpm, will it be suitable for a ventilating installation where a minimum of noise is desirable?

Yes. The tip speed will be 2720 fpm and this, together with the 1440 fpm outlet velocity, falls within the limits given in Table 1 for 1-in. resistance pressure.

8 ● What objectionable feature is inherent in the ordinary propeller fan when it is operating at high resistance pressures?

It must operate at a high speed with consequent noise.

9 ● At what point should a fan be selected for operation, and why?

At its point of maximum efficiency because the cost of operation and the noise produced will be least.

10 ● In Fig. 3, a static pressure of 85 per cent of blocked tight pressure corresponds to three different volumes, namely 11 per cent, 30 per cent and 48 per cent of wide open volume. What will determine which volume the fan delivers?

The fan can operate only at the intersection of its pressure-volume curve and the system characteristic. The type of system, together with the specification of the volume at a certain static pressure, completely defines the system characteristic.

As illustrated in Fig. 5, a given system characteristic will intersect the fan curve in only one point.

If the 85 per cent value for static pressure is specified for the 48 per cent value of volume, it is at once obvious that the same system will not have the same resistance at any other volume.

Chapter 28

AIR DISTRIBUTION

Definitions, Grille Locations, Standards for Satisfactory Conditions, Factors Affecting Distribution for Cooling and Heating, Air Outlet Noises, Selection of Supply Outlets, Balancing System

CORRECT air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor. Supplying the proper amount of air is one problem; properly distributing it from the point where it leaves the fan is another. The distribution problem may be further divided into: (*a*) distribution *to* the various spaces served by the system, (*b*) distribution *in* these spaces. This discussion is primarily limited to division (*b*), reference being made to the duct system only insofar as it affects the performance of the air distribution outlets.

Definitions

1. *Supply Opening:* Any opening through which air is delivered into a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

2. *Exhaust Opening:* Any opening through which air is removed from a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

3. *Outside Air Opening:* Any opening used as an entry for air from outdoors.

4. *Grille:* A covering for any opening and through which air passes

5. *Damper:* A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.

6. *Multiple Louvre Damper* A damper having a number of adjustable blades.

7. *Single Louvre Damper:* A damper having one adjustable blade.

8. *Face:* A grille with provision for attaching a damper.

9. *Register:* A face with a damper attached.

10. *Flange:* The portion (either integral or separate) of a grille, face, or register extending into the duct opening for the purpose of mounting.

11. *Frame:* The portion (either integral or separate) of a grille, face, or register extending around the duct opening for the purpose of mounting.

12. *Margin:* The margin of a grille, face, or register is one-half of the difference between the duct dimension and overall dimension measured either horizontally or vertically.

13. *Fret:* The member separating the openings of a grille, face, or register.

14. *Free Area:* The total minimum area of the openings in the grille, face, or register through which air can pass.

15. *Core Area:* The total plane area of the portion of a grille, face, or register bounded by a line tangent to the outer edges of the outer openings through which air can pass.

16. *Mean Area:* The total of the core and free areas divided by two.

17. *Duct Area:* The area of a cross-section of the duct based on the inside dimensions at the point where the grille, face or register is mounted.

18. *Percentage Free Area:* The ratio of the free area to the core area expressed in percentage.

19. *Dimensional Ratio:* The ratio of length of the core of a grille, face or register to the width.

20. *Throw:* The distance air will carry measured along the axis of an air stream from the supply opening to the position in the stream at which air motion reduces to 50 fpm.

21. *Envelope:* The outer boundary of an air stream.

GRILLE LOCATIONS

The location of supply and exhaust outlets is extremely important if a satisfactory installation is to be secured. Very frequently, however, the room or building is planned and constructed with practically no consideration of this problem. The engineer of today is more likely than not to have as his problem a building that was constructed long before any consideration whatever was given to air conditioning it. Consequently, the room shapes, the location of columns and beams, and other details of architecture frequently make it difficult to properly locate the outlets. In general, for a cooling installation, the grilles should be located high enough from the floor to prevent the discharge of air directly upon the occupants of the room, and far enough down from the ceiling to minimize

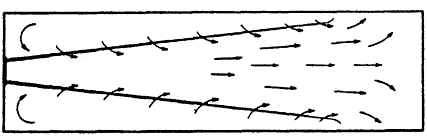

FIG. 1. PLAN VIEW LONG THROW SUPPLY OUTLET

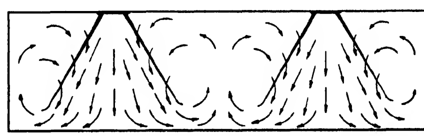

FIG. 2. PLAN VIEW SHORT THROW SUPPLY OUTLETS

the possibility of streaking, and to permit induction of air from all sides of the stream. If the stream actually strikes the ceiling, but at a small angle, the throw will be increased somewhat if the ceiling is smooth. If the angle at which the stream hits the ceiling is 20 deg or more, or if the flow along the ceiling is obstructed by panel mouldings or beams, air velocity may be rapidly lost and a decreased throw result. The air stream also should be so directed that it will not strike nearby columns or beams in such a way as to cause misdirection of the air stream or drafts. Where the room is of irregular shape, as an ell, or where it has an alcove in one side, consideration should be given to obtaining satisfactory circulation in these corners. Frequently this cannot be done except by the use of multiple supply outlets. In using multiple outlets, care must be taken that the several air streams do not interfere with each other, until their velocities have been reduced to values which will not cause high turbulence and a drafty condition. Beams and offsets in the ceiling will cause little difficulty when substantially parallel to the direction of flow, unless they are of considerable depth, but when positioned across the air stream, may cause drafts and failure to secure satisfactory circulation in that portion of the room farthest from the outlet. In the case of a heating installation, down-drafts produced by such obstructions may not be serious, because the air will rapidly lose its downward motion, but the possibility of failure to obtain satisfactory circulation still exists.

The location of supply outlets should, if possible, be such as to take advantage of the maximum velocity permissible from a noise standpoint. For instance, the spaces illustrated in Figs. 1 and 2 may be satisfactorily served by either arrangement. However, by taking advantage of the long throw, to which the arrangement in Fig. 1 lends itself, fewer outlets are required and additional savings are effected in the sheet metal work.

In solving the problem of properly conditioning a room of irregular shape, where multiple wall supply grilles are objectionable, a ceiling outlet of the type illustrated in Figs. 3 and 4 may very often be the best solution.

In choosing the most desirable location for the return air grille, consideration should be given to its effect on circulation of the air through the room. It is generally true that the return air grille should be placed on the same wall as the supply and near the floor level. This results in a U-shaped air path (Fig. 5) which covers the room thoroughly. The arrangement shown in Fig. 6 should be avoided, because it tends to create

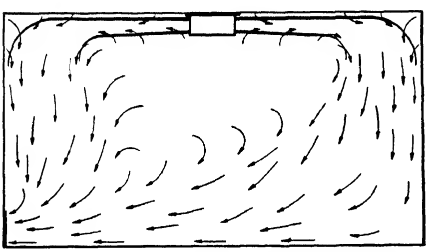

FIG. 3 ELEVATION VIEW CEILING SUPPLY OUTLET WITH RETURN WALL OUTLET

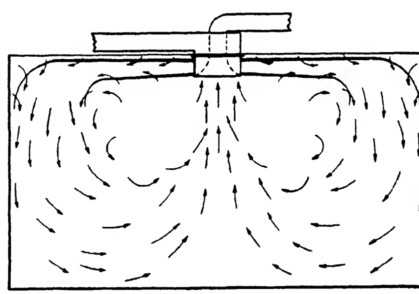

FIG. 4. ELEVATION VIEW CEILING SUPPLY AND RETURN OUTLET

a stagnant section below the supply grille. What would otherwise be an unsatisfactory dead spot in a room may in some instances be taken care of by location of the return air grille near that area (Fig. 7).

STANDARDS FOR SATISFACTORY CONDITIONS

The most satisfactory air condition cannot be definitely stated for any particular individual without conducting a series of tests with that individual as subject; some persons are less sensitive than others to variations in temperature, humidity, air velocity and noise. The best that can be done is to attempt to set limiting conditions leaning toward the values of these variables which produce a condition of comfort for the greatest number of individuals. On a cooling installation, the allowable deviation from average room temperature, that is, the temperature of puffs of air which may strike a person momentarily, is a function of the room temperature as well as the velocity of the air. For instance, in a room controlled at 72 F, a puff of air at 70 F might be uncomfortable to an individual, even at relatively low velocities, whereas if the average room temperature were 80 F, air at 78 F, even at moderate velocities, might be very satisfactory. However, air at 78 F in an average room

temperature of 83 F would be cold. In general, other conditions being equal, for the range of temperatures normally encountered in living quarters on cooling installations, the permissible deviation from average room temperature varies from approximately 1 F at the low end of the range to about 3 F at the high end of the range. In this matter, it is important to consider the particular problem in the light of the type of occupancy. For instance, greater deviations from room temperature and higher velocities may be permitted in a garage or a hotel hallway than would be permissible in an office or living room. The velocity which may be considered the permissible maximum differs with the temperature deviation for a given installation, but an absolute maximum under any conditions might be considered that which would produce a mechanical

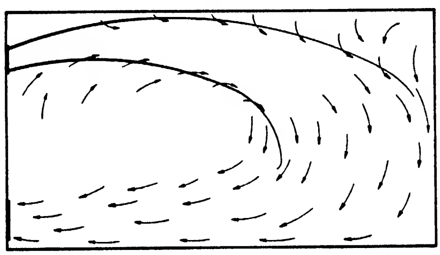

FIG. 5. ELEVATION VIEW CORRECTLY LOCATED RETURN OUTLET

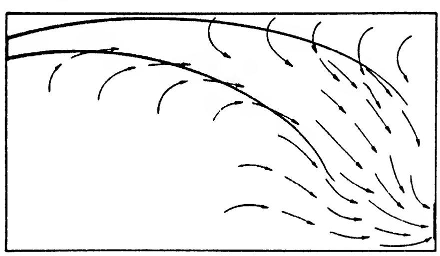

FIG. 6. ELEVATION VIEW OF IMPROPERLY LOCATED RETURN OUTLET

disturbance, such as the movement of a person's hair or disturbance of papers on a desk. Humidity is an important consideration in the determination of one's feeling of comfort; however, if the room generally is assumed to be at a satisfactory value of relative humidity, the designer is justified in neglecting this factor when considering permissible fluctuations in temperature and velocity in the occupancy zone. This is true because the maximum allowable temperature fluctuation results in an unnoticeable humidity change.

The standards that might be set up for maximum allowable room temperature deviation and air velocity would not be the same for both heating and cooling installations. In the former case, any appreciable temperature deviation is likely to be above rather than below the average room temperature, whereas the reverse is most likely to be true on a cooling installation. Further, because air movement has a cooling effect in itself, the feeling of warmth due to temperatures above room tem-

perature is counteracted to a certain extent so that an individual may be subjected to higher velocities of warm air without the feeling of discomfort occasioned by the same velocities of cool air. In every case, it should be the purpose of the designing engineer to keep the conditions within the zone of occupancy as nearly uniform as possible, securing minimum temperature deviations and low velocities. The air velocity at all points in the room should be at least 25 fpm for good results.

It is impractical to measure momentary temperature differences with any degree of accuracy in the field, but in checking a given installation it will generally be found satisfactory to measure velocity only, since on cooling installations high velocities normally occur with low temperatures, and on heating installations high velocities occur with high temperatures. That is, in the former case, the chilled supply air loses its velocity and undergoes an increase in temperature as it settles into the occupancy zone, whereas in the latter case the heated supply air loses its velocity and undergoes a decrease in temperature during this process. Therefore, if the

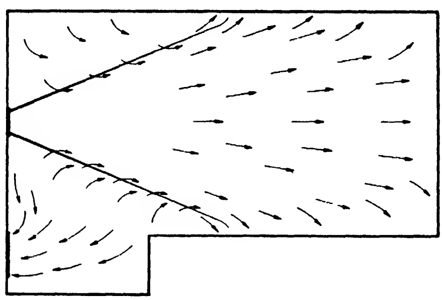

FIG 7. PLAN VIEW CORRECTLY LOCATED RETURN OUTLET ELIMINATING STAGNANT SPACE

average velocities within the occupancy zone are not excessive, one is fairly safe in assuming that the temperature difference is also within permissible limits.

The subject of sound control is covered in Chapter 30 and it is recommended for detailed review before consideration of the problem of air outlet noise. An understanding of the relation between sound intensity and loudness level in decibels, as well as the effect of the presence of sound absorbent materials in the room, is particularly necessary. A more detailed discussion of the nature of this problem appears later; whereas the following comments refer to what constitutes a satisfactory noise condition.

Obviously, the nature of the conditioned space is important when considering the allowable outlet noise. In factories, press rooms, and similar spaces where the noise level is 65 db or higher, no complaints of grille noise are likely to be made. On the other hand, some homes, offices, hospitals, and, most of all, radio broadcasting and movie sound studios present a real problem which must be intelligently attacked if a satisfactory installation is to be made. In this chapter the noise of the air outlets (and returns) only is considered, it being assumed that the noise or sound level of the room without the outlet noise includes that which may

be contributed by fans, motors, duct work, and other items of conditioning equipment. The control of noise from these sources is another problem (see Chapter 30). Where sound control is important, the actual room sound level without conditioning equipment should be known. If feasible, the contribution of the conditioning equipment, less outlets, should be estimated to secure the working sound level. If this correction is not made, the use of the first value errs in the direction of safety.

It is evident that the point within the room which should concern the designer in this problem is that at which the outlet noise is greatest. A tentative standard *listening point* relative to the outlet is suggested later in this discussion, and it is assumed that the outlet noise data are taken with reference to this point. If it is desired that the outlet noise result in an inaudible addition to the existing noise level, it is safe to assume the total outlet noise to be 5 db below room level. This results in an increase in total noise of slightly over 1 db, which is unnoticeable. If an increase of 3 db is permissible, the outlet noise level may be equal to the room noise level alone. All outlets in the room must be considered, as will appear later, and the returns may be ignored only if they are so sized that the velocity of air through them is much less than through the outlet.

DISTRIBUTION FACTORS IN ROOM COOLING

In attempting to design a satisfactory air distributing system, it is first necessary to properly locate the grilles in accordance with the recommendations already stated. Assuming that the best locations have been selected, it then becomes necessary to choose the proper grille for that location. The considerations involved are the amount of air to be handled, the velocity permissible from the standpoint of noise, and the distance the air should carry. The distance it will carry, assuming no obstructions, is affected by a number of factors which are listed below:

1. The temperature difference between incoming and room air.
2. Height of grille above floor.
3. Face velocity.
4. Core area.
5. Core aspect ratio.
6. Design of grille.

The manner in which the above factors affect throw may be generally stated. All other things being constant, a lower temperature of incoming air will result in shorter throw; a greater height above the floor will affect a longer throw; a higher velocity will produce a longer throw; greater area will give longer throw; larger aspect ratio will decrease throw. The variation in throw with type of outlet will, of course, depend upon the design characteristics of the outlet.

In consideration of what constitutes the possible throw of an outlet under a given set of conditions, it is important to remember that the throw may be unsatisfactory for any one of several reasons:

1. It may be so long that it will strike the far side of the room and come down the wall with velocities higher than are permissible,

2. It may be so short that it will fail to carry the full length of the room, and short-circuit to the return air outlet, or

3. It may spill into the center of the room.

In the first case, the system fails for lack of uniform distribution and the presence of cold areas. In the second case, the standards as to velocity and temperature difference in the zone of occupancy may be satisfactorily met, but air distribution and circulation throughout the entire room is not accomplished, with the result that the end of the room away from the outlet would not be satisfactorily conditioned. In the third case, the shortcomings of both case one and case two are present. It is evident, therefore, that for a given outlet discharging air at a given velocity, there is a maximum and a minimum length of room which can be satisfactorily handled. In the latter, the velocity of the air down the far wall is just within the maximum permissible, while in the former, satisfactory circulation is barely accomplished.

In general, the higher the outlet is above the floor, the greater may be the difference between room air and incoming air temperatures.

Assuming that proper supply outlets for a given installation have been selected, unsatisfactory performance may still result due to the construction of the duct work immediately back of the outlets. Performance data on the grilles and registers of various manufacturers should be based upon results obtained with the air approaching the grille perpendicularly and at uniform velocity over the entire duct cross-section. Where this condition does not exist in practice, performance predictions based on published data cannot be expected to be realized. Every precaution should be taken to secure as nearly ideal conditions in the approaching air stream as are possible.

In addition to disturbances due to the construction of the duct work itself are those which may be created by dampers immediately behind the grille. Where either multiple louvre or single blade dampers are used, considerable deflection of the air stream may result, if it is throttled appreciably by these means. This is particularly true when the fins of the register core are perpendicular to the damper blades. If the core has sufficient depth and the fins are parallel to the blades, there is a marked tendency to straighten the air stream, although some deflection may still result.

Any attempt to secure a low face velocity and high duct velocity by the construction of any expanding chamber immediately behind the grille is very likely to be unsuccessful. In order to expand from a small duct to a larger one, and have the air stream fill the duct at the end of the diverging section without turbulence, angle A in Fig. 8 should be about 3 deg for four-sided expansion and about 5 deg for two-sided expansion. From this it is apparent that an attempt to secure equivalent results with a short connection would be futile. What actually happens when this is attempted is illustrated by the arrows in Fig. 8. When localized high velocities through the outlet exist from this cause or any other, the noise produced will naturally exceed that which the outlet area and average face velocity would lead one to expect. This fact should be remembered in considering the use of register dampers, particularly in those cases where there must be considerable throttling with the damper to balance a poorly designed system. Where reduction of noise is important, it is recommended that balancing dampers be placed in the duct *ahead* of the acoustic duct lining.

Similar unequal face velocities, aggravated by a deflection of the air

stream, are obtained with the arrangement shown in Fig. 9. The latter may be corrected by inserting a turning member in the elbow back of the outlet face as shown in Fig. 10. The importance of straightening the air stream and affecting uniform distribution over the entire face of the outlet cannot be over-emphasized

DISTRIBUTION FACTORS IN ROOM HEATING

The problem in the case of a heating installation is substantially the same as in cooling, with a few exceptions. Because the temperature of the incoming air is above that of the room, there is no tendency for it to drop and consequently the throw is not particularly affected by temperature difference in a low ceiling room. In general, the air should be deflected downward where the grille is above the occupancy zone, and this is particularly desirable where the ceiling is high. For the same reason, that is, to keep the heat in the occupancy zone and to avoid excessive

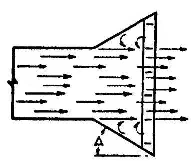

FIG. 8. EFFECTS OF EXPANDING DUCT

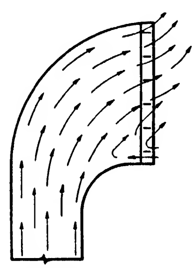

FIG. 9. UNEQUAL FACE VELOCITIES

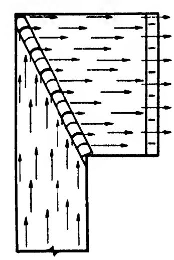

FIG. 10. EFFECT OF TURNING MEMBER

temperature at the ceiling, it is desirable to have the grille comparatively low on the wall, and just slightly above the occupancy zone. If the grille is lower than this, it may create an unsatisfactory condition of very warm air at quite high velocities where it can possibly strike the occupants of the room. Where the velocities are very low, the grilles may even be satisfactorily located below the 6 ft level, although the immediate vicinity of the supply outlets will probably be useless for occupancy because of high temperature. Essentially, the problem is to keep the incoming air up for cooling, and down for heating, until it is thoroughly mixed with the room air. Grilles and registers which are adjustable for deflection upward and downward, either by moving the fins or inverting the grille, are in general use.

AIR OUTLET NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at a constant rate. Due to partial reflection at the boundaries of the enclosure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the outlet) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated and inversely

proportional to the number of sound absorption units (sabins) in the room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db. However, it is not satisfactory to consider the grille noise on this basis (wherein the sound power received directly from the source is small compared with that received by reflection) since in practice the occupants of the room may be quite close to the grille. The nearer the listener is to the sound source, the greater the proportion of the sound intensity which is due to direct transmission.

In the absence of generally accepted standards at this time it is suggested that the loudness level 5 ft from the lower edge of the outlet, measured downward at 45 deg in a plane perpendicular to the outlet at its center, represents about the maximum within the zone of occupancy. The cases where persons are nearer to the outlet than this are rare and are ignored in the consideration of this problem. Although the effect of sound absorbent material on the intensity at the 5 ft station is not nearly so great as at more remote points in the room, it should not be ignored without consideration of the error involved. An average living room may contain 100 sabins (absorption units). If this be decreased to 50 sabins, the *diffuse* or reflected sound level would be increased 3 db. However, at the 5 ft station the increase would be less than 2 db. If the absorption of the room be increased to 200 sabins, one might expect a reduction in diffuse noise of 3 db; but at the 5 ft station the reduction would be less than 1½ db. Furthermore, even though the absorption be increased without limit (as in free space) the reduction would still be less than 2 db because of proximity to the source.

In comparing sound ratings of various grilles, the following must be known if the information is to be intelligently applied:

1. The threshold intensity on which the decibel ratings are based.
2. The distance from the grille at which data were taken.
3. If stated as loudness level versus velocity for a given grille, the *core area* (not nominal area) must be known.
4. The sound absorbing characteristics of the test room.
5. Whether or not corrected for test room loudness level; if not, the room level (without grille noise) must be known.
6. Methods used for recording data.

Data mentioned in this chapter are assumed to have been referred to the following:

1. Threshold intensity = 10^{-16} watts per square centimeter[1].
2. Microphone location 5 ft from lower edge of outlet on a line downward at 45 deg and in a plane bisecting the outlet perpendicularly.
3. Where data are given as loudness level versus velocity, the rating is per square foot of core area.
4. The room is assumed to have 100 sabins absorption.
5. Plotted data are loudness levels of *outlets only*, correction having been made for test room level.
6. Data taken with a direct reading sound-level meter with frequency weighing network intended to approximate the response of the human ear.

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[1]American Tentative Standards for Noise Measurement, *American Standards Association*.

If the published ratings are in terms of decibels per square foot, correction must be made for area to secure to total sound level of outlets of more or less than one square foot area. This can be done by use of the following formula:

$$\text{Decibel Addition} = 10 \log_{10} A \quad (1)$$

where:

A = core area, square feet.

In practice the allowable total sound and the required air flow are usually known, and it is desired to determine the maximum allowable velocity. Since total loudness and air flow are both functions of velocity and area, the solution of the problem by use of the previous analysis implies a trial and error method. It has been found possible to present

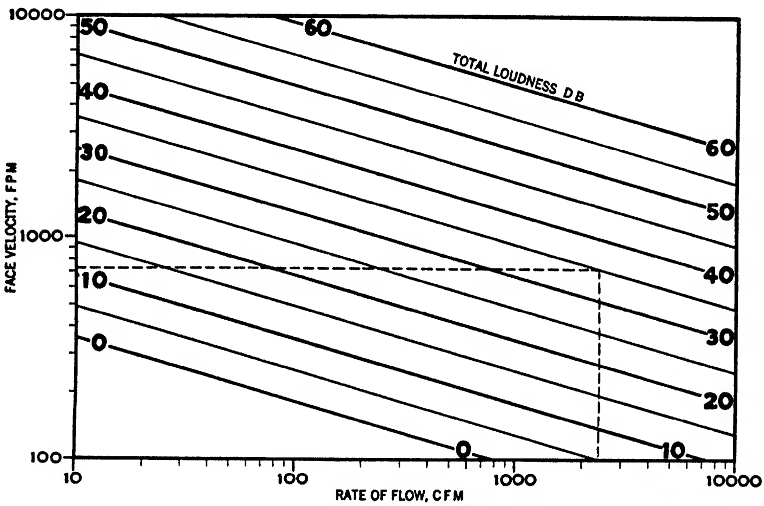

FIG. 11. AIR FLOW AND LOUDNESS CHART

these data with sufficient practical accuracy as a family of uniform curves as illustrated in Fig. 11. With this chart it is possible to find directly the velocity in feet per minute which will give a predetermined total loudness at a predetermined rate of flow expressed in cubic feet per minute. The values used are arbitrarily chosen for the purpose of discussion and do not necessarily represent data referring to any particular make of grille, register or air outlet. It is assumed that Fig. 11 is based on a room having 100 sabins of sound absorption. In such a room the sound level due to other sources may be 35 db. As previously stated an outlet having a noise level of 30 db would be substantially inaudible in such a room.

If 1000 cfm are required with a total noise due to outlet of 30 db, a velocity (Fig. 11) of about 675 fpm may be used. From this velocity and the rate of flow, the core area can be computed. This determination was on the basis of a room absorption of 100 sabins. If the absorption is greater, the 675 fpm velocity is safe, since the loudness level will go down. However, correction can be made if desired by the use of the chart of

Fig. 12. Thus, if the absorption is 200 sabins, a correction of +1.3 db may be made and the permissible velocity becomes that corresponding to a total loudness level of 31.3 decibels or approximately 750 fpm. If the room is highly reflecting and has an absorption of less than 100, correction is much more important. For instance, for 35 sabins a correction of −3 db

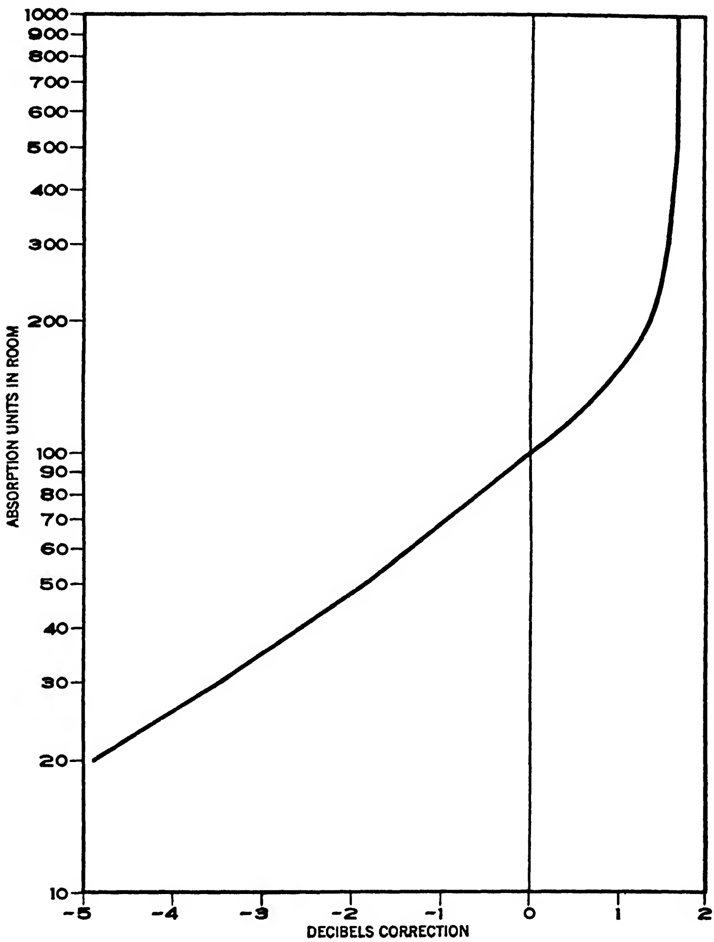

FIG. 12. ROOM ABSORPTION CORRECTION CHART

must be made and the maximum velocity corresponding to 27 db total loudness chosen; that is, approximately 550 fpm.

Where more than one outlet must be considered, the problem is more complicated. If a similar outlet is added in a far corner of a highly absorbent room, the change in noise level at the 5 ft station at the first outlet is small; however, if the room is small, or highly reverberant or both, the intensity at the 5 ft station may be almost doubled and the noise level increased nearly 3 db thereby. The simplest method of hand-

ling this problem, and one which errs in the direction of safety, is to treat the room as though all the air were being supplied by one outlet. Thus, if two outlets, each supplying 1000 cfm are used, the value 2000 cfm should be used with Fig. 11. Although this method may place an unwarranted limit on velocity when used in a large room, it is seldom that such a room has a noise level low enough to make this penalty serious or to justify a more complicated though more exact procedure.

In general, return grilles are selected for velocities about half the supply velocity, and when this is done, they may be neglected in sound computations. However, if supply and return grilles are the same size, resulting in the same face velocity, they must be treated as two supply outlets. That is, if 1000 cfm is supplied and exhausted through grilles of the same area, 2000 cfm must be used in the solution with Fig. 11.

SELECTION OF SUPPLY OUTLETS

After the heating and cooling load calculations have been made (Chapters 7, and 8), and a suitable supply air temperature selected, the

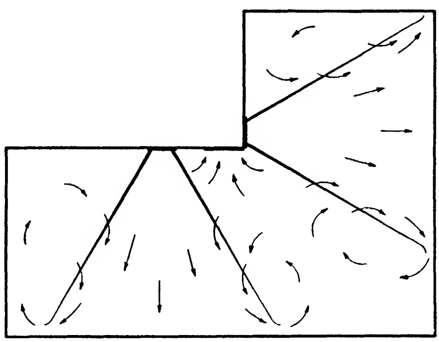

FIG. 13. PLAN VIEW TYPICAL GENERAL OFFICE

volume of air required for each space can be determined. The next step is to determine the velocity at which the air may be introduced into the space quietly and without creating objectionable drafts.

Present-day grille design coupled with the introduction of effective acoustical treatment for minimizing fan and duct noises have made grille face velocities in excess of 1500 fpm feasible, and 600 to 1200 fpm is now used in practice. This range of velocities is approximately three times higher than common practice values of a few years ago.

Since high velocities make for smaller ducts and outlets, and therefore savings in space as well as greater flexibility in locating the duct work to the best advantage, selection of design velocities is a very important step.

The selection of proper velocity requires that the designer have before him reliable data applicable to the particular make of outlet he proposes to use. Even under these circumstances, the problem is one of *cut and try* because permissible velocity may be determined by either noise or throw.

A method for selecting supply outlets is outlined below in the form of a

sample cooling problem, using numerical values which have no reference to any particular make of outlet.

1. The load calculations have been made; a suitable temperature differential has been selected (it is to be understood that the data referred to from this point on are based on this temperature differential), and the volume of air required determined. Assume that Fig. 13 represents a small general office having a noise level of 40 db and that 2500 cfm must be supplied for proper conditioning.

2. Select a tentative location for the outlet or outlets, having in mind the type of grille most likely to effect proper distribution. In this particular case, two outlets having a wide spread appears to be a logical choice.

3. Data from which to determine velocity which corresponds to 2500 cfm and a noise rating at least 5 db below the noise level of the office may be presented in a number of forms, one of which is shown in Fig. 11. (Fig. 11 represents assumed values only. In practice similar data should be obtained from the manufacturer whose outlets are being considered. Several similar charts or tables may be necessary to cover any one manufacturer's complete line.) From Fig. 11 it will be noted that for 2500 cfm the type of grille selected may be used at velocities up to 700 fpm without exceeding 35 db, that is, 5 db below the noise level of office

4 Having determined the velocity, the core area becomes fixed at 3 57 sq ft or 257 sq in. per outlet In this problem, the two grilles in question are so close together that consideration of their combined area in determining the permissible velocity from the standpoint of noise introduces little error.

5 The type grille selected has thus far been found satisfactory from a noise standpoint, provided the face velocity does not exceed 700 fpm. The next consideration is throw, which may be assumed to be 16 ft, and by reference to a manufacturer's catalogue the proper correlative test data may be checked with the throw assumed. It is of course evident that one or more types of grilles may satisfy the requirements, and that in any one type there will be a choice of outlet proportions It will also be evident that the tentative selection of an outlet having a wide spread may be unsatisfactory from the standpoint of throw, in which event a second choice should be made and the procedure repeated.

In the case of a heating problem, the method of solution is the same, but the manufacturer's data must, of course, be based on tests with air above room temperature.

TYPES OF SUPPLY OUTLETS

Grille, register or outlet design for attaining uniform distribution and minimum air resistance consists of various fixed and adjustable arrangements. Some types are designed with directing air blades, fins, bars, louvres, or thin metal strips shaped into a series of grooves or tubes, all of which may be set into a suitable round, square or rectangular frame. In order to attain desired long or short air throws, the emergence of air from the outlet may be directed to straight, deflecting, converging or jet air streams depending upon the outlet design. Designs which direct the air stream to produce an ejector effect within the enclosed space tend to mix the room air with the conditioned air to provide uniform distribution.

Centrally located ceiling or wall type outlets arranged for completely diffusing the air consist of several round, hollow, cone-shaped flaring members placed in the proper relationship to each other. The velocity of emergence of the air from the unit can be made practically uniform over the entire surface of the outlet, and the velocity in any direction may be varied to any desired value by adjusting the position of the cones. One or more of the smaller flaring members act as ejectors and injectors which

draw a small proportion of the room air into the air spreader where it mixes with the conditioned air before it is discharged.

An idea for producing even distribution of air consists of a perforated ceiling made of a suitable architectural surface and installed a small distance below the normal ceiling level of the room. In the space provided by this suspended ceiling a plenum chamber is formed into which the conditioned air is introduced. From the plenum space the air is permitted to diffuse through the large number of small ceiling openings into the room.

Railroad Cars

The early practice of air conditioning railroad passenger cars consisted of a system of bulkhead distribution for the conditioned air. The air was discharged through an inlet opening at each end of the car toward the middle with the flow parallel to the long dimension of the car. This type of installation resulted in drafts in the middle of the car and was considered unsatisfactory except for small sections that did not require large quantities of air. Later designs incorporated a duct delivery system on each side of the car roof directing the air through numerous inlet openings toward the middle of the car where the two air streams come in contact and deflect downward, gradually filtering into the aisle. At the present time, several center duct air distribution systems are used in railroad car applications. In some instances, square or circular ceiling outlets connected to a center duct have been used, which distribute the air along the ceiling in widening circles and at right angles to the inlet opening. Another method consists of a continuous slot in the bottom of the duct to which is attached a flat plate so that the air is deflected along the car ceiling. There are also installations in which the ceiling of the car is constructed of perforated metal through which the conditioned air flows through thousands of openings from a plenum chamber. Extensive tests of all methods of air distribution indicate that desirable results are obtained from an inside center duct with a large number of openings.

Sleeping cars present a special problem in air distribution on account of the berth curtains. In some cars, each lower berth is equipped with an individual fan to draw in cooled air from the aisle. In other cars, small individual ducts with adjustable air outlets deliver air from the central supply system to each lower berth. Upper berths require no special arrangement.

BALANCING SYSTEM

In designing an air conditioning system, it should be the aim of the engineer to so proportion the duct system that proper distribution of air to every outlet will be obtained. Since this is almost impossible to accomplish in practice, it becomes necessary to have means of balancing the system to secure the desired amount of air in each space. There are a number of ways in which this may be accomplished, some of which are listed:

1. Dampers on the supply grilles.
2. Dampers on the return grilles.
3. Dampers in the supply ducts.
4. Dampers in the return ducts.

5. Reducing the effective area of some outlets by blank-offs.
6. Combinations of dampers in both supply and return air.

Dampers on the supply grilles themselves are objectionable because of their effect on the air stream. Dampers on the return grilles are frequently helpful in building up a static pressure in the room to prevent infiltration of outside air, and at the same time reduce the volume of incoming air. However, it is frequently impossible to sufficiently reduce the incoming air by this method alone. A damper in the supply duct some distance back of the outlet forms a very satisfactory means of regulating the flow without disturbing distribution across the outlet face. A damper in the return air duct has the advantage over one immediately behind the grille in that it does not tend to create high localized velocities through the grille as the latter might do if nearly closed. Blank-offs consisting of pieces of sheet metal covering a portion of the outlet face can frequently be used satisfactorily, although determination of just what is required is a matter of experiment, and the balancing of the system is not nearly so conveniently accomplished as with dampers. Dampers in both supply and return air form the most flexible means of controlling the supply to the room and the static pressure within the room. When feasible, these dampers, particularly those in the supply ducts, should be a substantial distance from the outlet, and ahead of the acoustic duct lining if used. Due consideration should also be given to the use of the several volume control and uniform distribution devices now available. See *Catalog Data Section*.

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PROBLEMS IN PRACTICE

1 ● What important factors are involved in the correct distribution of air to an enclosed space?

Not only is it important to distribute the air from the fan to the various spaces served by the system, but also the air must be properly distributed within the enclosed space to give complete satisfaction.

2 ● Upon what basis should the selection of supply outlets be based?

If possible, the selection should take advantage of the maximum velocity permissible from a noise standpoint.

3 ● What factors in grille design affect the length of air throw?

a. The temperature difference between incoming and room air, *b* height of grille above floor, *c.* face velocity, *d.* core area, *e.* core aspect ratio, and *f.* design of grille.

4 ● How does the height of a supply outlet affect the temperature differential within a room?

In general, the higher the outlet is above the floor, the greater may be the difference between room air and incoming air temperatures.

5 ● Under conditions prevalent in a large room, how does the intensity of sound develop at an air outlet vary?

The intensity of sound energy is substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units in the room.

6 ● What are the essential differences between a high velocity long throw and short throw grille?

Generally, a high velocity long throw grille is used where a large compact mass of air is projected with a reduction in the periphery of the air stream whereas, with a short throw grille design the periphery of the air stream is expanded as much as possible to increase the scrubbing action between the incoming air stream and the stationary air.

7 ● What type of system is generally used in a large continuously operated theatre?

Most large continuously operated theatres are provided with a complete downward system of air distribution. With this system a large number of outlet openings are provided each of which discharges air in a thin horizontal stream at high velocity in order that the cool air would be mixed with the area in the theatre before it reaches the patrons. In this type of system the best distribution is obtained when a sufficient number of exhaust openings are located under the seats.

8 ● What means are available for balancing a system to secure the desired amount of air in each space?

Ways in which this may be accomplished are by *a* dampers on supply and return grilles, *b.* dampers in supply and return ducts, *c.* reduction of the effective area of some outlets by blank-offs, and *d.* combination of dampers in both supply and return air duct systems

Chapter 29

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Friction Loss Chart, Proportioning the Losses, Sizes of Ducts, General Rules, Procedure for Duct Design, Air Velocities, Proportioning the Size for Friction, Main Trunk Ducts, Equal Friction Method, Duct Construction Details

THE flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated, and the error occasioned by the assumption that the gas density remains constant throughout the flow may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice.

In the development of the formulae, diagrams, and tables for the flow of air, use is made of the following basic equation for the flow of fluids:

If H_V be the velocity head in feet of a fluid, and the velocity, V, be expressed in feet per minute, the fundamental equation is

$$V = 60\sqrt{2g\,H_V}$$

The factor g is the acceleration due to gravity, or 32 16 ft per second per second.

It is usual to express the head in inches of water for ventilating work and, since the heads are inversely proportional to the densities of the fluids,

$$\frac{H_V}{\frac{h_V}{12}} = \frac{62\ 4}{d}$$

or

$$H_V = 5.2\,\frac{h_V}{d}$$

therefore,

$$V = 1096.5\sqrt{\frac{h_V}{d}} \qquad (1)$$

where

V = velocity in feet per minute.
h_V = velocity head or pressure in inches of water.
d = weight of air in pounds per cubic foot.

For standard air (70 F and 29 921 in. barometer) $d = 0.07492$ lb per cubic foot. Substituting this value in Equation 1:

$$V = 1096.5\sqrt{\frac{h_V}{0.07492}} = 4005\sqrt{h_V} \qquad (2)$$

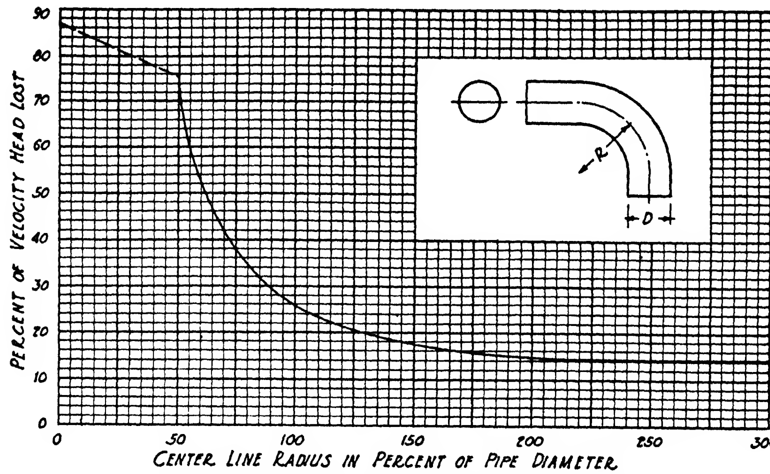


FIG. 1. CURVE SHOWING LOSS OF PRESSURE IN ROUND ELBOWS

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses are those due to the friction of the air against the sides of the duct. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Pressure Losses

Dynamic losses occur principally at the entrance to the piping, in the elbows, and wherever a change in velocity occurs. The entrance loss is the difference between the actual pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from 0.1 to

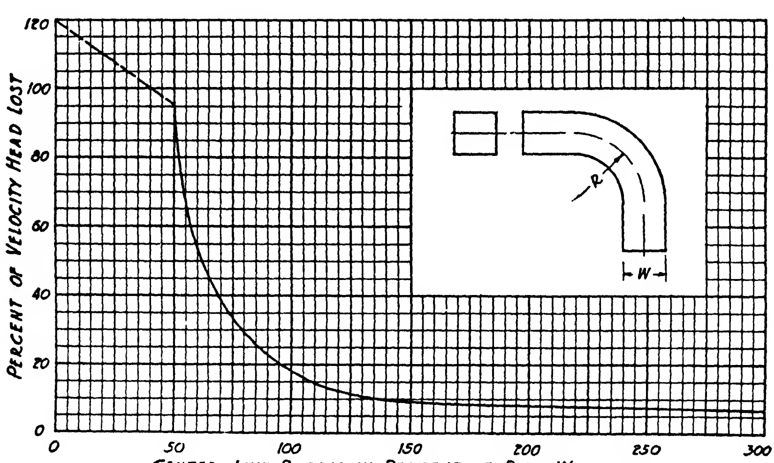


FIG. 2. CURVE SHOWING LOSS OF PRESSURE IN SQUARE ELBOWS

0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design. It is customary to express dynamic losses in terms of the percentage of the velocity head; in other words, the percentage of that pressure corresponding to the average velocity in the duct which is expressed in terms of inches of water gage. Figs. 1 and 2 show the effect of changing the radius of elbows of square and rectangular section[1]. These charts are based on tests of pipe elbows of ordinary good sheet metal construction. For example, a five-piece round pipe elbow having a centerline radius of one diameter has a loss of about 25 per cent of the velocity head. At a velocity of 2000 fpm the corresponding head is 0.25 in. water gage, and at this velocity the elbow just referred to would cause a pressure drop of 0.063 in. water gage. Experience has shown that good results may be obtained when the radius to the center of the elbow is 1½ times the pipe diameter. The pressure drop will then be approximately 17 per cent of the velocity head for round ducts, and 9 per cent for square ducts. Very little advantage is gained in making elbows with a radius of more than two diameters[2].

Friction Losses

Friction losses vary directly as the length of the duct, directly as the square of the velocity, and inversely as the diameter. Since length is a fixed quantity for any system, the factors subject to modification are the area and the velocity, which determine the relation between the first cost of the duct system and the cost of the power for overcoming friction.

The friction between the moving air and pipe surface causes a loss of head which is numerically equal to the pressure required to maintain a given velocity, and is expressed in the following modification of Fanning's formula:

For round pipe and standard air (70 F and 29 921 in. barometer)

$$h_L = f\frac{L}{D}h_v = \frac{L}{CD}\left(\frac{V}{4005}\right)^2 \tag{3}$$

For rectangular ducts

$$h_L = fL\left(\frac{a+b}{2ab}\right)h_v = \frac{L}{C}\left(\frac{a+b}{2ab}\right)\left(\frac{V}{4005}\right)^2 \tag{4}$$

where

h_L = loss of head, inches of water.

$h_v = \left(\frac{V}{4005}\right)^2$ = velocity head, inches of water.

V = velocity of air, feet per minute.

L = length of pipe, D = diameter of pipe, a, b = sides of rectangular duct — all in feet.

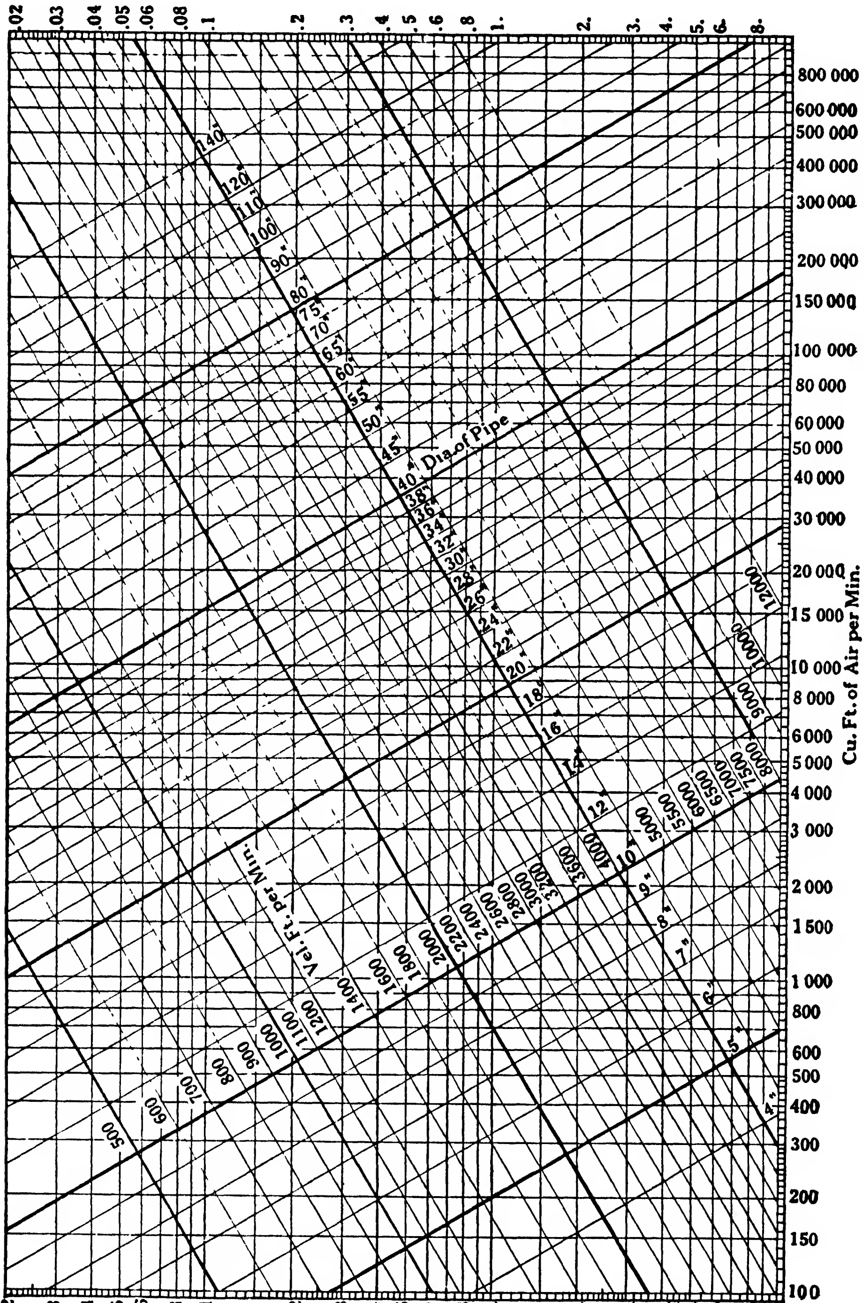
f = coefficient of friction.

$C = \frac{1}{f}$ = length of pipe in diameters for one head loss

For all practical purposes C varies only with the nature of the pipe surface: C = 60 for perfectly smooth pipe; = 55 for pipe as used in planning

[1]Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey (A S H V E TRANSACTIONS, Vol 19, 1913, p 366).

[2]Pressure Losses in Rectangular Elbows, by R D Madison and J R. Parker (*Heating, Piping and Air Conditioning*, July, p 365, August, p 427, September, p 483, 1936).



Friction in Inches of Water per 100 Ft.

FIG. 3. FRICTION OF AIR IN PIPES

mill exhaust systems; = 50 for heating and ventilating ducts; = 45 for smooth and 40 for rough conduits of tile, brick or concrete. However, Fritzsche states (and numerous tests check very closely) that f varies inversely as the 2/7 power of the pipe diameter, and inversely as the 1/7 power of the velocity, or inversely as the 1/7 power of capacity, which is the same thing. Thus Formula 3 may be revised as follows, based upon a loss of one velocity head (at 2000 fpm) in a length equal to 50 diameters of 24-in. galvanized swedged pipe:

$$h_L = 1.1 \frac{L}{CD^{9/7}} \left(\frac{V}{4005}\right)^{13/7} \tag{5}$$

The preceding formulae are based on standard air, and for other conditions the friction varies directly as the air density and inversely (approximately) as the absolute temperature. The increase of friction due to increase of air viscosity with increased temperature is small and is generally neglected.

Friction Loss Chart

Fig. 3 is a convenient chart for determining the friction loss for various air quantities in ducts of different sizes. The general form of this chart is familiar, but it should be noted that it is corrected for changes in the coefficient of friction based on the rule that the coefficient of friction varies inversely as the 2/7 power of the diameter, and inversely as the 1/7 power of the velocity. Fig. 3 is based on a loss of one velocity head (at a velocity of 2000 fpm) in a length equal to 50 diameters of 24-in. round galvanized-iron duct of the usual construction. Although this chart is laid out for a value of C equivalent to 50, it may be used for other values of C by varying the friction inversely as this constant. For example, if a rougher pipe is used with 40 as the value of C, the friction loss as read from the chart should be multiplied by $\frac{50}{40}$.

Example 1. Assume that it is desired to pass 10,000 cfm of air through 75 ft of 24-in. diameter pipe. Find 10,000 cfm on the right scale of Fig. 3 and move horizontally left to the diagonal line marked 24 in The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.59 in ; then for 75 ft the friction will be 0 75 × 0.59 = 0 44 in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Proportioning the Losses

Other losses of pressure occur at the entrance to the duct, through the heating units, and at the air washer. In ordinary practice in ventilation work it is usual to keep the sum of the duct losses ⅓ to ½ and the loss through the heating units at less than ½ of the static pressure. The remainder is then available for producing velocity. In the design of an ideal duct system, all factors should be taken into consideration and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

SIZES OF DUCTS

The sizes of ducts and flues for gravity or mechanical circulation of air are usually based on the losses due to friction, and these losses must be

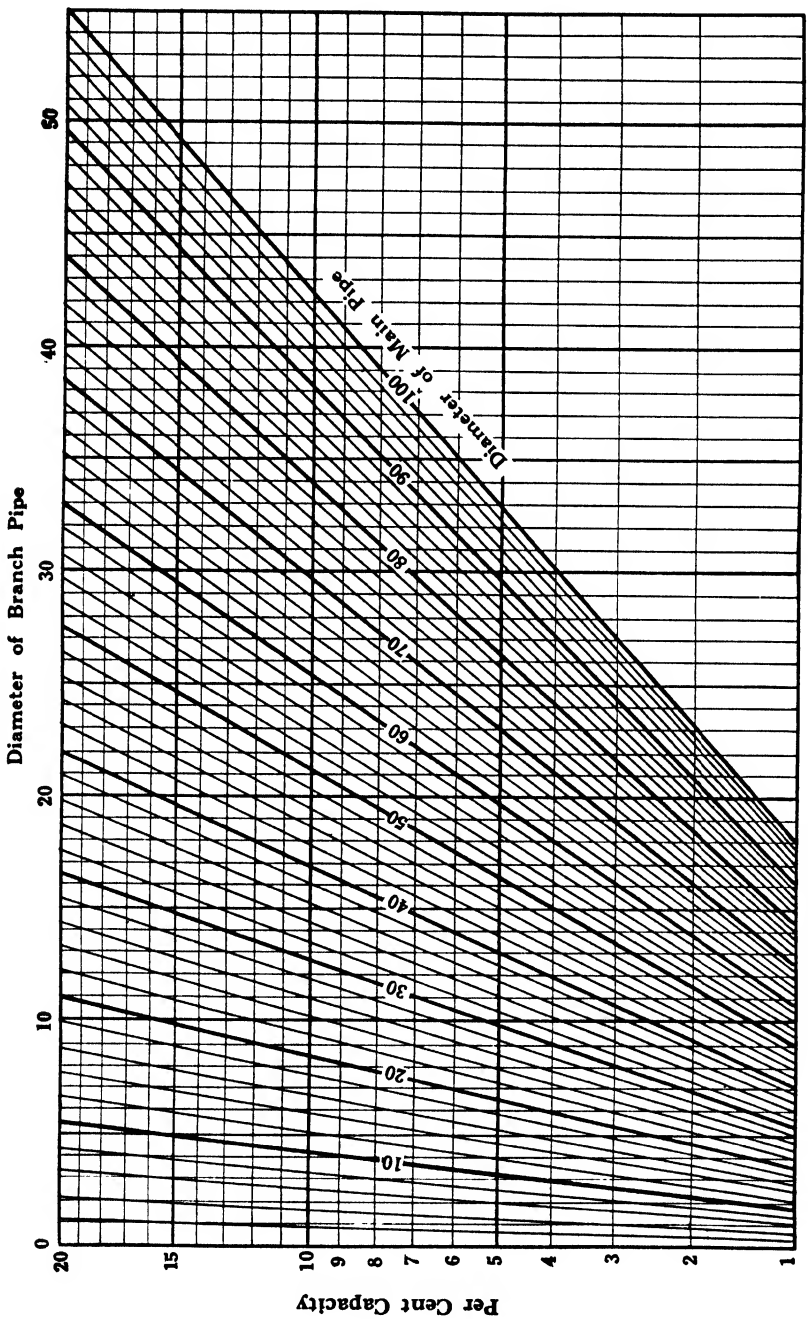


FIG. 4. MAIN AND BRANCH PIPES FOR EQUAL FRICTION PER FOOT OF LENGTH
(1 TO 20 PER CENT CAPACITY)

kept within the available pressure difference. This pressure difference in mechanical ventilation is that derived from the fan, while in gravity ventilation the aspirating effect due to the temperature and height of the column of heated air causes the pressure difference.

General Rules

The general rules to be followed in the design of a duct system are:

1. The air should be conveyed as directly as possible at reasonable velocities to obtain the results desired with greatest economy of power, material and space.
2. Sharp elbows and bends should be avoided.
3. The sides of all ducts or flues should be as nearly equal as possible. (In no case should the ratio between long and short sides be greater than 10 to 1.)

Procedure for Duct Design

The general procedure for designing a duct system is as follows:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.
2. Arrange the positions of duct outlets to insure the proper distribution of heat.
3. Divide the building into zones and proportion the volume of air necessary to supply the heat for each zone.
4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity.
5. Calculate the sizes of all main and branch ducts by either of the following two methods:

    a. Velocity Method. Arbitrarily fix the velocity in the various sections, reducing the velocity from the point of leaving the fan to the point of discharge to the room. In this case the pressure loss of each section of the duct is calculated separately and the total loss found by adding together the losses of the various sections

    b. Friction Pressure Loss Method Proportion the duct for equal friction pressure loss per foot of length

6. Calculate the friction for the duct offering the greatest resistance to the flow of air, which resistance represents the static pressure which must be maintained in the fan outlet or in the plenum space to insure distribution of air in the duct system. The duct having the greatest resistance will usually be that having the longest run, although not necessarily so.

Air Velocities

The following velocities of air are considered standard for public buildings:

1. Through the outside air intakes, 1000 fpm.
2. Through connections to and from heating unit, 1000 to 1200 fpm.
3. Through the main discharge duct, from 1200 to 1600 fpm.
4. In branch ducts, 600 to 1000, and in vertical flues, 400 to 800 fpm.
5. In registers or grilles, 200 to 400 fpm depending upon the size and location If diffusers of proper design are used, 25 per cent higher air velocities are permissible.

These duct velocities may safely be increased 20 per cent if first class construction is used to prevent any breathing, buckling, or vibration. High velocities at one point in the system neutralize the effect of proper design at all other points; hence the importance of splitters in elbows and similar precautions. For industrial buildings noise is seldom considered, and main duct velocities as high as 2800 or 3000 fpm may be used where conditions will permit. For department stores and similar buildings,

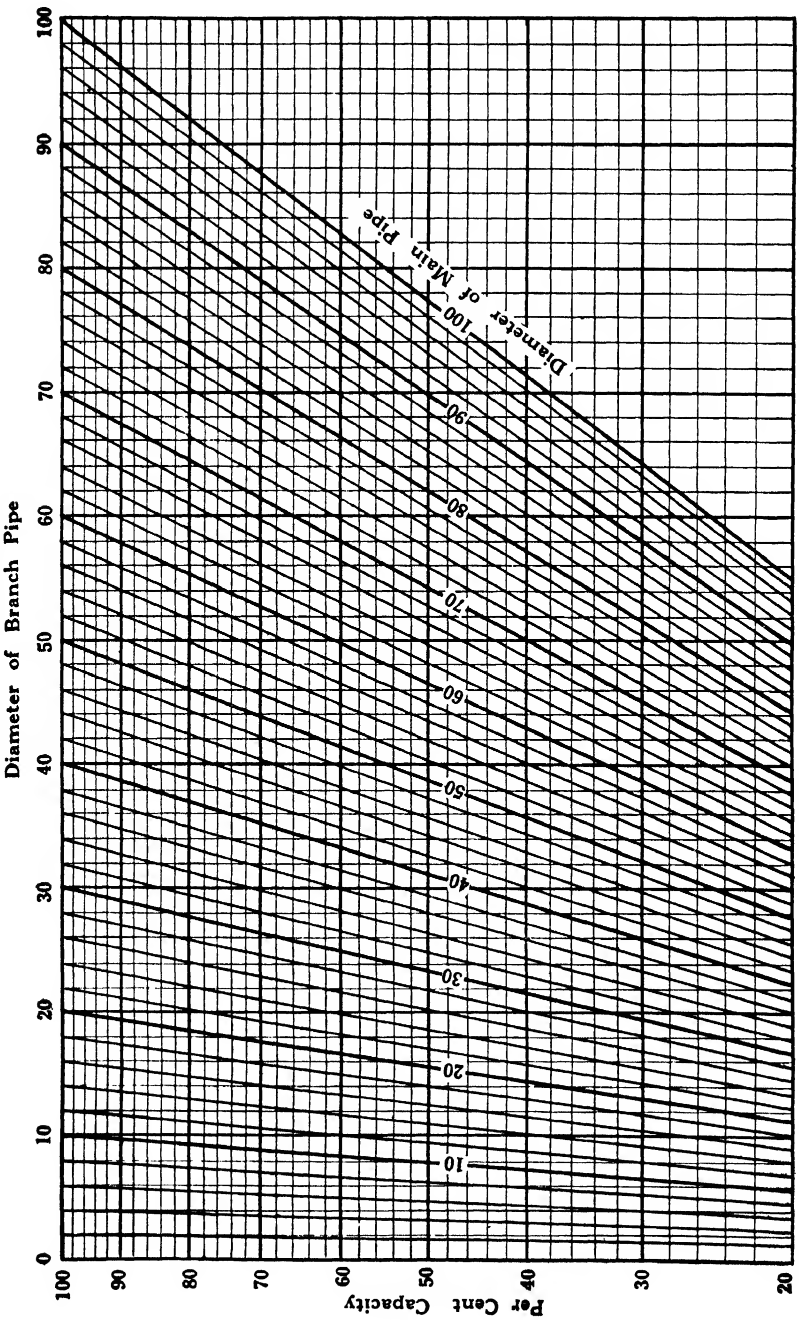


FIG. 5. MAIN AND BRANCH PIPES FOR EQUAL FRICTION PER FOOT OF LENGTH (20 TO 100 PER CENT CAPACITY)

maximum velocities with good construction and design may be as high as 2000 or 2200 fpm in main ducts, with suitable reduction in branches and outlets. With these velocities first-class duct construction is essential.

Proportioning the Size for Friction

By means of Figs. 4 and 5 the diameter of branch pipes necessary to carry a given percentage of the total air in the main pipe and to maintain equal friction per foot of the length through the entire system may be determined. These charts, as well as Fig. 3, are based on the assumption that the coefficient of friction varies inversely as the 1/7 power of the capacity.

Example 2. Suppose a 60-in. main pipe is to be used, and it is desired to know the size of branch pipe required to carry 50 per cent of the total air in the main. Find 50 per cent at the left of the chart, move right to the 60-in. diagonal line and note directly above at the top of the chart that the branch pipe will be 46.5 in. in diameter.

Where rectangular ducts are used it is frequently desirable to know the equivalent diameter of round pipe to carry the same capacity and have the same friction per foot of length. Table 1 gives directly the circular equivalents of rectangular ducts for equal friction and capacity, which are based on values determined from Formula 6:

$$d = 1.265 \sqrt[5]{\frac{(a\,b)^3}{a + b}} \tag{6}$$

where

a = one side of rectangular pipe, feet or inches

b = other side of rectangular pipe, feet or inches.

d = equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity, feet or inches.

To obtain the size of rectangular ducts for different capacities, but of the same friction per foot of length, first obtain the equivalent round pipe for equal friction. Thus, if a branch of sufficient size to carry 30 per cent of a 12 x 36-in. pipe is desired, it is found from Table 1 that the main is equivalent to a 22.2-in. diameter round pipe. From Fig. 5, 30 per cent of this is a pipe 14.3 in. in diameter, and referring again to Table 1, the rectangular equivalent branch is a 12 x 14 in., 10 x 17¼ in., or any other desirable combination.

Multiplying or dividing the length of each side of a pipe by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24-in. duct is required, it will be twice that of a 40 x 12-in. duct, or $2 \times 23.3 = 46.6$ in.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Side Rectangular Duct	8	8.5	9	9.5	10	10.5	11	11.5	12	12.5	13	13.5	14	14.5	15	15.5	16
3	5.2	5.4	5.5	5.7	5.8	5.9	6.0	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0	7.1
3.5	5.7	5.9	6.0	6.2	6.3	6.5	6.6	6.7	6.9	7.0	7.1	7.3	7.4	7.5	7.6	7.7	7.8
4	6.1	6.3	6.5	6.7	6.8	7.0	7.1	7.2	7.4	7.5	7.7	7.8	7.9	8.1	8.2	8.3	8.4
4.5	6.5	6.7	6.9	7.1	7.2	7.4	7.6	7.7	7.9	8.0	8.2	8.4	8.5	8.6	8.7	8.9	9.0
5	6.9	7.1	7.3	7.5	7.7	7.8	8.0	8.2	8.3	8.5	8.7	8.8	8.9	9.1	9.2	9.4	9.5
5.5	7.3	7.5	7.7	7.8	8.1	8.3	8.5	8.6	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.9	10.1

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION[a]—(Continued)

Side Rectangular Duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	8.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

[a]Additional sizes: 4 × 5 = 4.9; 4 × 6 = 5.4; 4 × 7 = 5.8; 5 × 5 = 5.5; 5 × 6 = 6.3, 5 × 7 = 6.5.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Concluded)

Side Rectangular Duct	26	28	30	32	34	36	38	40	42	44	46	48
26	28 6											
28	29.7	30 8										
30	30.7	31.9	33.0									
32	31.7	32 9	34.1	35 2								
34	32.7	33.9	35.1	36.3	37.4							
36	33.7	34.9	36.1	37.3	38.5	39.6						
38	34.6	35.9	37.1	38.4	39.5	40 7	41 8					
40	35.3	36.7	38 0	39 3	40.5	41.7	42 9	44 0				
42	36 0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46.2			
44	36.9	38 5	39 9	41.2	42.5	43 7	44 9	46.1	47 2	48 4		
46	37.8	39 3	40.8	42.2	43 5	44 8	46 0	47.2	48 4	49 5	50 6	
48	38.5	40.0	41.5	43.0	44.4	45 6	46.9	48.1	49 3	50 5	51 6	52 8
50	39.2	40 8	42.3	43.8	45.2	46.5	47 9	49 1	50 4	51 6	52 9	54 0
52	40.0	41.6	43.1	44 7	46.1	47 5	48 9	50 1	51 3	52 5	53 8	55.0
54	40.7	42 4	44.0	45.5	47 0	48 4	49.9	51.1	52.3	53.5	54 8	56.0
56	41 3	43 0	44 6	46.2	47.7	49.1	50.6	52 0	53 3	54.6	55 9	57 0
58	42.1	43 8	45.4	47.0	48.5	50 0	51 5	52.9	54.2	55 5	56 8	58 0
60	42.7	44.5	46.1	47.8	49.3	50.9	52.3	53.8	55.0	56.4	57.7	58 9
62	43 4	45.1	46 8	48 4	50 0	51.7	53 0	54.5	55.9	57.2	58 5	59 7
64	44.0	45.8	47.5	49 2	50 9	52.4	53 9	55.4	56.8	58 1	59.4	60 6
66	44.7	46.5	48.2	50 0	51.6	53 1	54.7	56.2	57.6	59.1	60.4	61.6
68	45.3	47.2	48 9	50.7	52.2	53 8	55 5	56.9	58.4	59.9	61.3	62 6
70	46.0	47.8	49.5	51 3	52.9	54.5	56 2	57.7	59 1	60 6	62 1	63 5
72	46.5	48 4	50.1	51.9	53.7	55.4	57.0	58.7	60.0	61 3	63.0	64 5

Side Rectangular Duct	50	54	60	66	72	78	84	88
50	55 0							
52	56.1							
54	57.2	59.4						
56	58.3	60.5						
58	59.3	61.6						
60	60.3	62.7	66.0					
62	61.3	63.7	67.1					
64	62.2	64.7	68 2					
66	63.2	65.7	69.3	72.6				
68	64.1	66.6	70 3	73.7				
70	65 0	67.6	71 3	74.8				
72	65.9	68 5	72.3	75.9	79.2			
74	66.8	69 4	73 3	76.9	80.3			
76	67.6	70.3	74.2	77.9	81.4			
78	68.4	71 2	75.2	78.9	82.5	85.8		
80	69 2	72.1	76.1	79 9	83 6	86.9		
82	70.1	73.0	77.1	80.9	84.6	88 0		
84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
92	74 1	77.1	81 4	85 6	89.5	93.2	96 7	99.0
94	74 8	77 8	82 2	86.5	90.4	94.2	97.8	100.1
96	75.5	78 7	83.0	87.4	91.3	95.2	98.8	101.2

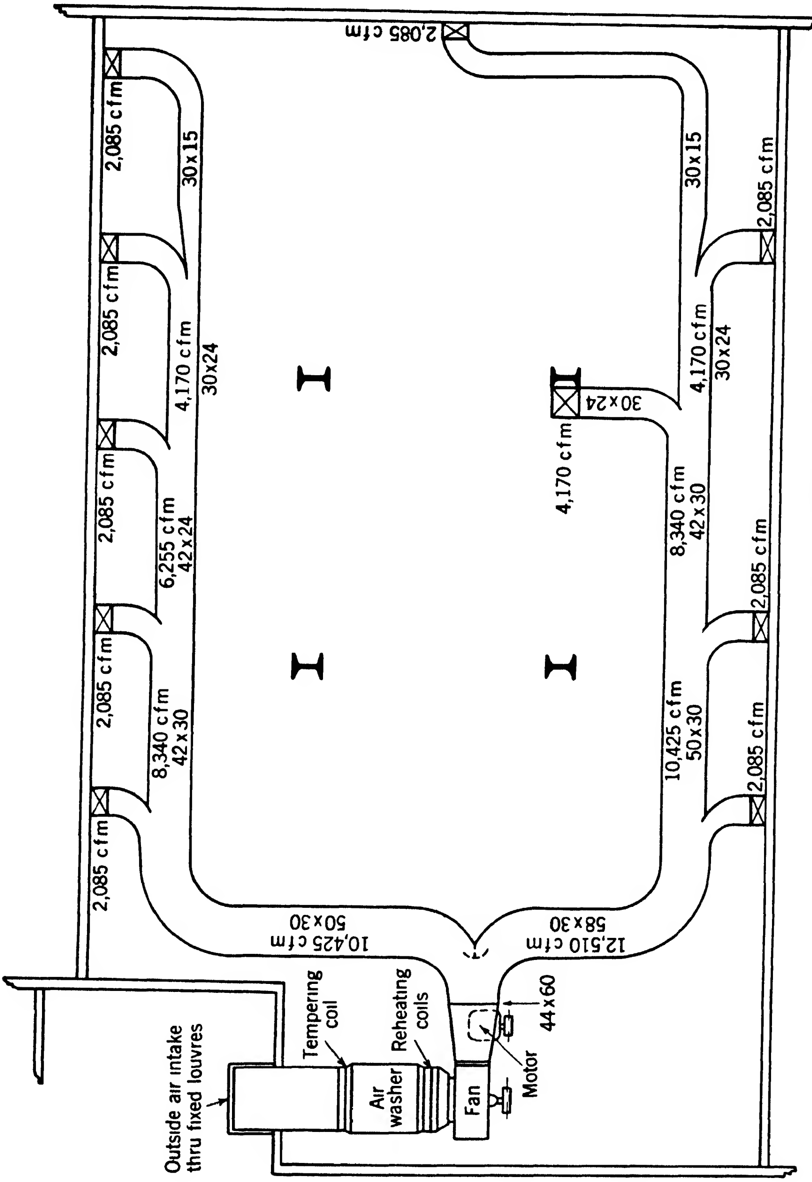


FIG. 6. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

MAIN TRUNK DUCTS

A main duct with branches is generally used to convey tempered air for ventilation purposes only. In place of individual ducts, a comparatively large main duct supplies air by branches to the room or rooms. The velocities vary according to the nature of the installation and the degree of quietness required. At the start of the run a velocity as high as 2000 fpm may be used, but this is considered the maximum for public building work, and is reduced to from 400 to 800 fpm in the risers. This duct system may be designed so that the loss of pressure in the branches is equalized in a manner similar to that previously described.

Equal Friction Method

Example 3. Fig. 6 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms and offices.

The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft. A 7½-minute air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The clear area of the fresh air inlet is based on a velocity of 1000 fpm or $\frac{22,935}{1000}$ = 22.94 sq ft. If the air washer is provided with automatic humidity control, the tempering coil should raise the temperature of the entering air to 32 F. The washer with its automatic control will then raise the temperature from 32 F to 42 F. If the washer is not provided with automatic humidity control, the tempering coil must raise the temperature of the entering air to at least 55 F to allow for some temperature drop in the washer due to evaporation. The reheating coil is selected to raise the temperature of the air from that leaving the air washer to 70 F. The air washer should have a maximum velocity of 500 fpm through the clear area, which, in this case, is 46 sq ft. For more detailed information on tempering coil and air washer control, see Chapter 37

Since the plan shows a moderately short run of main duct with no risers near the fan outlet, a fan should be selected which will have the required capacity of 22,935 cfm with a maximum velocity through the fan outlet of 1400 fpm. The outlet area, therefore, should be 16½ sq ft

The main pipe size should be selected to give a velocity equal to or less than the velocity at the fan outlet. Choosing a 56-in. pipe with a cross-sectional area of 17.1 sq ft, the velocity in the main pipe will be 1340 fpm. Using the friction pressure loss method this 56-in main pipe will be taken as the basis of calculation.

Fig 6 shows the amount of air to be handled by each section of pipe. Expressing the volume handled by each section as a percentage of the total volume and using the charts, Figs. 4 and 5, the pipe sizes are as shown in Table 2.

TABLE 2. PIPE SIZES FOR EXAMPLE 3[a]

Volume of Air (cfm)	Per Cent of Total Volume	Diameter of Pipe (Inches)	Equivalent Size of Rectangular Duct (Inches)
22,935	100.0	56	60 x 44
12,510	54.6	45	58 x 30
10,425	45.4	42	50 x 30
8,340	36.3	39	42 x 30
6,255	27.2	35	42 x 24
4,170	18.2	29½	30 x 24
2,085	9.1	23	30 x 15

[a]Velocity through diffusers (not shown) to be approximately 300 fpm.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base of each riser and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to that shown in Fig. 6 where the main duct divides into the 58 in. × 30 in. and 50 in. × 30 in. branches.

The rectangular equivalents are selected from Table 1; the width to depth proportion will be determined by construction requirements and ease of fabrication. The calculation of the friction is as follows:

The longest run from the fan outlet to diffuser is 150 ft 0 in.; 150 ft of 56-in. pipe is equivalent to $\frac{150 \times 12}{56}$ .... 32.2 diam

Two 45-in., 90-deg elbows $(2 \times \frac{45}{56} \times 8.5)$ .... 13.7 diam

(Assume each elbow equivalent to 8.5 diameters of duct, Fig. 1).

Two 23-in., 90-deg elbows $(2 \times \frac{23}{56} \times 8.5)$ .... 7.0 diam

Two 23-in., 90-deg elbows in riser $(2 \times \frac{23}{56} \times 30)$ .... 24.7 diam

(Two bad elbows in riser, each equivalent to 30 diameters of duct)

Total diameter of 56-in. pipe 77.6

The velocity head corresponding to a velocity of 1340 fpm is $\left(\frac{1340}{4005}\right)^2 = 0.112$ in.

Taking 50 diameters as one head loss, then $\frac{77.6}{50} \times 0.112 = 0.174$ in. static loss in duct.

Where the connection pieces are made with long easy slopes and the general workmanship is good, a regain in static pressure may be deducted from the foregoing pressure loss. This can be taken as approximately two-thirds the difference in velocity pressures at the fan outlet and the last run of pipe. The velocity in the riser is 667 fpm with a corresponding velocity pressure of 0.027 in. The fan outlet velocity is 1400 fpm with a corresponding velocity pressure of 0.122 in. The regain equals $\frac{2}{3}$ (0.122 − 0.027) = 0.063 in.

The net static pressure loss in the duct is:

0.174 in. − 0.063 in. .... 0.111 in.

Other friction losses are as follows:

(1) Fresh air intake 1000-fpm velocity (1½ heads × 0.0625) .... 0.094 in.
(2) Tempering coil loss (from manufacturer's tables) .... 0.100 in.
(3) Air washer loss (from manufacturer's tables) .... 0.250 in.
(4) Reheating coil loss (from manufacturer's tables) .... 0.100 in.
(5) Allowance for regulating dampers and diffusers .... 0.100 in.

Static pressure loss of system .... 0.755 in.

The fan should be selected from the manufacturer's ratings which according to the Standard Test Code for Centrifugal and Axial Fans[1], will deliver 22,935 cfm at a static pressure of 0.755 in. and which has an outlet area of 16½ sq ft.

The method of design used in Example 3 is the *equal friction method* described under the heading Procedure for Duct Design. This involves

[1]See Chapters 27 and 45.

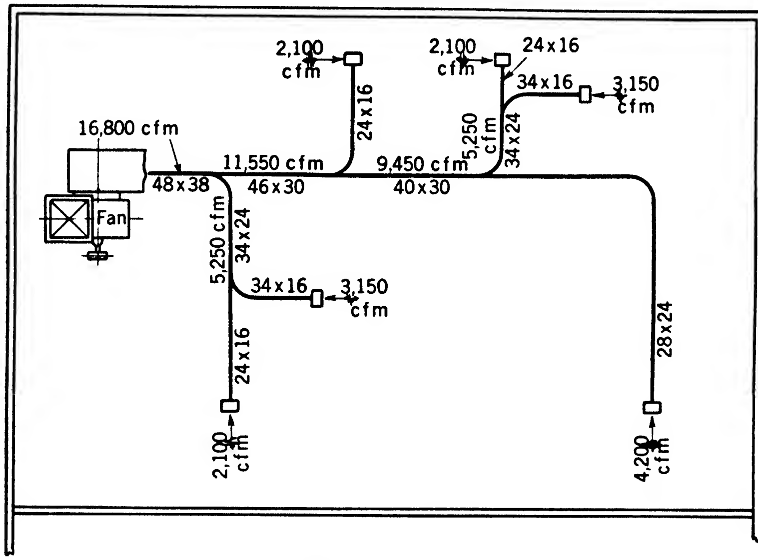


FIG. 7. EXHAUST SYSTEM LAYOUT

the arbitrary reduction of velocity from the fan outlet to the point of discharge to the room, and the friction is calculated by adding the pressure losses of each section of duct. This method requires dampering in the risers and supply branches in order that equalization of air flow can be attained.

Example 4. Fig. 7 shows an exhaust system layout for exhausting from buildings of the same type as in Example 3. Assume the air requirements based on the number of air changes per hour to be 16,800 cfm. Using a velocity of 1400 fpm in the main duct at the fan inlet, which is an average velocity for this type of system, the area of the main is 12 sq ft, which corresponds to a 47-in. pipe Referring to Example 3, and using the charts, Figs. 4 and 5, the pipe sizes are as indicated in Table 3 for both round and rectangular ducts.

TABLE 3. PIPE SIZES FOR EXAMPLE 4[a]

VOLUME OF AIR (CFM)	PER CENT OF TOTAL VOLUME	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
16,800	100.0	47	38 x 48
11,550	68.8	41	30 x 46
9,450	56.2	38	30 x 40
5,250	31.3	31	24 x 34
4,200	25.0	28.5	24 x 28
3,150	18.8	25.3	16 x 34
2,100	12.5	21.6	16 x 24

[a]Velocity through intake grilles (not shown) to be approximately 400 fpm.

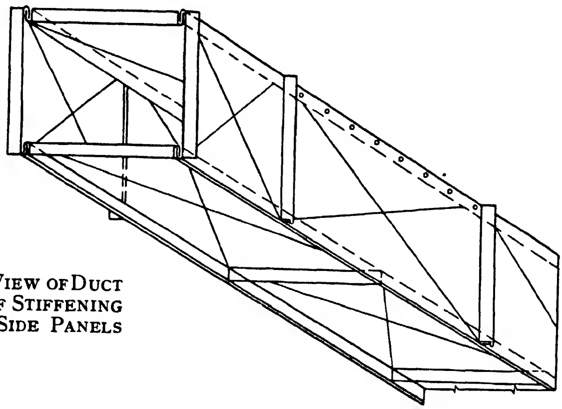


FIG. 8. ISOMETRIC VIEW OF DUCT SHOWING LOCATION OF STIFFENING SEAMS ON TOP AND SIDE PANELS OF DUCT

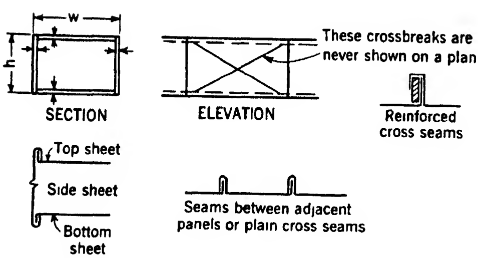


FIG. 9. DETAILS OF SEAMS

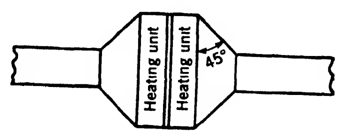


FIG. 10. METHOD OF INSTALLING HEATING UNIT

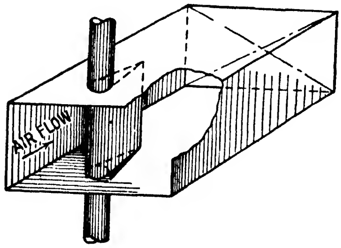


FIG. 11. INSTALLATION OF EASEMENT IN DUCT AROUND OBSTRUCTION

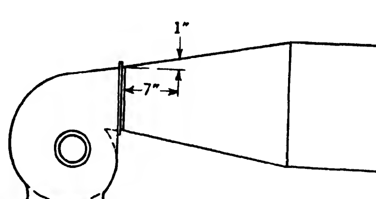


FIG. 12. FAN DISCHARGE CONNECTION

All risers will require dampering as in Example 3. The calculation of the friction is as follows:

The longest run from the intake grille to fan inlet is 100 ft.

(1) Duct friction 100 ft of 47-in. pipe $\left(\frac{100 \times 12}{47}\right)$ .... 25.6 diam

Two $28\frac{1}{2}$-in., 90-deg elbows in riser $\left(\frac{2 \times 28.5 \times 30}{47}\right)$ .... 36.4 diam

(Two bad elbows in riser each equivalent to 30 diameters of duct).

One $28\frac{1}{2}$-in., 90-deg elbow in horizontal run $\left(\frac{28.5 \times 8.5}{47}\right)$ .... 5.2 diam

Total diameter of 47-in. pipe .... 67.2 diam

Velocity head corresponding to 1400 fpm is $\left(\frac{1400}{4005}\right)^2 = 0.122$ in.

Taking 50 diameters as one head loss, then $\frac{67.2 \times 0.122}{50}$ .... 0.164 in.

(2) Intake loss from grille ($1\frac{1}{2}$ heads at a 400 fpm velocity $1\frac{1}{2} \times 0.01$) .... 0.015 in.

(3) Static pressure required to produce one velocity head at 1400 fpm .... 0.122 in.

(4) Loss occasioned by step-up of velocity (0.20×0.122) .... 0.024 in.

(This loss varies from 0 05 to 0 40 velocity head depending upon the nature of the change For average systems 0 20 velocity head is a close approximation).

Static pressure loss on inlet side .... 0.325 in.

To this must be added the resistance on the discharge side of the fan A fan outlet velocity of approximately 1500 to 1600 fpm may be used. Assuming the fan outlet to be equivalent in area to a 45-in. pipe, the velocity is 1525 fpm.

Loss on discharge (15 ft from fan outlet to discharge):

$$\frac{15 \times 12}{45} = 4 \text{ diameters of 45-in. pipe.}$$

The velocity head corresponding to a velocity of 1525 fpm is 0 145 and the discharge-side loss is $\frac{0\ 145 \times 4}{50} = 0.012$ in. The total static pressure loss of the system is then:

$$0.012 + 0.325 = 0.337 \text{ in.}$$

The fan will be selected to handle 16,800 cfm at a static pressure of 0 337 in. and to have an outlet velocity of 1525 fpm Outlet area 11 sq ft

Where there are one or more ducts with branches, the velocity of air in the ducts may be either chosen arbitrarily or calculated for friction losses. When arbitrary values are assigned, a certain amount of dampering should be provided for; this will be small when the method chosen permits a drop in velocity as the quantity of air is reduced.

After the total air quantity and the size of fan are ascertained, the main duct is usually fixed as being at least equal in area to the fan outlet, or perhaps 10 per cent greater. From this main pipe all others are proportioned. For example, if the main duct is 30 in. in diameter, a branch to carry 10 per cent of the total capacity should be 12.7 in. in diameter (see Fig. 4) in order to have the same friction per foot of length, while one

carrying one-half the total capacity of a 30-in. main with the same friction loss per foot would be 23.4 in. in diameter. By this method of equalizing friction it is unnecessary to consider the resistance of each section of pipe independently, but only to know the distance from the fan outlet to the end of the longest run of pipe, the number and size of elbows, and the diameter and velocity in the largest pipe.

Example 5. If the greatest length of piping in a system is 130 ft with a 26-in. diameter main pipe and one 20-in. elbow, the piping having been designed for equal friction per foot of length, the friction would be the same as for 130 linear feet of 26-in. pipe, or 60 diameters. To this should be added the friction loss in elbows, in this case one 20-in. elbow, which has a loss equivalent to 8.5 diameters of 20-in. pipe. This in turn is $\frac{20}{26} \times 8.5 = 6.6$ diameters of 26-in. pipe. The total equivalent length of the system will then be $60 + 6.6$, or 66.6 diameters. Since 50 diameters is equivalent to one velocity head, the loss is $\frac{66.6}{50} = 1.33$ times the velocity head. If the velocity is, for example, 2200 fpm, corresponding to 0.3-in. pressure, the friction loss of the system will be $1.33 \times 0.3 = 0.399$ in.

TABLE 4. SHEET METAL GAGES FOR RECTANGULAR DUCT CONSTRUCTION[a]

GAGE	WIDTH OF DUCT	SEAM	REINFORCED SEAM
26	Up to 12 in.		
24	13 in. to 30 in.	1	
22	31 in. to 48 in.	1	
22	49 in. to 60 in.	1½	⅛ in. x 1⅜ in.
20	61 in. to 90 in.	1½	⅛ in. x 1⅜ in

[a]If panels are not cross-broken two gages heavier material should be used.

Frequently the prevention of sound in a heating or ventilating system imposes more severe restrictions than the prevention of excessive pressure drop. This question is highly involved and requires consideration of many factors. The air velocities to be used will vary with the standard of construction used in the ducts themselves as well as with the nature of the occupancy and the construction of the building. In general, architects and engineers who leave the details of duct construction to the contractor must, of necessity, design for lower velocities than might be required for quiet operation if proper construction details were always followed. The contractor may be expected to build the ducts by the least expensive methods, and the engineer must anticipate this. For further information on noise reduction, see Chapter 30.

DUCT CONSTRUCTION DETAILS

If panel construction is used with standing seams or similar reinforcement, and the panels are cross-broken to give rigidity, there is less likelihood of vibration due to air flow, or deflection due to air pressure. Elbows made without splitters, and improperly shaped transformation sections produce high local velocities which are the cause of noise in duct work. The use of first-class duct construction with well-designed transformation sections and splitters in elbows tends to maintain relatively uniform velocities with decrease in turbulence and in the noise produced.

Figs. 8 to 12 show acceptable construction details for rectangular ducts, elbows, and transformation pieces or connections. Other methods are also acceptable, such as the use of angle iron stiffeners for large ducts. Good construction is essential to the elimination of duct noises and for the prevention of a flimsy installation.

Fig. 8 is an isometric view of a duct showing the location of the stiffening seams on the top and side panels. The cross seams should not occur at the same place but should be staggered as indicated. Heating units should be installed as shown in Fig. 10 with the duct connections making an angle of not less than 45 deg, but preferably 60 deg. Fan discharge connections should have a maximum slope of 1 in 7, as indicated in Fig. 12. Whenever a pipe or other obstruction passes through a duct an easement should be placed around the pipe as indicated in Fig. 11. The recommended gages for rectangular sheet metal duct construction are given in Table 4.

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Fan Engineering, Buffalo Forge Co.

Heat Power Engineering, by Barnard, Ellenwood, and Hirshfeld, Part III.

Mechanical Engineers' Handbook, by Lionel S. Marks, McGraw-Hill Book Co.

The Flow of Liquids, by W. H. McAdams (*Refrigerating Engineering*, February, 1925, p. 279).

A Study of the Data on the Flow of Fluids in Pipes, by Emory Kemler (*A.S.M.E. Transactions*, Hydraulics Section, August 31, 1933, p. 7).

PROBLEMS IN PRACTICE

1 ● Determine the equivalent number of diameters of straight pipe equivalent to a 90 deg elbow having center line radii of (a) 100 per cent, (b) 150 per cent, and (c) 200 per cent of the pipe diameter.

Assume 1 velocity head lost in 50 diameters.

From Fig. 1 the per cent of velocity head lost:

a. For 100 per cent radius is 25.5 per cent × 50 = 12.8 diameters straight pipe.
b. For 150 per cent radius is 17.0 per cent × 50 = 8.5 diameters straight pipe.
c. For 200 per cent radius is 14.5 per cent × 50 = 7.3 diameters straight pipe.

2 ● Why is it desirable to make elbows with a radius equal to one and one-half times the pipe diameter?

Reference to Figs 1 and 2 will show that while the loss of velocity head, as indicated by the curves, shows considerable variation for elbows between the range of 50 and 150 per cent radius, the line is practically straight after 150 per cent, indicating very little variation in loss of head for elbows of larger radius.

3 ● What is the best shape to use for ducts?

The shapes to be used in designing ducts, in the order of their preference, are round, square, and rectangular.

4 ● What determines which shape to use?

Structural and space conditions. Because ducts are as a rule part of the building or structure, it is necessary to proportion their sizes to fit the spaces available.

5 ● What is meant by "arbitrarily fix the velocity in the various sections"?

When using the velocity method as a basis for design, the maximum allowable velocity is fixed for the main supply duct at the fan, and this velocity is gradually decreased as each branch or outlet is taken off the main supply duct.

6 ● Which system of duct design is to be preferred, the velocity method or the friction pressure loss method?

The friction pressure loss method can be used to advantage where no structural or building conditions limit the shape of the ducts. Where these limiting conditions exist the velocity method is to be preferred.

7 ● Are the grille sizes figured on the same basis as the outlets?

The free area through the grilles is figured the same as the outlets, and this area is increased from 20 to 50 per cent, depending on the design of the grille, to allow for the loss of area caused by the construction of the face of the grille.

8 ● Where it is necessary to provide steel angle braces, how far apart should they be spaced?

Angle braces for large ducts should be placed on 3-ft 0-in. centers.

9 ● How much air will a 10-in. by 24-in. duct handle if it is part of a system designed on a pressure drop of 0.1 in. per 100 feet of run?

1450 cfm (Table 1 and Fig. 3).

10 ● How does a splitter at a duct junction influence the volume of the air going through each branch?

A splitter facing the direction of air flow cuts off the air and delivers the desired amount to the branch.

11 ● Why does a wide, shallow duct offer more resistance to the flow of air than does a square duct of equal cross-sectional area?

The perimeter of the wide, flat duct is greater than that of the square-section duct, so the former has the greater frictional area which increases the resistance and thus reduces the volume at any given pressure.

12 ● What methods are used to keep large ducts from vibrating because of air pulsations, and from sagging because of their own weight?

External bracing, such as standing seams, or structural shapes, like tees or angles, should be placed across the top and bottom Exterior braces or cross buckling of metal sheets in diagonal panels may be used for the sides of large ducts.

13 ● What velocities of air flow should be used in the trunk ducts of a ventilating system in a public building?

From 1200 to 1600 fpm.

14 ● In a ventilating system in a residence, what is the recommended air velocity through supply registers and grilles?

400 fpm.

Chapter 30

SOUND CONTROL

Decibel Defined, Apparatus for Measuring Noise, Problem of Sound Control, Acceptable Noise Levels, Controlling Vibration from Machine Mountings, Controlling Noise through Room Wall Surfaces, Noise Transmitted Through Ducts, Duct Lining Factor

IN ventilating and air conditioning a building or a room, the effect of the mechanical system employed must be considered on the acoustics of the space conditioned. It is important to consider also that the use of air conditioning often permits keeping the windows closed, thus giving relief from certain external noises, but at the same time increasing the necessity of providing adequate sound control.

It is not assumed that the ventilating and air conditioning engineer will attempt to improve the acoustics of the space that is being conditioned, but the designer should have at least enough fundamental knowledge of the acoustical effects of the system which is being designed to be sure that no damaging effects occur to the existing acoustical properties. It is assumed that in a given space the architect and acoustical engineer have produced a room or rooms which are satisfactory for speech, music, or other uses. The ventilating engineer's sole function is to ventilate and air condition these rooms properly so that they will be physically comfortable without adding any acoustical hazards.

UNIT OF NOISE MEASUREMENT

By a recently adopted international standard, two terms are used for noise measurement. The *decibel* (db) is the physical unit for expressing intensity or pressure levels. The *phon* is applied to what was formerly called loudness level; that is, the equivalent level in decibels of the equally loud thousand cycle tone. The decibel is defined by the relation $N = 10 \log \frac{I_1}{I_0}$, where N is the number of decibels by which the intensity flux I_1 exceeds the intensity flux I_0. The intensity flux is the measure of the energy contained in a sound wave and is defined in terms of microwatts per square centimeter of wave front in a freely traveling plane wave. It is usually more convenient to select an arbitrary reference intensity for I_0 and express all other intensities in terms of decibels above that level. For this purpose the threshold of audibility for the average human ear at a frequency of 1000 cycles per second has been selected. This reference threshold is 10^{-16} watts per square centimeter or 10^{-10} microwatts per square centimeter. This reference level also corresponds to a pressure of 0.0002 dynes per square centimeter.

A stated sound level in decibels, unless otherwise defined, will thus be related to a threshold of 10^{-16} watts. For example, a level of 60 db above

this reference threshold is 10^{-10} watts. In a similar manner, when sound measurements are given in actual intensity or energy units, they can be converted to decibels by this relation.

Since the decibel is a ratio, it can only be employed when related to a reference threshold level as given. Noise levels, which vary with frequency as well as intensity, must not only be related to this reference threshold level, but also to a reference frequency, which is taken as 1000 cycles. These terms and procedures may be found in tentative standards[1] published by the *American Standards Association.*

APPARATUS FOR MEASURING NOISE

Since the relative loudness to the ear, rather than the actual physical intensity, is the quantity in which engineers are usually interested, it has been found necessary to allow for the varying sensitivity of the ear at different frequencies in designing noise measuring equipment. The most satisfactory method of measuring noise is by means of a sound level meter which usually consists of a microphone, a high gain audio-amplifier, and a rectifying milliammeter which will read directly in decibels. This meter is calibrated to give readings above the threshold of audibility and usually contains a weighing network to make it less sensitive at those frequencies where the ear is less sensitive. For complete specifications relative to the approved type of sound level meters refer to the information[2] published by the *American Standards Association.*

GENERAL PROBLEM OF SOUND CONTROL

As previously stated, the function of the ventilating and air conditioning engineer is to add no acoustical hazard to the conditions already present in the room or building and the problem can be stated as:

a. To determine the noise level existing without the equipment.

b. To ascertain the noise level which would exist if the equipment were installed without sound control.

c. To provide as a part of the installation sufficient sound control appliances to reduce the noise level substantially to that found in (*a*).

To accomplish this the engineer should have information of three kinds:

1. A knowledge of the noise levels currently considered acceptable in various rooms in order that he may have a basis on which to proceed.
2. A knowledge of the nature and intensity of the noise created by the various parts of the equipment.
3. A knowledge of how, when necessary, to vary and control the noise level between the equipment and the conditioned space.

In addition, the engineer should have information available to deal with noises which may enter the room due to openings made into it to accommodate the equipment, such as cross talk between rooms connected with common ducts and noise transmitted to portions of duct system outside the conditioned space and through to its interior.

While the general problem may be logically outlined and the items of

[1]American Tentative Standards for Noise Measurement, *American Standards Association*

[2]American Tentative Standards for Sound Level Meters for Measurement of Noise and Other Sounds, *American Standards Association.*

knowledge necessary to its solution can be listed, the available information at present is lacking in certain respects. However, attention may be directed to that information which is currently available, and to furthermore outline a solution of the noise problem based on these data.

ACCEPTABLE NOISE LEVELS

Measurements of noise levels have been observed by several investigators in various rooms and locations. The information compiled in Table 1 is based on these data, which represent the best opinion on the

TABLE 1. TYPICAL NOISE LEVELS

Rooms	Noise Level in Decibels to be Anticipated		
	Min	Representative	Max
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence, Apartments, etc	25	35	40
Theatres, Legitimate	25	30	35
Theatres, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc	25	30	40
Churches	25	30	35
Executive Offices, Acoustically Treated Private Offices	25	33	40
Private Offices, Acoustically Untreated	35	45	50
General Offices	45	55	60
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Building, Court Houses, Post Offices, etc	45	55	60
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	60	70	80
Office Machine Rooms	60	70	80
Vehicles			
Railroad Coach	60[a]	70	80
Pullman Car	55[a]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	80	85	100

[a]For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated

subject now available. All levels are given in decibels above a reference threshold of 10^{-16} watts (corresponding to a pressure of 0.0002—dynes per square centimeter). Minimum, representative, and maximum levels are given for each application. These values are intended to indicate the variation which may be expected in different locations of the same type, but not the time variation which may be expected in each location.

The values shown in Table 1 are typical of those found currently in

existing spaces. They are, however, the noise levels of the room and not the noise levels of the ventilating or air conditioning equipment. If the noise level at the room of the equipment is kept at the levels shown in the table the equipment will not add to the acoustical hazard existing without it, provided the equipment noise is heard alone, but if both are heard together the total noise level in the room will be increased about 3 db. This is usually considered an acceptable result.

In some cases it is desirable to keep the equipment noise level at the room at such a value that it actually will not increase the noise level in the room to any measureable degree. This can usually be accomplished if the equipment noise at the room can be kept 10 db below the noise level shown in the table.

NOISE CREATED BY EQUIPMENT

Information concerning the noise levels created by ventilating and air conditioning equipment such as fans, motors, air washers, and similar items is not yet on a basis which permits tabular presentation although certain manufacturers are prepared to offer such data and do state the noise producing properties of their products.

Absence of this information makes it necessary to resort to indirect means in solving certain problems and also prevents a direct logical solution.

KINDS OF NOISE

To solve a sound problem of this type it is desirable to consider separately the several means by which noise reaches the room. This avoids to some extent the necessity of knowing the noise level at the source and places the emphasis on ascertaining the level at the point where the sound enters the room rather than on its point of origin.

The noise introduced into a room or building by ventilating or air conditioning equipment may be divided into two kinds depending on how it reaches the room as:

1. Noise transmitted through the building construction.
2. Noise transmitted through the ducts.

It is convenient to further sub-divide these two methods of delivery as

1. Noise transmitted through the building construction.
   a. From machine mountings as vibration.
   b. From equipment through room wall surfaces
2. Noise transmitted through the ducts
   a. From equipment such as sprays, fans, etc.
   b. From outside, and transmitted through duct walls into air stream.
   c. From air current, including eddying noises.
   d. Cross talk and cross noises between rooms connected by the same duct system

The next step in the solution of this problem is to present data and discuss methods whereby solutions to the noise problem can be obtained when the allowable room noise level and the path through which the noise reaches the room are known.

NOISE THROUGH BUILDING CONSTRUCTION

It is impossible to select ventilating equipment which will operate without producing some mechanical noise, and since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building itself to such a degree as to make noisy conditions in the rooms which are to be air conditioned. Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations but these mountings must be carefully designed so that they will actually reduce the contact between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if it is improperly engineered, may actually increase the contact between the equipment and the floor upon which it is supported. In general, the flexible material should be loaded as heavily as possible without impairing its load-carrying capacity.

Controlling Vibration from Machine Mountings

The theory of the insulation of vibration was first worked out by Soderberg[3]. If a machine of mass m be supported by an elastic pad the amount of vibratory force communicated by the machine to the floor or foundation upon which it rests will be determined by the elastic and viscous properties of the pad. The ratio of the vibratory force communicated to the floor or foundation with the machine resting upon the pad, and with the machine resting directly upon the floor, is given by the following equation:

$$\tau' = \sqrt{\frac{r^2 + \dfrac{1}{4\pi^2 n^2 c^2}}{r^2 + \left(2\pi n m - \dfrac{1}{2\pi n c}\right)^2}} \qquad (1)$$

where

τ' = the so-called *transmissibility* of the support.
c = the compliance (that is, the reciprocal of the force constant)
r = the mechanical resistance owing to the viscous forces within the support
n = the frequency of vibration generated by the machine which is to be insulated, such as the commutation frequency of a motor or the blade frequency of a fan.
m = the mass of the machine to be insulated.

It should be noted that not only must vibrations within the audible range of frequencies be considered, but those in the sub-audible range as well, since these may cause objectionable vibrations All the possible frequencies should be considered in the calculation. Sometimes beat effects are introduced by slight irregularities of belts or pulleys that have much lower frequencies than those of the rotating elements.

[3]C R Soderberg, (*The Electric Journal*, January, 1924), and succeeding articles See also V O Knudsen, (*Physical Review*, Vol 32, 1928, p 324), and A L Kimball, (*Journal of the Acoustical Society of America*, Vol 2, 1930, p 297)

If r, the mechanical resistance, is very small, formula 1 may be written

$$\tau = \frac{1}{\frac{n^2}{n_0^2} - 1} \tag{2}$$

where n_0 is the natural frequency of the machine upon the elastic pad,

$$n_0 = \frac{1}{2\pi}\sqrt{\frac{1}{mc}}.$$

In most cases of design of resilient machine mounting the effect of frictional resistance is small, and Equation 2 may be used. In such cases it is only necessary to know the natural frequency of the elastic pad or platform used under the desired loading and the transmissibility for any vibrational frequency of the machine may be obtained. However, this formula gives the theoretical maximum insulation which may be obtained and should be used with a liberal factor of safety. (A factor of 2 is common practice.)

If the pad is to be of any value in the prevention of solid-borne vibrations, the value of τ' must be considerably smaller than unity. If the fundamental frequency of vibration generated by the machine happens to coincide with the natural frequency of the mass of the machine resting on the elastic pad, a condition of resonance will be established, and the machine will exert a greater force upon the foundation than it would if the pad were completely removed. It is necessary, therefore, that the elastic support be sufficiently compliant, and the mass of the machine sufficiently heavy, that the natural frequency of the mass m upon its elastic support will be low in comparison with the frequencies which are generated by the machine. Thus, if the principal vibrations in the machine be of the order of 100 vibrations per second, the natural frequency of the machine mounted on its elastic support should not exceed about 50 vibrations per second, and for best results preferably 20.

When the forced frequency is low, it is frequently impossible to insulate for the fundamental forced frequency due to connecting pipe work and other relevant factors. In cases of this kind an effective installation of sound insulation may be obtained with a mounting which functions far above the fundamental forced frequency. For example, a compressor operating at 500 rpm has a forced frequency of 8.3 vibrations per second. By designing a mounting having a natural frequency of 20 to 25 vibrations per second, it is possible to isolate practically all of the noise.

The elastic support under the machine acts as a low-pass filter which passes all frequencies below about two times the natural frequency of the machine mounted on its elastic support, but prevents all frequencies above about $\sqrt{\frac{mc}{\pi}}$ from reaching the solid structure of the building. The principal influence of the internal mechanical resistance r is to limit the vibration at the resonant frequency. It is generally advisable, therefore, to use materials which have an appreciable internal resistance.

The values of c and r can be determined for any specimen of flexible material and, when known, can be used to determine the insulation value

of any particular set-up. The value of c can be obtained by making static measurements of the amount of displacement of the compressed support for each additional unit of the compressing force. If this be done for a specimen of the flexible material of a certain thickness and area of cross section, the compliance can be determined for any other thickness or area from the relation that c will be directly proportional to the thickness and inversely proportional to the area of the flexible support. When the internal resistance r is not too large, it can be determined by observing the successive amplitudes of the free vibrations of a mass m which rests upon a specimen of the flexible material, and solving for r by the usual log-decrement method. Or, if the damping be so great that the free motion of m is non-oscillatory, r can be obtained from measurements on the experimentally-determined resonance curve of the forced vibrations of m, or from measurements of the rate of return of m when it is given an initial displacement.

If the resistance of a certain specimen of material, as cork, felt, or rubber, has been determined by any of these methods, the resistance for any other thickness or area of the material can be determined approximately because the resistance will be inversely proportional to the thickness and directly proportional to the area of cross-section of the flexible support. Thus, if the values of c and r for a flexible material be known, it is possible to calculate, by means of Equation 1, the amount of insulation that will be obtained from the use of this material as a flexible support for a piece of equipment having a mass m. For the routine calculations in practice, r may be neglected with only a slight sacrifice

TABLE 2 COMPLIANCE AND RESISTANCE DATA FOR TYPICAL SPECIMENS OF FLEXIBLE MATERIALS[a]

The compliances and resistances given in the table are for specimens 1 in thick and 1 sq cm in cross-section

MATERIAL	DESCRIPTION OF MATERIAL	APPROXIMATE UPPER SAFE LOADING IN POUNDS PER SQUARE INCH	COMPLIANCE c IN CENTIMETERS PER DYNE	RESISTANCE r IN ABSOLUTE UNITS
Corkboard	1 10 lb per board foot	12	$0\ 25 \times 10^{-6}$	0.15×10^{5}
Corkboard	0.70 lb per board foot	8	$0\ 50 \times 10^{-6}$	$0\ 25 \times 10^{5}$
Fiber Board	1.35 lb per board foot	4 to 6	$0\ 60 \times 10^{-6}$	$0\ 50 \times 10^{5}$
Fiber Board	Carpet lining	10	$0\ 40 \times 10^{-6}$	....
Fiber Board	Insulating board	12	0.18×10^{-6}	....
Fiber Board	Insulating board	15	$0\ 16 \times 10^{-6}$	....
Fiber Board	Insulating board	15	$0\ 12 \times 10^{-6}$	....
Anti-Vibro-Block	....	5	$0\ 60 \times 10^{-6}$	$1\ 5 \times 10^{5}$
Sponge Rubber	25 lb per cubic foot	1 to 3	$3\ 0 \times 10^{-6}$	....
Soft India Rubber	55 lb per cubic foot	3 to 6	$1\ 2 \times 10^{-6}$	....

[a]From *Architectural Acoustics*, by V O Knudsen, p. 278.

of accuracy. Table 2 gives the values of c and r for a number of commonly used flexible materials.

Example 1. A machine weighing 1000 lb has a base area of 20 sq ft. Assume that the principal vibration of the machine has a frequency of 100 cycles per second (most machinery vibrations are less than 150 vibrations per second, and the assumed frequency of 100 is quite representative of typical machines). Suppose that a 1-in. slab of corkboard weighing 1.10 lb per board foot be placed between the machine and the floor. The loading on the cork will then be only 50 lb per square foot, or slightly more than ⅛ lb per square inch. (It is assumed that the compliance c in centimeters per dyne for a specimen 1 in. thick and 1 sq cm in cross-section is 0.25×10^{-6} and the resistance r in mechanical ohms is 0.15×10^{5}.)

The *transmissibility* is calculated in the following manner:

Mass of machine in grams = $1000 \times 454 = 4.54 \times 10^{5}$
Area of base in square centimeters = $20 \times 144 \times 2.54 \times 2.54 = 1.86 \times 10^{4}$.

Therefore, the compliance of the entire support, 1 in. thick and 20 sq ft in cross section, is $0.25 \times 10^{-6} \times \frac{1}{1.86 \times 10^{4}} = 0.134 \times 10^{-10}$ cm per dyne, and the resistance of the entire support is $0.15 \times 10^{5} \times 1.86 \times 10^{4} = 0.28 \times 10^{9}$ mechanical ohms (or absolute units). Therefore,

$$\tau' = \sqrt{\frac{(0.28 \times 10^{9})^2 + \frac{1}{4\pi^2 \times 100^2 \times (0.134 \times 10^{-10})^2}}{(0.28 \times 10^{9})^2 + \left((2\pi \times 100 \times 4.54 \times 10^{5}) - \frac{1}{2\pi \times 100 \times (0.134 \times 10^{-10})}\right)^2}}$$

$$= \sqrt{\frac{0.0784 \times 10^{18} + \frac{10^{18}}{4\pi^2 \times 10^2 \times 0.018}}{0.0784 \times 10^{18} + \left(2\pi \times 4.54 \times 10^{7} - \frac{10^{8}}{2\pi \times 0.134}\right)^2}} = 0.935$$

Consequently, it is seen that the *transmissibility* is nearly equal to unity, and that the support therefore is not satisfactory for insulating 100 or fewer vibrations per second.

If the amount of cork be reduced so that it is loaded to 10 lb per square inch, the total area of the supporting cork will be only 100 sq in. or 645 sq cm. The compliance of the entire support will now be $0.25 \times 10^{-6} \times \frac{1}{645} = 0.39 \times 10^{-9}$ cm per dyne, and the resistance will be $0.15 \times 10^{5} \times 645 = 0.97 \times 10^{7}$ mechanical ohms (or absolute units) Therefore

$$\tau'' = \sqrt{\frac{(0.97 \times 10^{7})^2 + \frac{1}{4\pi^2 \times 100^2 \times (0.39 \times 10^{-9})^2}}{(0.97 \times 10^{7})^2 + \left((2\pi \times 100 \times 4.54 \times 10^{5}) - \frac{1}{2\pi \times 100 \times (0.39 \times 10^{-9})}\right)^2}}$$

$$= \sqrt{\frac{0.94 \times 10^{14} + \frac{10^{14}}{4\pi^2 \times 0.1521}}{0.94 \times 10^{14} + \left(2\pi \times 4.54 \times 10^{7} - \frac{10^{7}}{2\pi \times 0.39}\right)^2}} = 0.0375$$

It is seen, therefore, that with the bearing surface on the cork reduced to 100 sq in. (that is, with the cork loaded to 10 lb per square inch), the *transmissibility* is reduced to 0.0375, or the amplitude of vibration transmitted to the floor will be only about 1/27 of what it would be if the machine were mounted directly upon the floor. These two numerical examples will serve to show not only the manner of making the calcu-

lations, but also the importance of selecting the proper type and design of flexible supports for insulating the vibrations of a machine from the rigid structure of a building.

Controlling Noise Through Room Wall Surfaces

The ventilating equipment is usually housed in a separate room where the noise produced by the mechanical operation of the equipment can be isolated from the rest of the building. If the vibration of the machinery is absorbed by flexible mounting and is not transmitted to the building, the only noise to be eliminated by the walls of the room will be the air-borne mechanical noise. Acoustical measurements on average brick, tile, lath, and plaster walls indicate that the usual wall of these types is sufficient to satisfactorily attenuate this air-borne mechanical noise.

Three wall sections and two floor and ceiling sections which are satisfactory for the wall insulation of the equipment room are shown in Fig. 1.

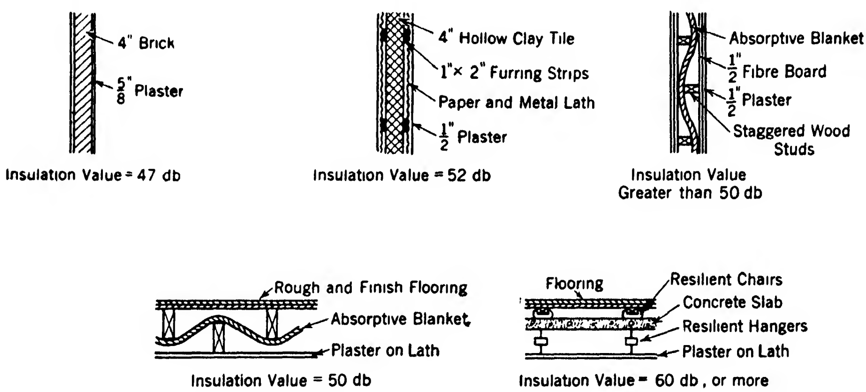

FIG 1 THREE WALL SECTIONS AND TWO FLOOR AND CEILING SECTIONS WHICH ARE SUITABLE FOR THE INSULATION OF EQUIPMENT ROOMS[a]

[a]Acoustical Problems in the Heating and Ventilating of Buildings, by V. O Knudsen (A S H V E TRANSACTIONS, Vol. 37, 1932, p 211)

Attention should be given to the equipment room door, since this door may leak badly and allow sound to escape into parts of the building which should be quiet. Where the equipment noise is particularly severe, double doors should be used and in all cases, the doors of the equipment room should be fitted with tight thresholds and weather-stripping. The door itself may transmit considerable sound if it is thin but it will not transmit a tenth as much as will be transmitted by a ¼-in. crack between the door and the threshold.

In cases where the equipment noise is extraordinarily high, it may be necessary to treat acoustically the walls and ceiling of the equipment room. If the equipment room is not entirely closed, partition walls may be necessary.

NOISE TRANSMITTED THROUGH THE DUCTS

After noise reaches the air stream in the ducts it can be controlled by lining the ducts on the inside with a sufficient quantity of sound absorbing

material. Lagging material of similar characteristics placed on the outside of ducts serves to prevent noise originating outside the ducts being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lined some of the mechanical noise from the equipment room air may be transmitted through the wall of the duct, thus reaching the air stream and be carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material such as cork, pipe covering or other sound damping material to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Measurements in one laboratory have shown that the loss through a sheet of No. 22 gage metal is 24 db. When a sheet of rock wool insulation 1 in. thick and weighing 1.4 lb per square foot is added to this, the insulation value is increased to 29 db. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense, or unless it is surfaced with another sound impervious layer such as metal or board.

Inside lining material used in the case previously mentioned would serve as an absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream.

Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays and coils; noise due to eddying currents set up by elbows, dampers and similar obstructions; and noise transmitted from room to room where there is a common duct system.

To use the lining effectively it must be properly located, well installed and be applied in sufficient quantity to reduce the noise level of the air stream to the level desired.

At present there are no wholly rational or generally recognized methods of calculating the amount of duct lining necessary to accomplish a given reduction of noise level in the air traveling in a duct system; consequently some empirical method has to be used.

Perhaps the commonest *rule of thumb* is that a length of duct should be lined which is equivalent to 10 to 15 diameters. If the duct is square, this may be interpreted to mean 10 times the average dimension. Tests have shown that this amount of lining will usually suffice to eliminate the majority of the high frequency noise which is prevalent. Since this is the more objectionable component of the noise, and since much of the low frequency noise is usually eliminated in passing through any extended supply system, this procedure is usually satisfactory except in severe conditions. It should be noted that the lining should be installed at or near the outlet, in order to effectively reduce all sounds which may be generated in the system up to this point. A more complete method of determining the necessary length of lining material has been described and is available for detailed reference[4]. Another empirical method uses

[4]The Nature of Noise in Ventilating Systems and Methods for Its Elimination, by J S Parkinson (A S H V.E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, March 1937, p 183)

a *duct lining factor* evaluated by experience. Attention is specifically called to its empirical nature and to the necessity of exercising judgment in applying it.

Use of Duct Lining Factor

A duct lining factor (*f*) giving numerical values for use at various equipment noise levels is shown in Fig. 2. When properly used with Table 1 this chart (Fig. 2) provides a solution which may be both useful and simple. It is important to understand that the levels referred to in this chart are the *average* noise levels set up in the room by the ventilating

Duct lining factor f	Equipment		
	Noisy	Average	Quiet
0	75	65	55
5	65	55	45
10	55	45	35
15	45	35	25
20	35	25	15
25	25	15	5
30	15	5	-5

FIG. 2 CHART FOR DETERMINING NOISE REDUCTION IN DECIBELS FROM DUCT LINING FACTOR[a]

[a]Values for equipment noise are only general Wherever possible substitute actual values as supplied by equipment manufacturer or as measured

or air conditioning equipment. In the case of a piece of equipment which generates a noise level of 95 db, when the noise is measured immediately next to the machine, there might be a reduction of 15 db in passing through the duct, and a further difference of 15 db between the noise at the outlet supply grille and the average level in the room, leaving an effective level of 65 db in the room. Reductions of noise level ranging from 5 to 25 db through duct systems have been encountered without the use of sound absorbing linings and the drop from supply opening to average room level may vary from 5 to 20 db.

To determine whether to use column 1, 2, or 3 in Fig. 2, in forming an estimate of the relative amount of noise generated by the system, the

length of the untreated duct system and the number of bends or elbows or splitters should be considered, since the longer and the more complex the system, the more reduction of noise level will occur before the sound reaches the room grilles. Also the sound absorbing power of the room should be taken into account, since in rooms where there is a great deal of absorptive material, such as rugs, draperies, curtains and furniture, there will be a higher loss between the outlet grille noise and the average room level. The ventilating engineer will have to judge whether the conditions deviate from the typical.

Manufacturers ratings on equipment should be considered in connection with the foregoing discussion. The quantity determined involves the noise level which will be produced in the room and the manufacturer's method of rating must be considered before allowances previously mentioned are accepted.

To use Fig. 2, proceed by consulting Table 1 and determine the probable noise level already existing in the room, and, as suggested, assume that this level is satisfactory for current practice. This gives a noise level in decibels and with this enter the chart of Fig. 2. Read across the chart and determine the value of the duct lining factor (f) in the column at the left.

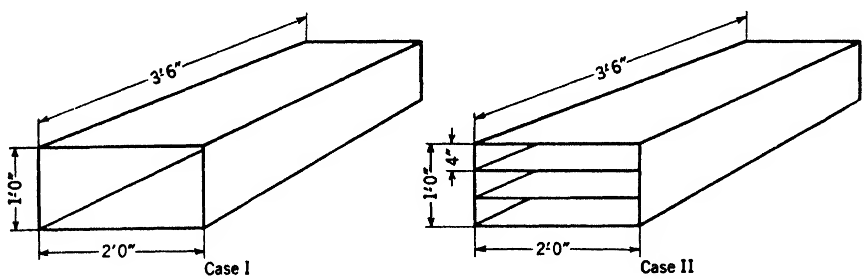

FIG. 3 DIAGRAM OF BRANCH DUCT TREATMENT WHERE LENGTH IS INSUFFICIENT FOR ADEQUATE ABSORPTION

Then multiply the smallest cross sectional dimension (inches) of the duct by this factor. The result will be the length of duct in inches to be lined to attenuate an average fan noise. If circular ducts are used, the length to be lined will be (f) $\times$ diameter of duct.

Example 2. A 7 x 30 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate a fan noise satisfactorily.

From Table 1 the noise level in this office will be 35 db.

Length to be lined for noisy equipment is $22 \times 7 = 154$ in.
Length to be lined for average equipment is $17 \times 7 = 119$ in.
Length to be lined for quiet equipment is $12 \times 7 = 84$ in.

The sound absorbent properties of duct lining are extremely important and materials which have coefficients as high as possible should be used. This is particularly true of the coefficients at the low frequencies. Fig. 2 is based on materials having a noise quieting coefficient of 0.60 or more. For materials which are less efficient a factor of safety should be added[5].

[5]For coefficients of commercial sound absorbent materials see Bulletin *Acoustical Manufacturers' Association*, 919 No. Michigan Ave., Chicago, Ill.

Only certain sound absorbent materials among those listed in various publications will be found to be suitable for duct lining. In addition to a high sound absorbent coefficient a duct lining material should have a low surface coefficient of friction, high resistance to moisture absorption, and should be fireproof and vermin proof. A number of building codes now specify that any sound absorbent material used for duct lining shall have no fire hazard. There are no existing specifications on moisture resistance but the manufacturer should be required to show that the material will not absorb sufficient moisture to cause deterioration or to decrease the sound absorbing efficiency.

If, as is often the case, the length of duct from the main duct to a grille is shorter than the length of lining indicated by using the factor found, this duct may be sub-divided[6] into smaller ducts, so that the value found may be used as shown in Fig. 3.

Example 3. Assume a branch duct, as shown in Fig. 3, is 24 in. wide by 12 in. high and 42 in. long. Use a duct lining factor of 10.

Case I. (No splitters).

Length of lining $= f \times$ minimum dimension $= 10 \times 12 = 120$ in.
In this case the duct should be lined for 120 in. which is obviously impossible.

Case II. (Two splitters).

Results in 3 ducts 24 in. wide and 4 in. high.
Length of lining $= f \times$ minimum dimension $= 10 \times 4 = 40$ in.
This length of lining fulfills the space limitations of the branch duct which is 42 in. long.

General Suggestions

In some instances where high velocity air is used, a considerable amount of whistle is generated at the grille. This noise is obviously produced after the air leaves the duct and there is no treatment which can be installed in the duct that will reduce this noise. The engineer must take into consideration the type of grille which he intends to use and provide sufficient grille area so that the velocity through the grille is reduced to a point where the grille is not too noisy.

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be sub-divided to provide ample acoustical treatment.

Very often in ventilating duct work the engineer feels that it will not be necessary to line ducts if the sound is traveling against the airflow. This, however, is untrue since sound travels so much more rapidly than does the air in even high velocity systems, that it will travel as easily against the airflow as it does with it.

Sounds which are low in pitch are much harder to eliminate from a duct system than sound which is high in pitch, consequently equipment which produces low pitched sounds should be avoided as much as possible.

REFERENCES

How Sound is Controlled, by V. O. Knudsen (*Heating, Piping and Air Conditioning*, October 1931, p. 815).

[6]Patents exist covering the sub-dividing of ducts for installing sound absorbent materials

Effect of Humidity upon the Absorption of Sound in a Room, by V O Knudsen (*Journal of the Acoustical Society of America*, July 1931). Also see report presented at the May 1933, meeting of *A.S of A*.

Acoustics and Architecture, by P. E. Sabine.

Architectural Acoustics, by V. O Knudsen

Acoustical Engineering, by West

Modern Acoustics, by Davis.

PROBLEMS IN PRACTICE

1 ● Does a soft pad under the ventilating machinery prevent building vibration?

It may or it may not. In some cases a soft pad causes more vibration than no pad A flexible mounting should be carefully designed to be effective.

2 ● Are especially designed walls necessary in an equipment room to keep noise out of adjoining spaces?

Ordinarily good brick, tile, or concrete walls are satisfactory Window and door openings should be made as tight as possible with weather-stripping, etc

3 ● How should mechanical noise be eliminated from the duct system?

A flexible connection between the fan discharge and the duct should be used. The duct should be lined from the fan end for a certain length, depending on the degree of quietness desired.

4 ● Given the choice of two types of equipment, one generating high-pitched sounds and the other low-pitched sounds, which would you choose? Why?

The equipment generating the high-pitched noise should be chosen since high-pitched sounds are more easily absorbed than are low-pitched sounds.

5 ● In building an acoustic filter in a short duct 32 by 24 in. which direction should the splitters run?

The splitters should be installed parallel to the longest dimension, since they will provide more acoustical material per splitter

6 ● What should be the characteristics of a good duct-lining material?

a. High noise reduction.
b. Physical strength.
c Easy working and installation
d Fire resistance
e Cleanliness, absence of loose fibers or pieces
f Smooth surface to reduce air friction

7 ● Should a ventilating duct be lagged or covered on the outside?

Yes, in some locations, and particularly in the equipment room and where the duct runs through noisy rooms to serve a quiet room. This lagging will prevent air-borne sounds from entering the duct through its sides and causing annoying sound in the quiet room

8 ● How can cross-talk be eliminated when one duct serves two or more rooms?

Install proper filters adjacent to the grilles in each room, using splitters if the duct leads to the rooms are short.

9 ● Space limitations and maximum air velocities for the introduction of air to a broadcasting studio restrict the size of duct to 30 by 16 in. and in addition the length of branch duct which is suitable for lining with sound absorption material is limited to 22 ft. Determine the length of duct lining necessary to attenuate an average fan noise and establish a permissible room noise level.

Referring to Fig. 2 the noise level for broadcasting studio is 14 db and the corresponding duct lining factor *f* is 28. Minimum cross-sectional dimension of duct = 16 in.

$$\frac{16 \times 28}{12} = 37.3 \text{ ft duct lining required}$$

Maximum length of duct is 22 ft, therefore it is necessary to divide the duct with a splitter, resulting in a minimum duct dimension = 8 in.

$$\frac{8 \times 28}{12} = 18.7 \text{ ft duct lining required to attenuate an average fan noise.}$$

Chapter 31

AIR CONDITIONING IN THE TREATMENT OF DISEASE

Operating Rooms, Reducing Explosion Hazards, Post-operative Heat Stroke, Nurseries for Premature Infants, Fever Therapy, Control of Allergic Disorders, Oxygen Therapy, General Hospital Air Conditioning

IN the past few years air conditioning has made considerable progress as an adjunct in the treatment of various diseases. Among the important applications are those in operating rooms, nurseries for premature infants, maternity and delivery rooms, children's wards, clinics for arthritic patients, in heat therapy, oxygen therapy, X-ray rooms, and in the control of allergic disorders.

AIR CONDITIONING OPERATING ROOMS

The most wide application of air conditioning in hospitals is that in operating rooms. Complete air conditioning of operating wards is not only desirable but often necessary for reducing the risk of explosion of modern anesthetic gases in dry winter atmospheres, and for the protection of the patient and operating personnel against excessive summer heat.

Reducing Explosion Hazard

Explosion hazards in operating rooms have begun with the introduction of modern anesthetic gases and anesthesia apparatus. Ether administered by the old drop method is still regarded as comparatively safe; but when mixed with pure oxygen or with nitrous oxide in certain concentrations (see Table 1) the explosion hazard may be as great as with ethylene-oxygen mixtures.

During the course of ethylene anesthesia the mixture, usually 80 per cent ethylene and 20 per cent oxygen, is so rich that the danger of explosion is slight, confined to an area in the immediate vicinity of the face mask, where leakage of ethylene into the air may accumulate to the lower explosion concentration (see Table 1). The most dangerous period is at the end of the operation when the patients' lungs and apparatus are customarily *washed out* with oxygen with or without the addition of carbon dioxide. Even when this procedure is omitted, it is difficult in practice to avoid dilution of the anesthetic gas with air during the normal course of breathing following the administration of anesthesia. In either case the mixture would pass through the explosion range and extra-

ordinary precaution is necessary for the safety of the patient and operating personnel.

Copious ventilation, from 6 to 12 air changes per hour, is necessary to preclude accumulation of explosive mixtures and to reduce the concentration of anesthetics to below the physiologic threshold so that the surgeon and his personnel will not be affected.

The most important cause of accidents is probably static sparks which may result from accumulation of frictional charges on the rubber surfaces of the anesthesia apparatus, on woolen blankets, and on the bodies of the operators as they walk on insulated floors, when the humidity is quite low. Grounding the various parts of the anesthesia apparatus is not entirely effective, so long as rubber remains in use in the conventional equipment.

To prevent accumulation of static charges within the apparatus or on persons coming near to it, the measures proposed[1] are humidification of air to between 55 and 60 per cent relative humidity, grounding the

TABLE 1. APPROXIMATE LIMITS OF INFLAMMABILITY OF ETHYLENE AND ETHER[a]

Mixed With	Ethylene		Ether	
	Lower Limit Per Cent	Upper Limit Per Cent	Lower Limit Per Cent	Upper Limit Per Cent
Air	3 0	30 ±	1 7	50 −
Oxygen	3 0	80 −	1 7	40 ±
Nitrous Oxide	.....	.....	3 8	26 ±

[a]Limits of Inflammability of Gases and Vapors, H F Coward and G W Jones, *U S Department of Commerce, Bulletin* No 279, 1931

apparatus and operating table, and using conducting floors and shoes so that the operating staff and attendants will be always grounded as they move about. The significant factor is the absolute humidity, rather than the relative humidity, because upon it depends the electrical conductivity of the atmosphere. The principal objection to artificial humidification is the necessity of constant supervision to make sure that the apparatus is functioning properly.

Artificial humidification in operating rooms during cold weather may also prove beneficial in reducing evaporation from exposed tissues and from the wet skin of the patient, and by allowing a lower room temperature.

Operating Room Conditions

Little is known about optimum air conditions that are necessary to maintain a normal body temperature during the course of anesthesia and in the immediate post-operative period.

Under the influence of anesthesia a patient is at a very low ebb. All anesthetics, as a rule, produce dilation of the vessels in the skin and much sweating, particularly in the case of ether anesthesia. The loss of body heat is increased considerably, while the general metabolism may be

[1]The Hazard of Explosion of Anesthetics, by Y Henderson Report of the Committee on Anesthesia (*Journal American Medical Association*, 94 1491, 1930)

depressed. The organism loses ability to regulate its own body temperature and becomes unusually sensitive to chilling and post-operative complications. In order to maintain a normal body temperature, a high air temperature is necessary, as high as 90 F or higher in the case of ether anesthesia, judging from experiments on animals[2].

Such high temperatures are obviously uncomfortable for the operating personnel, and in order to alleviate the condition the room temperature is usually kept between 72 and 80 F in cold weather with the patient carefully guarded with blankets and hot water bottles during and for some time after the operation.

Post-operative Heat Stroke: It would seem that surgeons have learned to fear so much the occurrence of post-operative pneumonia and shock that even in hot summer weather patients are sometimes needlessly *bundled up* with detrimental consequences.

In 1916 several deaths were reported[3] of heat stroke following surgical operations, and a number of cases suffering from a mild isolation, often recognized as post-operative reaction or shock. From these observations it was concluded that all operating room activities should cease during summer heat waves with the exception of urgent operations, when every effort should be made to keep the patient cool and comfortable.

In cases of exophthalmic goitre, one investigator[4] warns most emphatically against the performance of operations in extremely warm weather, for under such conditions the risk in spite of all precautions (prior to the introduction of summer cooling in operating rooms) is too great. An analysis of several cases over a 10-year period shows a striking rise of post-operative deaths in June, July, and August, resulting unexpectedly from extreme post-operative reaction passing onto acute hyperthyroidism.

More recently four cases were reported[5] of post-operative heat stroke admitted 24 hours preceding operation and sheltered from direct sun rays. All four were not ill and apparently were good risks. There occurred, however, at the time of operation and for several days preceding it, a heat wave with a moderately high temperature, a high relative humidity, and no wind. In addition to warm weather, excessive loss of body fluids is believed to have been a factor in the production of heat stroke in those four cases.

Aside from the possibility of post-operative heat stroke in warm and sultry weather, the surgeon is also concerned with the lowered recuperative power of the patients, and with his own discomfort as well as the discomfort of his team, which impairs the efficiency of the technic to the disadvantage of the patient.

In view of this experience it is customary to defer major operations as much as possible until the passing of heat waves, in hospitals not equipped with cooling facilities. But there are exceptional cases, like acute appen-

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[2]Heat Regulation and Water Exchange The Influence of Ether in Dogs, by H G Barbour and W Bourne (*American Journal Physiology*, 67 399, 1924)

[3]Post-operative Heat Stroke, by A V Moschcowitz (*Surgery, Gynecology and Obstetrics*, 23 443, 1916)

[4]The Effect of Heat Upon Operations for Exophthalmic Goitre, by A J Walton (*British Medical Journal*, 1 1045, 1923)

[5]Post-operative Heat Stroke, by T M Martin (*Journal Missouri Medical Association* July, 1928 *Abstract Anesthesia and Analgesia*, 8 23, 1929)

dicitis for instance, which sometimes come with summer heat waves, and develop dangerously unless promptly operated upon. Complete air conditioning of operating rooms would therefore seem to be a necessity in many sections of the United States.

Satisfactory Air Conditions: Although the comfortable air conditions for the operatives are not identical with those of the patient, a compromise is as a rule not difficult; with a relative humidity of 55 to 60 per cent, a temperature of 80 F in warm weather and between 72 and 75 F in cold weather will probably prove satisfactory. Additional heat may be furnished to the patient locally or by suitable covering according to body temperature in individual cases.

Central station air conditioning plants and individual unit air conditioners proved satisfactory in operating rooms when producing between 8 and 15 air changes per hour of filtered and properly humidified air, with full provision for summer cooling and dehumidification and without recirculation during the course of anesthesia. A separate exhaust fan system is as a rule necessary in order to confine and remove the gases and odors. Double windows are desirable and often necessary to prevent condensation and frosting on the glass in cold weather and to minimize drafts. The high air flow of 8 to 15 air changes in operating rooms is desirable for three reasons: (*a*) to reduce the concentration of the anesthetic to well below the physiologic threshold in the vicinity of the operating personnel, (*b*) to remove excessive amounts of heat and sometimes moisture from sterilizing equipment if inside the operating room, from the powerful surgical lights, solar heat, and from the bodies of the operatives, and (*c*) to provide extra capacity for quickly preparing the room for emergency operations. Much can be gained by careful insulation of sterilizing equipment and by thorough exhaust ventilation of sterilizing rooms adjoining the operating rooms.

It is generally believed that in addition to operating rooms, an adjoining ward should also be conditioned to provide for the treatment of post-operative fever. Such a post-operative ward may also prove valuable in treating patients with heat stroke, fevers, summer diarrhea and other cases affected by high temperature, when the room is not used for anything else.

Sterilization of Air in Operating Rooms: Of considerable significance to operating rooms and contagious wards is the use of ultra-violet radiation for sterlizing the air.[6] Results reported[7] would seem to indicate that the post-operative temperature rise of patients during the first few days is in most instances caused more by bacterial contamination of the operative wound than by the absorption of blood and traumatized tissues. Operating room infections, which were quite frequent before the installation of special ultra-violet lamps, are said to have practically disappeared.

NURSERIES FOR PREMATURE INFANTS

One of the most important requirements in the care of premature infants is the stabilization of body temperature. This is necessary because

[6]Air-Borne Infection and Sanitary Air Control, by W F Wells (*Journal Industrial Hygiene*, 17 253, 1925)

[7]Sterilization of the Air in the Operating Room by Special Bactericidal Radiant Energy, by Deryl Hart (*Journal Thoracic Surgery*, 6 45, 1936).

their heat regulating system is not fully developed; the metabolism is low and the infants generally exhibit marked inability to maintain a normal body temperature by their own efforts. The resistance to infection is low and the mortality rate, very high.

Air Conditioning Requirements

The optimum air conditions for the growth and development of these infants were determined by extensive research at the Infants Hospital, Boston, Mass.,[8] using four valid criteria, namely, stability of body temperature, gain in weight, incidence of digestive syndromes, and mortality. Wide variations were found in individual requirements for temperatures from 72 to 100 F, according to the constitutional state of the infants and body weights. The optimum relative humidity was about 65 per cent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F temperature and 65 per cent relative humidity was found to satisfactorily fulfill the requirements of the majority of premature infants. Additional heat for weak or debilitated infants may be furnished in the cribs or by means of electric incubators placed inside the conditioned nursery and the temperature adjusted according to individual requirements. In this way multiplicity of chambers and of air conditioning apparatus is obviated; the infants in the heated beds derive the benefit of breathing cool humid air, and the nurses and doctors need not expose themselves to extreme conditions.

Importance of Humidity: Although external heat is an important factor in the maintenance of normal body temperature, humidity appears to be of equal or greater importance. When the premature nurseries at the Infants Hospital were kept at relative humidity between 25 and 50 per cent for two weeks or longer, the body temperature became unstable, gains in weight diminished, the incidence of gastro-intestinal disturbances increased, and the mortality rose. On the other hand, continuous exposure to air conditions with 55 to 65 per cent relative humidity gave satisfactory results over a period of years.

The initial physiologic loss of body weight (loss occurring within first four days of life) was found to vary inversely with the humidity. In the old nurseries with natural humidity it averaged 12.4 per cent of the birth weight; in the conditioned nurseries it was 8.9 per cent with 25 to 49 per cent relative humidity, and 6.0 per cent with 50 to 75 per cent relative humidity. The number of days required to regain the birth weight was correspondingly maximum in the old nursery, minimum in the conditioned nurseries under high humidity, and intermediate in the conditioned nurseries with low humidity.

Maximum gains in body weight occurred in the conditioned nurseries under high humidity (55 to 65 per cent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 per cent) in the same nurseries, and in the old nurseries prior to the installation of air conditioning apparatus.

The incidence and severity of digestive syndromes, with diarrhea,

[8]The Premature Infant. A Study of the Effects of Atmospheric Conditions on Growth and on Development, by K. D. Blackfan, C. P. Yaglou and K. McKenzie (*American Journal Disease of Children*, 46 1175, 1933)

persistent vomiting, diminishing gains or loss of body weight, and other symptoms, were generally from two to three times as high under low than under high humidity.

Finally, the mortality of premature infants was found to be greatly affected by humidity. In Table 2 is given the net mortality according to the humidity in which the derangement of body function began. In the old nurseries, prior to the installation of the air conditioning system, the death rate from acute and chronic infections was 26.5 per cent as compared with 9.7 per cent in the conditioned nurseries under low humidity and 0.0 per cent under high humidity.

Summarizing the conclusions of these studies, the best chances for life in premature infants are created by maintaining a relative humidity of

TABLE 2. NET MORTALITY OF PREMATURE INFANTS ACCORDING TO HUMIDITY[a]

Infants Hospital, Boston, Mass

Cause of Death	Unconditioned Nurseries (1923-1925)	Conditioned Nurseries (1926-1929)	
	Natural Humidity	Relative Humidity	
		25-49 Per Cent	50-75 Per Cent
	Per Cent Mortality	Per Cent Mortality	Per Cent Mortality
Acute and chronic infections	26 5	9 7	0 0
Congenital deformities	1 2	0 0	0 7
Unclassified	1 2	4 8	0 0
All causes	28.9	14 5	0 7

[a]Excluding cases with multiple congenital anomalies incompatible with life, and also deaths occurring within 48 hours after admission to the hospital

65 per cent in the nursery and by providing a uniform environmental temperature just sufficiently high to keep the body temperature within normal limits. Medical and nursing care are, of course, factors of equal and sometimes of greater importance.

Air Conditioning Equipment: Most of the installations now in use are of the central station type providing for filtration, for humidification and heating in cold weather, and for cooling and dehumidification in hot weather. A high ventilation rate, between 15 and 25 air changes, is desirable to remove odors and maintain uniformity of temperatures in extremes of weather. Recirculation is not used extensively in these wards owing to odors and the possibility of infection.

AIR CONDITIONING IN FEVER THERAPY

Artificial production of fever in man is an imitation of nature's way of overcoming invading pathogenic organisms. The action may be direct and specific by obliteration or destruction of the invading organism within the safe limit of human fever temperature; or an indirect one in case of heat resistant organisms, through general mobilization of the defensive

mechanisms of the body, by means of which the activity of pathogenic bacteria and their toxins may be retarded or neutralized.

The limits of induced systemic fever are usually between 104 and 107 F (rectal), and the duration from 3 to 6 hours at a time. The total period of fever treatment varies with the type of the organism involved from a few hours to 50 hours or more.

The diseases reported to respond favorably to artificial fever are: gonorrhea, neurosyphilis, chorea, asthma, peripheral vascular diseases, occular gonorrhea and syphilis. There are a number of other conditions in which the usefulness of artificial fever is not yet settled. The most striking results are seen in gonorrhea in which various strains of organism can be killed by artificial fever within the limits of tolerance of man.

Equipment for the Production of Systemic Fever

Various means have been tried for producing artificial fever, including injections of various crystalloid or colloid substances; a number of physical methods, such as hot baths, radiant heat, diathermy, radiothermy, and, in the last few years, an air conditioned chamber. The relative advantages and disadvantages of these various methods were discussed in recent papers.[9] The results by the use of air conditioned cabinets have not been fully explored, and it is therefore difficult to determine the advantages and disadvantages of the value of air conditioning at this time. Under certain conditions a combination of systemic fever and additional local heating by diathermy or other means is claimed to yield better results than systemic fever alone by reducing considerably the killing time of the organism and rendering the treatment less trying to both patient and attendants.[10]

The air conditioned chamber[11] consists of an insulated cabinet approximately 6 ft long, 3 ft wide, and 2.5 ft high, containing in a small rear compartment, electric air heaters, a water pan for humidification, a centrifugal fan, and controls. The nude patient lies on an air mattress inside the front compartment with his head protruding outside the front end through a rubber collar. Warmed air at 130 to 150 F and 30 to 50 per cent relative humidity is blown upon the body of the patient, and the rectal temperature rises to 105 F usually in from 40 to 60 min. The heat is then turned low and adjusted so as to maintain the desired body temperature in each individual case.

More recently a heat cabinet was described[12] in which saturated air between 100 and 120 F in temperature is used for elevating the patient's body temperature. This gives a rapid rise of body temperature with a relatively low air temperature; it eliminates skin burns, and the room in which the heat box is located is not overheated unduly.

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[9]Fever Therapy for Gonococcic Infections, by A U Desjardins, L G Stuhler and W. C Popp (*Journal American Medical Association*, 106 690, 1936)
Artificial Fever Therapy as a Therapeutic Agent, by H P Doub (*Radiology*, 25 360, 1935)

[10]The Treatment of Gonorrheal Arthritis by Means of Systemic and Additional Focal Heating, by W Bierman and C Levenson (*American Journal Medical Science*, 191 55, 1936)

[11]Artificial Fever Therapy of Syphilis, by W M. Simpson (*Journal American Medical Association*, 105 2132, 1935)

[12]Fever Therapy Induced by Conditioned Air, by F C Houghten, M B Ferderber and Carl Gutberlet (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February, 1937, p. 115).

Extensive research is now in progress to determine the usefulness and limitations of fever therapy on a wide variety of pathogenic conditions. While this form of therapy is rapidly gaining wide recognition, its application, according to the *American Medical Association*, should be strictly a hospital procedure surrounded with the safeguards commonly employed in a major operation and under the direction of skilled physicians.

CONTROL OF ALLERGIC DISORDERS

Although there is some division of opinion over the ultimate cause of allergy, the prevailing belief is that it is due to an inherited or acquired hypersensitiveness to foreign or pollen proteins in certain individuals who react abnormally to the offending substance. The reaction may be induced by inhalation, eating, or absorption of the allergens through the skin. The clinical manifestations are hay fever, asthma, eczema, hives contact dermatitis, etc.

Symptoms of Hay Fever and Asthma

The respiratory tract is probably the most usual site of allergic manifestations, the so-called hay fevers and asthma. In hay fevers, the nose and eyes are red and itchy, and there is considerable discharge. Nasal obstruction is the most common and most distressing symptom. The severity of the symptoms varies widely from day to day depending chiefly on the amount of pollen in the air.

Seasonal asthma comes in attacks. The most popular theory concerning the mechanism of action is that the offending substance irritates the nerve endings in mucous membranes of the respiratory tract, causing spasmodic contraction of the small bronchioles of the lungs, which interferes with breathing, particularly with expiration. Non-seasonal allergic disturbances are sometimes attributed to house or street dusts, fungi, odors and irritating gases, and heat or cold, particularly sudden temperature changes. It is often stated in the literature that heat regulation in asthmatic individuals is likely to be unstable, with a tendency to subnormal body temperature. Many allergic cases who are apparently well, develop their attacks when cold weather appears, or upon changing from warm to cool outdoor air.

Air Conditioning Apparatus

In recent years considerable effort has been directed toward the elimination of the principal cause of allergy from the air of enclosures by filtration or other air conditioning processes capable of removing pollens, in the hope of providing relief to individuals who failed to respond to medical treatment (desensitization or immunization).

Paper or cloth filters, mounted in inexpensive window or floor units, proved quite satisfactory in removing all but traces of pollen. Allergens may also be removed by passing the air through a water spray, or over cooling coils kept at a temperature low enough to cause condensation of atmospheric moisture on the surface of the coils.

Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, a certain amount of cooling and dehumidification

appears to be desirable. A comfortable temperature between 75 and 82 F in warm weather and a relative humidity well below 50 per cent proved satisfactory.[13] Direct drafts, overcooling or overheating are apt to initiate or aggrevate the symptoms.

Limitations of Air Conditioning Methods

The results obtained with air filtration or other air conditioning processes in the control of allergic conditions are fairly comparable to those obtained by desensitization treatment so long as the patients remain in the pollen free atmosphere. But while specific desensitization is preventive and in a few instances curative for all practical purposes, filtration gives only temporary relief. With rare exceptions, the symptoms recur on exposure to pollen laden air. Moreover the usefulness of air conditioning methods is limited because all cases are not caused by air-borne substances. Cases of bacterial asthma do not respond at all to the treatment with filtered air.

Despite these limitations air conditioning methods possess definite advantages in the simplicity of treatment, convenience, and under certain conditions almost immediate relief. Hay fever cases are usually relieved of most of their symptoms within an hour or so after exposure to properly filtered air. In pollen asthma cases relief comes more slowly, usually after an exposure of from 1 to 12 days depending upon the severity of asthma.

A pollen free atmosphere is especially valuable for patients in whom desensitization has given little or no relief, and in instances in which desensitization is not advisable owing to intercurrent illness. On the whole, conditioning methods are considered to be a valuable adjunct in medical diagnosis and treatment of allergic disorders.

AIR CONDITIONING IN OXYGEN THERAPY

Oxygen therapy is the principal measure employed for preventing and relieving the distressing symptoms of anoxemia, which is a deficiency in the oxygen content of the blood. Some of the more important conditions in which oxygen treatment is believed to be beneficial are pneumonias, anemia, heart affections, post-operative pulmonary disturbances, certain mental disturbances, asphyxia, asthma and atelectasis in new-born infants.

The necessity of air conditioning in oxygen therapy arises from the fact that oxygen is too expensive a gas to waste in the ventilation of oxygen tents and oxygen chambers. The oxygen rich atmosphere in these enclosures is therefore reconditioned in a closed circuit by removal of excess heat, moisture, and carbon dioxide given off from the occupants.

Oxygen Tents: In oxygen tents the air enriched with oxygen is usually circulated by means of a small motor blower which sends the air over soda lime to remove carbon dioxide and then over ice to remove excess heat and moisture. The concentration of oxygen in the tent is regulated by means of a pressure reducing valve and flow meter. In an inadequately

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[13]The Effect of Low Relative Humidity at Constant Temperature on Pollen Asthma, by B Z Rappaport, T Nelson and W H Welker (*Journal Allergy*, 6 111, 1935)

cooled oxygen tent, high temperatures and humidities are inevitable, increasing the discomfort of the patient and imposing an added strain on an already overburdened heart. Oxygen therapy under such conditions may do more harm than good[14]. An ice melting rate of about 10 lb per hour gives satisfactory results in patients with fever in a medium size tent.

Oxygen tents are somewhat confining to the patient; the restless type of person is difficult to control, and the delirious impossible to control. Medical and nursing care is complicated, as the tent must be opened or removed with attendant loss of oxygen. Oxygen concentrations of 50 per cent or more are difficult to maintain, and it is a problem to keep the temperature and humidity low enough in hot weather. The direct advantages are portability and low cost.

Oxygen Chambers: The conventional oxygen chamber is an air-tight sheet metal enclosure of fire-proof construction, large enough to accommodate one or two patients. Trap doors or curtains are provided for the personnel, food and service to avoid loss of oxygen. Glass windows in the ceiling and walls admit light from electric fixtures outside the chamber.

The air conditioning system may be of the gravity type, or of the fan type using mechanical refrigeration or silica gel for drying the air. The gravity system includes a bank of brine coils controlled thermostatically, which dehumidify and cool the air. The cool air falls over trays at the bottom of the coils, containing soda lime to remove the carbon dioxide given off by the occupants. A heater at the base of the opposite wall warms the air to the desired temperature. Ordinary industrial oxygen is introduced from storage tanks outside the chamber and the concentration is regulated according to the prescription of the physician. The only change of air in the chamber is that taking place by leakage through trap doors.

The chief objections to the gravity circulation system are stratification of cold air near the floor and accumulation of odors, which may require the use of activated charcoal or an excess of oxygen for deliberate aeriation.

The fan circulation systems include compact extended surface coolers, heaters, and sometimes silica gel beds installed outside the chamber for the removal of moisture. A spray dehumidifier is not suitable for this purpose because it is often desired to cool the air below 32 F in order to obtain low relative humidities.

The temperature and humidity requirements in oxygen therapy depend primarily upon the physical condition of the patient, and secondarily upon the type of disease. In pneumonias, the range of satisfactory conditions is placed between 60 and 75 F with 20 to 50 per cent relative humidity, depending on the condition of the patient.

Oxygen chambers are unquestionably more comfortable than oxygen tents. The patients receive unhampered medical and nursing care, and the oxygen concentration, the temperature and humidity can be adequately controlled at any desired level. The chief disadvantage is high initial and operating costs in comparison with oxygen tents or with the nasal catheter method of oxygen administration. The nasal catheter

[14]General Measures Employed in the Treatment of the Pneumonias, by J G M Bullowa (*Health Examiner* 5 12, 1936)

method is the simplest and most inexpensive of all but it may cause considerable discomfort to the patient and it is not satisfactory for continuous administration and in restless or delirious patients. Moreover, oxygen concentrations greater than 40 per cent in the inspired air are difficult to obtain.

The chamber method is of value in large hospitals and for research and experimental purposes, but for routine oxygen therapy alone it may prove a liability rather than an asset in many hospitals.

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of large hospitals involves a capital investment, depreciation and running expense which may not be justified.

In clean and quiet districts, the requirements of almost all general and private wards during the cool season of the year can be satisfactorily fulfilled by the use of rational heating in conjunction with window air supply and gravity or mechanical exhaust. Insulation against heat and sound is much more important than humidification in winter; it will also help considerably in keeping the building cool in warm weather. Excessive outside noise and dust may require the use of silencers and air filters in the window openings.

Cooling and dehumidification in warm weather are important. In new hospitals particularly, the desirability of cooling certain sections of the building should be given serious consideration. Financial reasons may preclude the cooling of the entire hospital, but the needs of the average hospital can be met by the use of built-in room coolers and a few portable units which can be wheeled from ward to ward when needed.

In the North and certain sections of the Pacific Coast, cooling is needed on but a few days during summer, while in the South, built-in room coolers can be used to advantage from May to October, and in tropical climates almost continuously throughout the year. Objectionable noise is an important drawback to the use of self-contained units, but the difficulty is gradually being overcome by improvements in design.

Aside from comfort and recuperative power of the patients, cooling is of great assistance in the treatment of pyrexias in the new-born and in post-operative cases, in enteric disorders, fevers, heat stroke, heart failure, and in a variety of other ailments which often accompany summer heat waves.

Considerable research is now in progress on the influence of air conditioning upon a wide variety of diseases such as pneumonia, upper respiratory diseases, tuberculosis, arthritis, nervous instability, hyperthyroidism, essential hypertension, skin diseases, vascular disorders, and others. The field is a fruitful one having many possibilities.

PROBLEMS IN PRACTICE

1 ● Where has air conditioning in hospital wards proved itself of sufficient value to justify the expense?

In nurseries for premature infants, anesthesia and operating rooms, oxygen therapy chambers, heat therapy rooms or cabinets and allergic wards

2 ● What is the major problem in conditioning hospitals?

For general hospital wards, the major problem seems to be one of providing adequate amounts of ventilation rather than air conditioning, with some provision for cooling over-heated wards on unusually warm summer days.

3 ● What are the usual requirements for ventilation of operating rooms?

To preclude the accumulation of explosive mixtures and to reduce the concentration of anesthetics below the physiologic threshold, it is desirable that ventilation to the extent of 6 to 12 air changes be provided.

4 ● What are the optimum air conditions for premature infants?

The best chances for life in premature infants are created by maintaining a relative humidity of 65 per cent and an environmental temperature sufficiently high to keep the body temperature within normal limits

5 ● How does air conditioning assist in the treatment of allergic disorders?

In cases in which the individual fails to respond to medical treatment, air conditioning may provide a valuable adjunct in relieving the symptoms Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, it has been found that in warm weather temperatures between 75 and 82 F and relative humidities well below 50 per cent are more conducive to comfort.

Chapter 32

RAILWAY AIR CONDITIONING

Passenger Car Ventilation, Quantity of Outside Air, Method of Air Distribution, Air Cleaning, Steam or Vapor Heating Equipment, Cooling Equipment, Humidity and Temperature Control, Power Supply, Installation and Operating Costs

THE general principles of air conditioning as applied to buildings also apply to railway passenger cars, but due to space and weight limitations and the severity of the service, equipment designed for stationary work is seldom suitable for car installations. Equipment for railway use must be safe, reliable, compact, light in weight, accessible for inspection and repairs, automatic in operation and in addition, have low initial, operating, and maintenance costs. To air condition a passenger car properly, ventilating, filtering, heating, cooling, humidifying, and control equipment must be provided together with an adequate power supply. Air from the interior of the car is mixed with air from the outside and passed through the air conditioning unit where it is heated or cooled, humidified or dehumidified and delivered to the interior of the car through suitable ducts and grilles.

PASSENGER CAR VENTILATION

One of the important problems in connection with air conditioning of cars is that of ventilation. In non air-conditioned cars, ventilation is accomplished by exhaust fans, roof ventilators and open doors and windows. This practice provides an ample supply of outside air but does not prevent the entrance of smoke, cinders, and dirt.

Quantity of Outside Air

An average passenger car contains approximately 5000 cu ft of air and may seat as many as 80 passengers. The occupants are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from their breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidification, but the other constituents can be successfully handled only by proper ventilation and air cleansing. In the average car from 2000 to 2500 cfm should be circulated by the air conditioning unit. Some of this air may be recirculated, but a portion of it should always be brought in from the outside. The amount of outside air required depends upon the type of car, number of passengers, air temperature, humidity, odors, and whether or not occupants are smoking, and will vary from 15 to 90 per cent of the total air circulated. The per-

centage of outside air should be kept as low as possible to maintain the air in the proper condition in order to minimize the heat or cooling load.

For normal conditions, 10 cfm of outside air per passenger is sufficient. When smoking is permitted, 12 to 15 cfm per passenger should be admitted. Under exacting demands and adverse condition, it may be necessary to increase the quantity to 20 cfm per passenger.

Method of Air Distribution

Various methods may be used to distribute the air delivered to the interior of the car by the circulating fan or blower. The methods commonly used are:

1. A duct lengthwise along the center of the car
2. One or two side ducts built on the outside of monitor-roofed cars, or on the inside of turtle-backed or arched-roofed cars.
3. Free discharge at the end bulkheads, or by free discharge from a unit placed overhead in the center of the car, discharging toward the ends.

For details of air distribution and duct design, see Chapters 28 and 29.

Smoking rooms present a special problem. The cloud of smoke that usually hangs near the ceiling can be broken up by having the incoming air directed along the ceiling in all directions at a velocity somewhat higher than that used for the rest of the car. The air should be exhausted from the room by a fan or through a grille to the washroom or lavatory, and then outside by a fan in a ventilator.

For compartments an adjustable supply duct outlet grille of suitable size and design should be provided and provisions made in the door or partition for the removal of the air to be recirculated.

The grille used for this purpose should be designed and arranged so as to obstruct the vision of passengers, but still allow the air to pass from the room to the recirculating grille at the air conditioning unit.

Lower berths in sleeping cars and office cars should be provided with an adjustable air outlet which will discharge the amount of air desired at low velocity in any direction so that the occupant can regulate the ventilation to meet his own requirements.

In cars containing but one or two rooms or compartments, satisfactory results may be obtained by discharging the air directly from the conditioning unit into the upper part of the car. Care must be taken to have a proper discharge velocity. If the velocity is too low, the air will drop before reaching the end of the car and if too high it will discharge against the end bulkhead and be reflected back. Care must be exercised to secure proper circulation, otherwise objectionable drafts will be experienced.

The recirculating air grilles are usually of the straight flow type, and should be located so that objectionable drafts will not be created by the return air. The outside air intakes, located in the car vestibule, on the side of the car, or on the roof of the car, depending upon the location of the cooling coils, should be of ample size to permit the entrance of sufficient outside air. On many of the recently air-conditioned cars, there are no dampers or shutters at the outside air intakes, the percentage of outside air being controlled by blocking the flow through the recirculating grille.

Air Cleaning

All of the air circulated by the blower is filtered before passing over the cooling coils. In some cars the outside and recirculated air are filtered separately before mixing, while on others the air from the two sources is mixed before passing through a common filter. Filters in use are made of metal, wool, cloth, spun glass, hemp, paper, hair, and wire screen. Most filters have a viscous coating of oil for greater cleaning efficiency. Some types may be cleaned, retreated, and returned to service while other types are discarded when dirty.

STEAM OR VAPOR HEATING EQUIPMENT

The majority of cars in service are heated by circulating low pressure steam or vapor through pipes located along the side walls near the floor. When an air conditioning unit, using air from the outside, is installed it is necessary to provide a heating coil to warm the air during cold weather. Usually from 30 to 40 per cent of the heat required is supplied from the air conditioning unit and the balance from the floor heating system. It is necessary to have sufficient floor radiation to keep water lines in the car from freezing while standing in the yard with the air conditioning unit shut off. In new and some rebuilt cars, finned pipes are used for the floor heat to provide greater radiation surface. A few cars have the heating pipes enclosed in a duct, through which air is forced by a fan, the warmed air discharging through numerous openings along the floor. The amount of heat required depends upon the type and construction of the car, especially the amount and kind of insulation, outside temperature, wind velocity and direction, train speed, number of passengers, and inside temperature desired. In severe weather, with temperatures from -10 to -20 F, an average of approximately 200 lb of steam per car per hour is required. Pullmans require approximately 250 lb per hour, coaches 150 to 175 lb per hour and baggage cars 150 lb per hour.

COOLING EQUIPMENT

Three general types of cooling or refrigerating equipment are being used with satisfactory results. These are the ice-activated, the steam-ejector and the mechanical compression systems. These systems when arranged for car use, function the same as in stationary service, but must be more compact and lighter in weight. See Chapter 23 for description of the general principles of the various systems.

The mechanical compression systems are divided into three general classes depending upon the type of drive for the compressor, namely, the electro mechanical, the direct mechanical, and the internal combustion engine mechanical. The compressor of an electro mechanical compression system is driven by an electric motor, the power for which is supplied by a generator and a storage battery. The generator is driven from the car axle by a gear, belt, or other type of mechanical drive. The compressor of a direct mechanical compression system is driven directly from the car axle by means of a mechanical drive, the speed of the compressor being regulated by an electric speed control which permits slippage at high train speeds. The compressor of the third type of mechanical compression

system is driven by an internal combustion engine operating on propane. Sufficient fuel for several days' operation is carried in drums mounted in a rack under the car.

The refrigerant frequently used in the mechanical compression systems is dichlorodifluoromethane. The condensers are cooled by blowing a large quantity of outside air over the dry condenser coils, or over the coils wet by a spray of water to obtain the benefits of evaporative cooling. The latter method gives lower discharge pressures which is a distinct advantage when operating at high outside temperatures. A device gaining in popularity is a liquid subcooler by means of which the liquid refrigerant is subcooled by evaporative cooling, producing more available refrigeration at the aii conditioning unit.

Another type of system which has been tried for railway passenger car air conditioning uses dry ice as the refrigerant, the equipment being essentially the same as for the water ice-activated system. Until an adequate supply of dry ice can be assured at a stable and reasonable price, this system will not be a serious competitor to the other three types now in use.

The capacity required in the refrigerating system depends upon a number of factors such as size and type of construction of car, thickness and kind of insulation used, the amount of heat produced within the car by motors, lights, and other appliances, the amount of outside air, the intensity of solar radiation, the number of occupants, and the inside temperature desired. The sun load on a bright day is about 1.2 tons. For average cars on sunny days, with high outside temperatures and humidities, from 65,000 to 80,000 Btu per hour will have to be removed from the interior of the car to maintain an inside effective temperature within the comfort zone. This means that a refrigerating capacity of from 5.5 to 7.0 tons will be required.

HUMIDITY CONTROL

The temperature to be maintained in a car depends upon the outside temperature and the humidity desired inside the car. With a low humidity it is necessary to maintain a higher temperature to establish a desirable comfort condition. Little humidity control has been attempted on cars up to the present time. A certain degree of automatic humidity control is secured with cooling, but the relative humidity obtained depends largely on the temperature of the evaporator, which should be below the dew-point temperature of the air. With certain outside atmospheric conditions it may not be possible to operate the conventional equipment with a sufficiently low evaporator temperature to reduce the humidity without dropping the temperature too low. One method has been developed whereby the evaporator temperature is carried below the dew-point a sufficient amount to insure dehumidification and then the cold air is heated to the proper temperature by passing it over coils through which part of the high temperature liquid from the condenser is by-passed. Such a system is costly and has not been generally applied.

During the heating season humidification is desirable from a comfort standpoint, but unless properly controlled, condensation will appear on

the windows. A steam or water spray controlled by a humidistat will provide the necessary moisture for humidification. There are several cars with this feature now in use.

TEMPERATURE CONTROL

The control of the air conditioning equipment should be simple and automatic in order to eliminate the human element for the selection of the control point. The use of a centralized panel for all switches, fuses, relay, etc., will simplify the installation and operation. Generally, separate thermostats are used for heating and cooling control. The best location for the thermostats depends upon the car layout and method of air distribution and can best be determined for any particular type of car and equipment by careful consideration of the several factors involved. The floor heat thermostats are usually located near the floor. The overhead heat and cooling thermostats are placed in the upper part of the car, sometimes in the air ducts or at the recirculating grille. All thermostats should be located so that the air can circulate freely around them. Maintenance of uniform comfort conditions for cooling, floor and overhead heating, has been satisfactory with provisions for a high, a medium, and a low thermostat setting and in some cases two settings have been satisfactory for cooling. In many cars the following points have been found to be satisfactory: 72, 74, and 76 F for cooling, and 60, 71 and 74 F for floor and overhead heating.

A few cars are in operation in which the inside temperature is varied dependent upon the outside temperature in order to prevent too high a differential between inside and outside temperatures. The maximum inside dry-bulb temperature permitted by these controls is usually 80F.

The heating and refrigerating equipment should be interlocked so that they cannot both operate at the same time. While heating, the control should be so arranged that in case of a steam failure the blower fan will stop or the outside air intake should be closed to prevent cold outside air from being introduced into the conditioned space.

POWER SUPPLY

One of the most important problems to be solved in connection with railway car air conditioning is that of power supply. The majority of non air-conditioned cars now in service are electrically lighted and equipped with fans. Power is furnished by storage batteries and axle generators of from 2 to 5 kw capacity.

Electric Power Requirements

When air conditioning is installed the electrical load is increased, according to the type of system as indicated in Table 1. To furnish this additional electric power, the capacity of the axle generators must be increased to 4 to 20 kw, and the storage battery capacity increased, in addition to that required by the car lighting system by 400 to 700 amp-hr for the electro-mechanical system, by 150 to 300 amp-hr for the steam-ejector and direct drive mechanical systems, and by 50 to 200 amp-hr for the ice-activated and internal combustion engine mechanical systems.

TABLE 1. ELECTRIC POWER REQUIRED TO OPERATE SYSTEM

System	Kilowatts
Electro-Mechanical	10 50
Direct Drive Mechanical	1 00
Internal Combustion Engine Mechanical	1 25
Steam-Ejector	3 35
Ice-Activated	1 20

Total Power Requirements

In addition to the electric power requirements, for continuous operation at average temperatures, the direct drive mechanical system requires 10.24 hp from the car axle and the steam-ejector system requires 230 lb steam per hour from the locomotive boiler for a 6-ton unit. The ice-activated system requires 463 lb of ice per hour and the internal combustion engine drive mechanical requires 7.3 lb propane per hour. This power, with the exception of the ice and propane, as well as the power required to move the extra weight of the equipment and the power required to overcome the axle bearing friction, must be supplied by the locomotive en route, and if a number of cars in the train are air conditioned, the effect on train performance should not be overlooked. The demand for power for cooling comes, however, at the time of the year when steam for heating is not required, and the demand for lighting is at a minimum.

The total power required by the air conditioning systems will vary with the speed of train operation because of the effect of speed upon the drive efficiency and upon the resistance due to the added weight of the equipment. Fig. 1 shows the effect of speed upon the efficiency of the direct drive used with the direct mechanical system, and upon the average efficiency of four mechanical drives and generators used for electric power generation. The total increase in weight of passenger cars because of air conditioning is approximately 9,600 lb for the electro-mechanical, 8,600 lb for the direct mechanical, 8,600 lb for the internal combustion engine drive mechanical, 11,300 lb for the steam, and 8,500 lb for the ice-activated system.

The average refrigeration load has been found to be 3.3 tons, and the average capacity of air conditioning systems is about 5.92 tons. The relation of load to capacity, $3.3 \div 5.92 = 0.56$ or 56 per cent, is that percentage of the time during the cooling season that the cooling equipment will be in operation. The average drawbar horsepower demand upon a locomotive, accordingly, consists of 56 per cent of the horsepower required for continuous operation and 44 per cent of the horsepower required for non-operation. Table 2 shows the drawbar horsepower that must be supplied by the locomotive for each air-conditioned car for continuous operation of the air conditioning system, for non-operation of the equipment, and for an average condition when the air conditioning equipment is operating continuously 56 per cent of the time and is not operating 44 per cent of the time. It is important not to overlook the horsepower demand on the locomotive when the air conditioning equipment is not operating, which includes the horsepower required to operate

the blower fan, to haul the weight of the equipment, to overcome drive and generator friction, and to replace the losses occasioned by the removal of current from the storage battery.

Fig. 2 shows the tractive resistance of a 75-ton passenger car with six wheel trucks without an axle generator, with a 4 kw generator load, and, for the same car with an increase in weight of 5 tons and a 20 kw axle

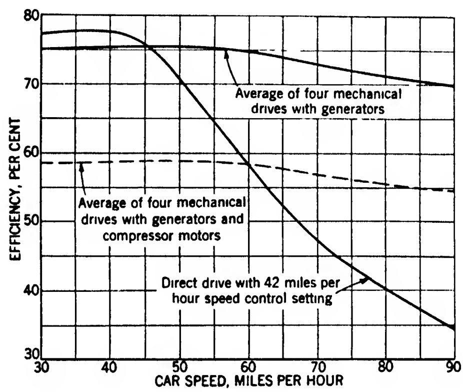

FIG. 1. EFFICIENCIES OF DRIVE MECHANISMS FOR RAILWAY AIR CONDITIONING SYSTEMS

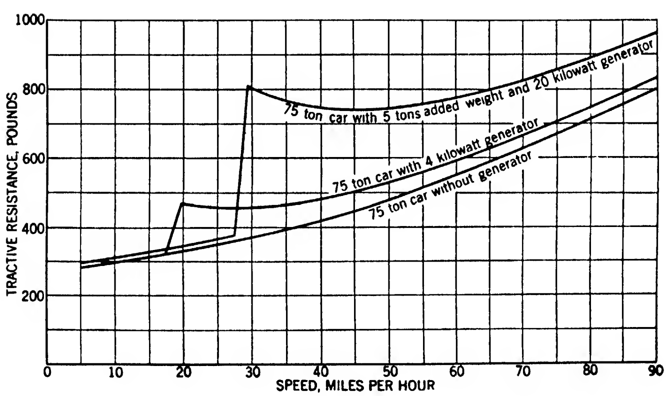

FIG. 2. TRACTIVE RESISTANCE OF 75 TON PASSENGER CAR WITH SIX WHEEL TRUCKS

generator load. The curve with the 4 kw generator is representative of a car before air conditioning, and the curve with the 20 kw generator and 5 tons added weight is representative of a car after air conditioning. At 50 mph, the tractive resistances of these two cars are 520 lb and 745 lb respectively, or a difference of 225 lb. Then: $\dfrac{225 \times 5280 \times 50}{60 \times 33000} = 29.7$ hp is required due to a 16 kw load and 5 tons added weight. Ten cars with a similar load would require 297 horsepower or roughly 10 per cent of the capacity of a 3,000 hp passenger locomotive.

Consideration must also be given to the power requirements for refrigeration while the car is standing or running at slow speeds. The electrical energy required for the ice-activated, steam, and internal combustion engine drive mechanical systems is easily supplied from the storage battery. Steam for the steam system can be supplied from the locomotive or from a stationary plant. The majority of the electro-mechanical systems are equipped with A. C.—D. C. motors. While standing in the yards and stations the A. C. motor is connected to a 220-volt, 3-phase circuit. The majority of these equipments are so ar-

TABLE 2. LOCOMOTIVE POWER DEMANDS FOR DIFFERENT AIR CONDITIONING SYSTEMS

System	Drawbar Horsepower per Car Required at Train Speeds			
	30 MPH	50 MPH	70 MPH	90 MPH
For Continuous Operation				
Electro-Mechanical	20 9	22 2	25 4	31 6
Direct Mechanical	16 8	19 5	29 5	42 6
Internal Combustion Engine Mechanical	3 4	4 6	7 0	11 9
Steam-Ejector[a]	7 6	9 3	12 5	19 0
Ice-Activated	3 2	4 5	6 8	11 6
For Non-Operation				
Electro-Mechanical	5 0	6 4	9 0	14 4
Direct Mechanical	5 2	6 3	8 7	13 5
Internal Combustion Engine Mechanical	2 2	3 5	5 8	10 6
Steam-Ejector	3 7	5 4	8 4	14 8
Ice-Activated	1 9	3 2	5 5	10 2
For Average Condition of 56 Per Cent Continuous Operation and 44 Per Cent Non-Operation				
Electro-Mechanical	13 9	15 2	18 2	24 0
Direct Mechanical	11 7	13 7	20 3	29 6
Internal Combustion Engine Mechanical	2 8	4 1	6 5	11 4
Steam-Ejector[a]	5 9	7 6	10 7	17 5
Ice-Activated	2 6	3 9	6 2	11 0

[a]In addition, steam is required from the locomotive to the extent of 230 lb per hour during the time the equipment is in operation

ranged that, while operating on A. C. power, the D. C. motor may be used as a generator for battery charging. If an auxiliary circuit is not available the D. C. compressor motor may be operated from the storage battery for short periods of time. The direct drive mechanical compression systems are, also, equipped with A. C. motors for operation from auxiliary circuits. As these equipments can only be operated when connected to the auxiliary circuit or while the train is running above the cut in speed of the drive, many cars are equipped with an auxiliary hold-over system by which reserve cooling is available. Due to the characteristics of the direct drive, the air conditioning system operates at reduced capacity when the car is moving at speeds below 42 mph.

COST OF RAILWAY AIR CONDITIONING

The cost of railway air conditioning is usually expressed in terms of 1000 car-miles. The actual costs for the different systems, however, are dependent upon a number of variables. Based upon a survey of the most prominent railroads in air conditioning, and upon extensive tests, the values given in Table 3 are indicative of the present costs of air conditioning to the railroads.

TABLE 3 AIR CONDITIONING COSTS FOR RAILWAY AIR CONDITIONING

SYSTEM	GROSS INSTALLATION COST	COSTS PER 1000 CAR-MILES[a]			
		Fixed Charges	Maintenance Cost	Operation Cost	Total
Electro-Mechanical	$6,484 00	$ 8 65	$3 33	$0 99	$12 97
Direct Mechanical	8,515 00	11 35	2 33	0 93	14 61
Internal Combustion Engine Mechanical	5,750 00	7 67	3 30	1 99	12 96
Steam-Ejector	8,475 00	11 30	2 15	1 02	14 47
Ice-Activated	3,982 00	5 31	0 97	5 29	11 57

[a] For an average cooling season of 5 months, an average train speed of 50 mph and an average car mileage of 150,000 miles per year

Gross Installation Cost

The gross installation cost, from which the fixed charges are derived, may be amortized on this basis:

1 Depreciation, at the rate of 12 5 per cent
2 Interest, at the rate of 6 per cent.
3 Taxes and insurance at the rate of 1 5 per cent.

The fixed charges per 1000 car-miles for any type of system are:

$$FC = \frac{1000}{m} (0\ 20A) \tag{1}$$

where

FC = fixed charges, dollars per 1000 car-miles.
A = gross installation cost, dollars.
m = total number of car-miles traveled in one year

Maintenance Cost

The average maintenance cost is based upon the experience of the railroads in maintaining several hundred air conditioning units. The maintenance cost per 1000 car-miles is.

$$MC = \frac{1000\,B}{m} \tag{2}$$

where

MC = maintenance cost, dollars per 1000 car-miles
B = total annual maintenance cost, dollars
m = total number of car-miles traveled in one year.

Operation Cost

The cost of operation is influenced by:

1. Speed of train operation.
2. Average drawbar horsepower required to operate air conditioning system.
3. Length of the cooling season.
4. Cost of power produced by the locomotive at $0.00493 per horsepower-hour
5. Proportion of time the cooling equipment is operated during the cooling season considered as 56 per cent.
6. Cost of additional necessities as: *a.* Ice in bunker, at $4.42 per ton, *b.* Propane on the car, at $0.039 per pound, *c* Steam at $0.021 per 100 pound.

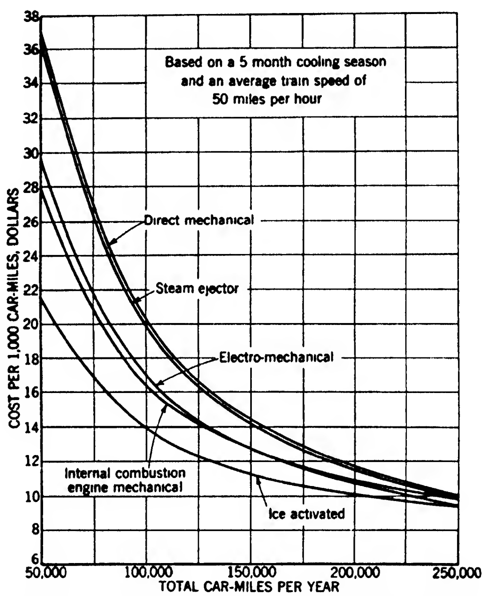

FIG. 3. COMPARATIVE TOTAL COSTS FOR RAILWAY PASSENGER CARS

The operation cost in dollars per 1000 car-miles is:

$$OC = \frac{1000\,K}{12\,S}\,(D \times E + 0.56\,F \times G) + \frac{1000\,(12 - K)}{12\,S}\,(H \times E) \qquad (3)$$

where

OC = operation cost, dollars per 1000 car-miles.
D = average drawbar horsepower demand on a locomotive at speed S.
E = cost per horsepower-hour, dollars.
F = additional necessities such as ice, steam or propane, pounds per hour.
G = cost of additional necessities, dollars per pound.
H = drawbar horsepower required when system is not operating.
S = speed of train operation, miles per hour.
K = length of cooling season, months.
0.56 = Proportion of operation time to total time during the cooling season.

Total Cost of Air Conditioning

When the fixed charges, maintenance cost, and operation cost are each expressed in terms of 1000 car-miles, addition of the three elements will give the total cost of air conditioning on that basis.

Comparisons of the total cost per 1000 car-miles for the five methods of air conditioning are shown in Fig. 3, representing costs for an average condition, namely a cooling season of five months and an average speed of 50 mph.

COOLING LOAD CALCULATIONS

The calculated heat gain for a railway passenger car is dependent on several variables which may be determined from the basic data given in Chapters 5, 6 and 8.

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PROBLEMS IN PRACTICE

1 ● What item is the greatest among the cooling loads figured in the design of a summer air conditioning system for a passenger car?

The heat from passengers.

2 ● To what extent does bright sunshine increase the cooling requirements of a passenger car?

About 1.2 tons of refrigeration.

3 ● What is the total refrigerating capacity generally required in a passenger car?

5.5 to 7 tons per car.

4 ● What is the effect of train speed upon the cooling requirements of a car?

Requirements are slightly increased because of increased heat transmission.

5 ● What is the fan capacity of the air conditioning unit in the average car?

2000 to 2500 cfm.

6 ● When is it economical to take all air for car cooling from outdoors?

When the outdoor wet-bulb temperature is lower than that in the car.

7 ● What various arrangements are used for distributing cooled air into cars?

Bulkhead delivery at center or ends of car, center duct, and side duct on one or both sides.

8 ● What types of cooling systems are used?

Ice-activated, steam-ejector, and mechanical compression systems.

9 ● What cooling medium is used for condensing the refrigerant in a railroad air conditioning system?

Outdoor air, sometimes with the aid of evaporative cooling.

10 ● How may adequate cooling of condensers be provided in hot desert regions?

By evaporative cooling with water sprays.

11 ● How is the temperature controlled in railroad cooling systems?

By intermittent operation of the compressor, the steam jet or the ice water circulating pump

12 ● How much steam is required for car heating on the coldest days?

Pullmans 250 lb per hour, coaches 150 to 175 lb per hour, baggage cars 150 lb per hour.

13 ● At present costs which is likely to be the greater, fixed charges or operating costs?

The fixed charges for steam and mechanical compression systems, and operating costs for ice systems

14 ● What would be the annual operating cost for a car equipped with an ice-activated system under the following conditions: a total mileage of 150,000 car-miles per year, a cooling season of 5 months, and an average train speed of 50 mph?

$1,705.00.

Chapter 33

INDUSTRIAL AIR CONDITIONING

Atmospheric Conditions Required, General Requirements, Classification of Problems, Control of Regain, Moisture Content and Regain, Conditioning and Drying, Control of Rate of Chemical Reaction, Control of Rate of Biochemical Reactions, Control Rate of Crystallization

IN the application of air conditioning to industrial processes, too much stress cannot be laid upon a thorough understanding by the air conditioning engineer of the problems involved. A complete knowledge of these problems is necessary before a satisfactory design can be made. Individual processes and machines are changing rapidly and air conditions must be constantly revised to meet the new conditions.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity during processing depends upon the product and the nature of the process. As far as the behavior of the material itself and its desired final condition are concerned, each material and process presents a different problem. The best relative humidity may range up to 100 per cent. Similarly the most desirable temperature may range between wide limits for different materials and treatments. Extremes in either relative humidity or temperature require relatively expensive equipment for maintaining these conditions automatically. In departments where people are working, their health, comfort, and productive efficiency must be considered and often a compromise between the optimum conditions for processing and those required for the comfort of the worker is desirable.

It is generally considered that relative humidities below 40 per cent are on the dry side, conducive to low regains, a brittle condition of fibrous materials, prevalence of static electricity, and a tendency toward dryness of the skin and membranes of human beings. At the other end of the scale, humidities above 80 per cent are relatively damp, conducive to high regains, extreme softness, and pliability.

Table 1 lists desirable temperatures and humidities for industrial processing. In using this table, care must be taken in qualifying the process. In preparing many materials, conditions are not maintained constantly, but different temperatures and humidities are held for varying lengths of time.

Table 1. Desirable Temperatures and Humidities for Industrial Processing

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
Automobile	Assembly line	65	40
Baking	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	80 to 90	80 to 95
	Storage of flour	70 to 80	60
	Storage of yeast	28 to 40	60 to 75
Biological Products	Vaccines	below 32	
	Antitoxins	38 to 42	
Brewing	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
Ceramic	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60	35
Chemical	General storage	60 to 80	35 to 50
Confectionery	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
Distillery	General manufacture	60	45
	Storage of grains	60	30 to 45
Drug	Storage of powders and tablets	70 to 80	30 to 35
Electrical	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
Food	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats	0 to 10	50
	Storage of sugar	80	35
Fur	Drying of furs	110	
	Storage of furs	28 to 40	25 to 40

TABLE 1. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING (Concluded)

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
INCUBATORS	Chicken	99 to 102	55 to 75
LABORATORY	General analytical and physical	60 to 70	60 to 70
	Storage of materials	60 to 70	35 to 50
LEATHER	Drying of hides	90	
LIBRARY	Book storage (see discussion in this chapter)	65 to 70	38 to 50
LINOLEUM	Printing	80	40
MATCHES	Manufacturing	72 to 74	50
	Storage of matches	60	
MUNITIONS	Fuse loading	70	55
PAINT	Air drying lacquers	70 to 90	25 to 50
	Baking lacquers	180 to 300	
	Air drying of oil paints	60 to 90	25 to 50
PAPER	Binding, cutting, drying, folding, gluing	60 to 80	25 to 50
	Storage of paper	60 to 80	35 to 45
PHOTOGRAPHIC	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
PRINTING	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic)	60 to 75	20 to 60
	Storage of rollers	60 to 80	35 to 45
RUBBER	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
SOAP	Drying	110	70
TEXTILE	Cotton— carding	75 to 80	50
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	60 to 70
	weaving	68 to 75	70 to 80
	Rayon— spinning	70	85
	twisting	70	65
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
TOBACCO	Cigar and cigarette making	70 to 75	55 to 65
	Softening	90	85
	Stemming or stripping	75 to 85	70

GENERAL REQUIREMENTS

In general, air conditioning apparatus for industrial purposes must be capable of absorbing heat from various sources such as machinery power, electric lights, people, sunlight and chemical reaction; of warming or cooling to any desired temperature, and of providing ample air supply at all times. Refrigeration may or may not be required, depending upon natural conditions, the required relative humidity and the maximum permissible temperature. Washing, purifying and recirculating of the air may be desirable. Good distribution is essential for the control of air motion and for the prevention of uneven conditions. Accurate, sensitive and reliable automatic control of humidity or temperature, or both, is vital in most cases.

Ordinarily, outside weather conditions and the ventilation required for workers are of secondary importance in relation to the total work to be done by the air conditioning system. In extreme cases of high concentration of industrial heat from machinery and ovens the error of entirely omitting the heat gain through the building structure would not be serious. At the other extreme, where low temperatures must be produced with refrigeration and where comparatively little power is used for driving the machinery, the heat gain through the building structure will become the major factor in determining the size of equipment and in this case the ventilation requirement assumes a normal degree of importance.

Buildings which are to be air conditioned should therefore be designed with careful consideration of over-all cost and efficiency. Condensation resulting from high humidities must be prevented by suitable materials and construction, or else collected and drained to prevent loss of product or quick deterioration of the structure. Air leakage or filtration may add greatly to operating costs or make the maintenance of low humidities (relative or absolute) wholly impossible. Low temperatures require good insulation.

It is apparent that the subject of air conditioning for industrial processes is extensive and greatly involved, and that a detailed treatment is therefore beyond the scope of this book. A few of the salient points of the general subject are covered in this chapter.

CLASSIFICATION OF PROBLEMS

In general, any industrial air conditioning problem may be listed under one or more of the following four classes:

1 Control of Regain
2 Control of Rate of Chemical Reactions.
3 Control of Rate of Biochemical Reactions
4 Control of Rate of Crystallization

CONTROL OF REGAIN

In the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and foodstuffs, the temperature and relative humidity of the air have a marked influence upon the rate of production and upon the weight, strength, appearance and general

quality of the product. This influence is due to the fact that the moisture content of materials having a vegetable or animal origin, and to a lesser extent minerals in certain forms, come to equilibrium with the moisture of the surrounding air.

In industries where the physical properties of a product affect its value, the percentage of moisture is of special importance. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Standards of regain are firmly fixed in trade with fair penalties for excesses. Deficiencies result in loss of revenue to seller and loss of desirable quality to buyer.

Manufacturing economy therefore requires that the moisture content be maintained at a percentage favorable to rapid and satisfactory manipulation and to a minimum loss of material through breakage. A uniform condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material and defective product.

In the processing of hygroscopic materials, it is usually necessary to secure a final moisture content suitable for the goods as shipped. Where the goods are sold by weight, it is proper that they contain a normal or standard moisture content.

MOISTURE CONTENT AND REGAIN

The terms *moisture content* and *regain* refer to the amount of moisture in hygroscopic materials. *Moisture content* is the more general term and refers either to free moisture (as in a sponge) or to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 grains is dried to a constant weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

The use of the term *regain* does not imply that the material as a whole has been completely dried out and has re-absorbed moisture. During the processing of certain textiles, for instance, complete drying during manufacturing is avoided as it might appreciably reduce the ability of the material to re-absorb moisture. A basis for calculating the regain of textiles is obtained by drying under standard conditions a sample from the lot and the dry weight thus obtained is used as a basis in the calculations to determine the regain.

The moisture content of an hygroscopic material at any time depends upon the nature of the material and upon the temperature and especially the relative humidity of the air to which it has been exposed. Not only do different materials acquire different percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying out varies with the nature of the material, its thickness and density.

TABLE 2. REGAIN OF HYGROSCOPIC MATERIALS

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

CLASSIFICATION	MATERIAL	DESCRIPTION	RELATIVE HUMIDITY—PER CENT 10	20	30	40	50	60	70	80	90	AUTHORITY
Natural Textile Fibres	Cotton	Sea island—roving	2.5	3.7	4 6	5 5	6 6	7 9	9.5	11.5	14 1	Hartshorne
	Cotton	American—cloth	2 6	3 7	4 4	5 2	5 9	6 8	8 1	10 0	14 3	Schloesing
	Cotton	Absorbent	4 8	9 0	12 5	15 7	18 5	20 8	22 8	24 3	25 8	Fuwa
	Wool	Australian merino—skein	4 7	7 0	8 9	10 8	12 8	14 9	17 2	19 9	23 4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5 5	6 9	8 0	8 9	10 2	11 9	14.3	18 8	Schloesing
	Linen	Table cloth	1 9	2 9	3 6	4 3	5 1	6 1	7 0	8 4	10 2	Atkinson
	Linen	Dry spun—yarn	3 6	5 4	6 5	7 3	8 1	8 9	9 8	11 2	13 8	Sommer
	Jute	Average of several grades	3 1	5 2	6 9	8 5	10 2	12 2	14 4	17 1	20 2	Storch
	Hemp	Manila and sisal—rope	2 7	4 7	6 0	7 2	8 5	9 9	11.6	13 6	15 7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average skein	4 0	5 7	6 8	7 9	9 2	10 8	12 4	14 2	16 0	Robertson
	Cellulose Acetate	Fibre	0 8	1 1	1 4	1 9	2 4	3 0	3 6	4 3	5 3	Robertson
Paper	M F Newsprint	Wood pulp—24% ash	2 1	3 2	4 0	4 7	5 3	6 1	7 2	8 7	10 6	U S B of S
	H M F Writing	Wood pulp—3% ash	3 0	4 2	5 2	6 2	7 2	8 3	9 9	11 9	14 2	U S B of S
	White Bond	Rag—1% ash	2 4	3 7	4 7	5 5	6 5	7 5	8 8	10 8	13 2	U S B of S
	Com Ledger	75% rag—1% ash	3 2	4 2	5 0	5 6	6 2	6 9	8 1	10 3	13 9	U S B of S
	Kraft Wrapping	Coniferous	3 2	4 6	5 7	6 6	7 6	8 9	10 5	12 6	14 9	U. S B of S
Misc. Organic Materials	Leather	Sole oak—tanned	5 0	8 5	11 2	13 6	16 0	18 3	20 6	24 0	29 2	Phelps
	Catgut	Racquet strings	4 6	7 2	8 6	10 2	12 0	14 3	17 3	19 8	21 7	Fuwa
	Glue	Hide	3 4	4 8	5 8	6 6	7 6	9 0	10 7	11 8	12 5	Fuwa
	Rubber	Solid tire	0 11	0 21	0 32	0 44	0 54	0 66	0 76	0 88	0 99	Fuwa
	Wood	Timber (average)	3 0	4 4	5 9	7 6	9 3	11 3	14 0	17 5	22 0	Forest P Lab
	Soap	White	1 9	3 8	5 7	7 6	10 0	12 9	16 1	19 8	23 8	Fuwa
	Tobacco	Cigarette	5 4	8 6	11 0	13 3	16 0	19 5	25 0	33 5	50 0	Ford
Foodstuffs	White Bread		0 5	1 7	3 1	4 5	6 2	8 5	11 1	14 5	19 0	Atkinson
	Crackers		2 1	2 8	3 3	3 9	5 0	6 5	8 3	10 9	14 9	Atkinson
	Macaroni		5 1	7 4	8 8	10 2	11 7	13.7	16 2	19 0	22 1	Atkinson
	Flour		2 6	4 1	5.3	6 5	8 0	9 9	12 4	15 4	19 1	Bailey
	Starch		2 2	3 8	5 2	6 4	7 4	8 3	9 2	10 6	12 7	Atkinson
	Gelatin		0 7	1 6	2 8	3 8	4 9	6 1	7 6	9 3	11 4	Atkinson
Misc. Inorganic Materials	Asbestos Fibre	Finely divided	0 16	0 24	0 26	0 32	0 41	0 51	0 62	0 73	0 84	Fuwa
	Silica Gel		5.7	9.8	12 7	15 2	17.2	18 8	20 2	21 5	22 6	Fuwa
	Domestic Coke		0 20	0 40	0 61	0.81	1.03	1 24	1 46	1 67	1 89	Selvig
	Activated Charcoal	Steam activated	7 1	14.3	22 8	26 2	28 3	29 2	30 0	31 1	32 7	Fuwa
	Sulphuric Acid	H_2SO_4	33.0	41 0	47 5	52 5	57 0	61.5	67 0	73 5	82 5	Mason

Table 2 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. In the case of cotton, for instance, an increase in temperature of 10 deg has the same effect on regain as a decrease in relative humidity of one per cent. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary.

The regain or moisture content affects the physical properties of textiles to a marked degree, changing the strength, pliability and elasticity.

The fact that the regain of textiles will come into equilibrium with the conditions of the surrounding air and vary with its temperature and relative humidity is the fundamental basis for the control of physical qualities during manufacture. During the preparation processes in a cotton mill, the cotton fibers should be in a condition to be easily carded.

These preliminary processes are carried out best in a relative humidity of 50 to 55 per cent. As the cotton fiber comes to the spinning operation, more flexibility is needed and the relative humidity is increased in this department. For many years, 65 per cent relative humidity was considered the optimum. To offset the extra work performed on the fiber as the spindle speed is increased, many cotton mills now carry 70 per cent relative humidity in the spinning rooms.[1] Winding, warping and weaving are all processes calling for great flexibility and a consequent need for higher humidity.

Other textile fibers, due to their different natural characteristics, are processed under relative humidities and temperatures applicable to each.

Rayons, on account of great loss of strength with the higher regains, should be processed in a relative humidity of 57 per cent. Acetate silk, another chemical fiber, with approximately 50 per cent of the regain of rayon, may be processed between 60 and 65 per cent relative humidity.

All hygroscopic materials release sensible heat equivalent to the latent heat of the moisture absorbed by the material, all of which may account for a large percentage of the total heat load.

CONDITIONING AND DRYING

In general, the exposure of materials to desirable conditions for treatment may be coincidental with the manufacture or processing of the materials, or they may be treated separately in special enclosures. This latter treatment may be classified as conditioning or drying. The purpose of conditioning or drying is usually to establish a desired condition of moisture content and to regulate the physical properties of the material.

When the final moisture content is lower than the initial one, the term *drying* is applied. If the final moisture content is to be higher, the process is termed *conditioning*. In the case of some textile products and tobacco,

[1]The Present Status of Textile Regain Data, by A E Stacey, Jr. (*National Association of Cotton Manufacturers*, 1927)

for example, drying and conditioning may be combined in one process for the dual purpose of removing undesirable moisture and accurately regulating the final moisture content. Either conditioning or drying are frequently made continuous processes in which the material is conveyed through an elongated compartment by suitable means and subjected to controlled atmospheric conditions.

CONTROL OF RATE OF CHEMICAL REACTIONS

A typical example of the second general classification, that is, the control of the rate of chemical reactions, occurs in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that during this process close control of both temperature and relative humidity should be maintained. Temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the surface of the solution and gives a solution of known strength throughout the mercerizing period.

Another well-known example of this class is the *drying* of varnish which is an oxidizing process dependent upon temperature. High relative humidities have a retarding action on the rate of oxidization at the surface and allow the gases to escape as the chemical oxidizers *cure* the varnish film from the bottom. This produces a surface free from bubbles and a film homogeneous throughout.

Desirable temperatures for *drying* varnish vary with the quality. A relative humidity of 65 per cent is beneficial for obtaining the best processing results.

CONTROL OF RATE OF BIOCHEMICAL REACTIONS

In the field of biochemical control, industrial air conditioning has been applied to many different and well-known products. All problems involving fermentation are classed under this heading. As biochemistry is a subdivision of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the dough room of the modern bakery. Yeast develops best at a temperature of 80 F. A relative humidity of 65 per cent is maintained so as to hold the surface of the dough open to allow the CO_2 gases formed by the fermentation to pass through and produce a loaf of bread, when baked, of even, fine texture without large voids.

Another example of a similar process is found in the curing of macaroni. The flour and water mixture is fermented and dried. As it is necessary to have a definite amount of water present to carry on a fermentation process, the moisture must be removed in a relatively short period to stop fermentation and prevent souring and in such a manner as to avoid setting up internal strains in the mixture. Best results are obtained with the correct cycles of both temperature and humidity.

The curing of fruits, such as bananas and lemons, also come under this classification. Bananas are treated somewhat differently and to accomplish the required results, a cycle of temperatures and relative humidities is used. The starches in the pulp of the fruit must be changed and the

skin cured and colored, after which the fruit is cooled to maintain as slow a rate of metabolism as possible. Ideal conditions range between 55 to 57 F and in no case should the temperature go below 49 F, as the starches then become fixed and are indigestible.

The curing of lemons is an entirely different problem. Bananas are cured for a quick market, while lemons are held for a future market. The process, therefore, varies in the temperature used. Temperatures from 54 to 59 F have been found to be best suited for this process. A high relative humidity of 88 to 90 per cent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so it will be sufficiently tough to permit handling.

Tobacco from the field to the finished cigar, cigarette, plug or pipe tobacco, offers another interesting example of what may be done by industrial air conditioning in the control of color, texture and flavor. In the processing of tobacco, the first three classifications of air conditioning are involved, and only through close atmospheric control can the best quality of the leaf be developed.

CONTROL RATE OF CRYSTALLIZATION

The rate of cooling of a saturated solution determines the size of the crystals formed. Both temperature and relative humidity are of importance, as the one controls the rate of cooling, while the other, through evaporation, changes the density of the solution.

In the coating pans for pills, gum and nuts, a heavy sugar solution is added to the tumbling mass. As the water evaporates, each separate piece is covered with crystals of sugar. A smooth, opaque coating is only accomplished by blowing into the kettle the proper amount of air at the right temperature and relative humidity. If the cooling and drying is too slow, the coating will be rough and semi-translucent, and the appearance unsightly; if too fast, the coating will chip through to the interior. Only by balancing temperature, relative humidity, and volume of air to the sugar solution, can the proper rate be obtained and a perfect coating assured.

The foregoing is presented as typical of a few of the problems met with in applying air conditioning to various industrial processes. They are far from complete but with the help of a few natural laws may assist in solving others where similar basic principles are involved.

CALCULATIONS

The methods for determining the proper heating and cooling loads for the various industrial processes are similar to those outlined in Chapters 7 and 8. Because of the large number of motors and heat processing units usually prevalent in an industrial application, it is particularly important that operating allowances for the latent and sensible heat loads be definitely ascertained and used in the calculations to determine the total equivalent design load.

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PROBLEMS IN PRACTICE

1 ● Why is air conditioning required for some industrial processes?

To control the physical properties of the materials being processed.

Example. In the manufacture of chewing gum, it is rolled into slabs and scored The scored slab must then be broken at the score marks to form the sticks. If the slab is too warm, breaking is impossible, if the slab is too cold or too dry, it becomes brittle and shatters, thereby producing much material to be reworked.

2 ● Why is it necessary to control the physical properties of the material being processed?

To permit permanent adjustment of machinery.

Example. In the manufacture of cigarettes, the amount of tobacco fed upon the paper tape is determined by pressure against springs. When the tobacco is over-moist and, therefore, over-soft, a great excess will go into the finished cigarette; when the tobacco is too dry and, therefore, harsh, too little goes into the finished cigarette.

3 ● A condition of 75 F dry-bulb temperature and 55 per cent relative humidity is being maintained in a cigarette manufacturing department. What will be the regain and moisture content of the tobacco?

$$\text{The regain, from Table 2} = 17.75 \text{ per cent.}$$

$$\text{The moisture content} = \frac{17.75 \times 100}{100 + 17.75} = 15.1 \text{ per cent.}$$

4 ● A 1-lb sample taken from a 100-lb batch of material is found to have a bone-dry weight of 0.89 lb. This material is to be processed under atmospheric conditions which should produce a regain of 15 per cent. Compute the finished weight for each original 100-lb batch.

Let W equal the number of pounds of moisture in a finished batch.

$$\frac{W}{89} = \text{regain} = 15 \text{ per cent} = \frac{15}{100}$$

$$W = 13.35$$

$$89 + 13.35 = 102\ 35 \text{ lb finished weight}$$

5 ● A bundle of sea island cotton is found to have a bone-dry weight of 9.26 lb. What is the proper relative humidity at 75 F to produce a weight of 10 lb at equilibrium?

$$\text{Desired conditioned weight} = 10\ 00 \text{ lb}$$
$$\text{Bone-dry weight} = 9.26 \text{ lb}$$
$$\text{Weight of moisture required} = 0.74 \text{ lb}$$

$$\text{Regain} = \frac{0.74}{9.26} \times 100 = 7.9 \text{ per cent.}$$

From Table 2, the proper relative humidity required is 60 per cent.

6 ● Compute the bone-dry weight of 1000 lb of manila rope which has been stored for a considerable period of time in a conditioned room at 75 F dry-bulb temperature and 50 per cent relative humidity.

Assuming that this material has come to equilibrium under the atmospheric conditions given, Table 2 shows a regain of 8 5 per cent.

Let W equal the total weight of moisture in pounds.

$1000 - W$ = bone-dry weight in pounds

$$\frac{W}{1000 - W} = \text{regain} = 8.5 \text{ per cent} = \frac{8\ 5}{100}$$

$$W = 78.3 \text{ lb moisture}$$

$$1000 - 78.3 = 921.7 \text{ lb bone-dry weight.}$$

7 ● An egg evaporating plant wishes to dry 2000 lb of egg whites (85 per cent water) to crystalline form each 24 hours. The maximum permissible air delivery temperature in the drier is 140 F. What air volume will be required, assuming that outside air is at 95 F dry-bulb and 78 F wet-bulb and that air leaves the drier 70 per cent saturated?

Moisture to be removed = 2000 × 0.85 = 1700 lb. Using psychrometric chart and starting at the intersection of the vertical 95 F dry-bulb temperature line and the 46 per

cent humidity line, move horizontally to the right to the intersection with the 140 F vertical temperature line at 13 per cent relative humidity; then move along the constant heat (or wet-bulb line) to its intersection with the 70 per cent relative humidity curve and read 97.5 F dry-bulb, which will be the temperature of the air leaving the drier.

Moisture per cubic foot at 97.5 F and 70 per cent relative humidity	=	13.2 grains
Moisture per cubic foot at 95 F and 78 F wet-bulb	=	8.3 grains
Moisture added per cubic foot of air handled	=	4.9 grains

$$\frac{1700 \times 7000}{24 \times 60 \times 4.9} = 1685 \text{ cfm}$$

No allowance is made for heat lost in the transmission to and from the drier or for the heat required to raise the product from its entering temperature to that maintained in the drier. This would necessitate a trial and error solution common to all drying problems

Chapter 34

INDUSTRIAL EXHAUST SYSTEMS

Classification of Systems, Design Procedure, Requirements for Suction and Velocity, Hoods, Design of Duct Systems, Collectors, Resistance of Systems, Efficiency of Exhaust Systems, Selection of Fans and Motors, Corrosion

IN almost every industry some type of exhaust or collecting system is essential to achieve efficient and economical control of dusts and fumes. General design information is included in this chapter which is intended to relate primarily to factory exhaust systems.

CLASSIFICATION OF SYSTEMS

There are two general arrangements, the central and the group systems. In the central system a single or double fan is located near the center of the shop with a piping system radiating to the various machines to be served. In the group system, which is sometimes employed where the machines to be served are widely scattered, small individual exhaust fans are located at the center of the machine groups. The group arrangement has the advantage of flexibility.

Exhaust systems are also classified by the means employed to collect dust or other material handled. The dust or refuse may be collected and controlled by enclosing hoods, open hoods, inward air leakage, or by exhausting the general air of the room.

With some classes of machinery it is not feasible to closely hood the machines and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. This class includes such machines as rubber mills, package filling machinery, sand blast, crushers, forges, pickling tanks, melting furnaces, and the unloading points of various types of conveyors.

The open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator. When the hood must be placed at some distance above the machine it should be large enough to encompass an area of considerable extent as diffusion is usually quite rapid.

Consideration must also be given to the natural movement of the fumes. For those that are lighter than air the hood should be over or above the machine and where a heavy vapor or dust-laden air at ordinary temperature is to be removed, horizontal or floor connections are required. If it is attempted to remove heavy dust such as lead oxides by an overhead hood the conditions may be worse than if no exhaust were used at all, owing to the rising air current carrying the dust up through the

breathing zones. The objective to keep in mind in all cases is to take advantage of the natural tendency of the material to move upward or downward.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, the removal of some dust from the machine or enclosure being merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air leakage, thus preventing escape of the dust. While the exhaust system is required to handle only the air which leaks in through the crevices and openings in the enclosure, yet in many installations leakages are very high and great care is required to obtain satisfactory results with a system of this kind. The inward-leakage principle is utilized for controlling dust in the operating of tumbling barrels, grinding, screening, elevating, and similar processes.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space. The compartment or room in which the work is performed should be as small as is consistent with convenience in handling the work. The ventilating system should be designed so that a strong current of clean air is drawn across the operator, and away from him toward the work, where the dust is picked up and carried from the room.

DESIGN PROCEDURE

The first step in the design of an exhaust system is to determine the number and size of the hoods and their connections. No general rules, however, can be given since hood and duct dimensions are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available. Finally, in choosing the fan, the area of the intake should be equal to or greater than the sum of the areas of the branch ducts. The speed, of course, must be sufficient to maintain the estimated suction and air velocities in the system. In general, the most important requirements of an efficient exhaust and collecting system are as follows[1]:

1. Hoods, ducts, fans and collectors should be of adequate size.
2. The air velocities should be sufficient to control and convey the materials collected
3. The hoods and ducts should not interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption
5. When inflammable dusts and fumes are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall.
6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity.
7. The design of an exhaust system should afford easy access to parts for inspection and care.

[1]For more detailed requirements see Safe Practice Pamphlets Nos 32 and 37 published by the *National Safety Council*, Chicago

REQUIREMENTS FOR SUCTION AND VELOCITY

The removal of dust or waste by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry it to a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the

TABLE 1. SIZE OF CONNECTIONS FOR WOOD-WORKING MACHINERY

TYPE OF MACHINE	DIAMETER OF CONNECTIONS IN INCHES
Circular saws, 12-in. diam.	4
Circular saws, 12-24-in. diam.	5
Circular saws, 24-40-in. diam.	6
Band saws, blade under 2 in. wide	4
Band saws, blade 2-3 in. wide	5
Band saws, blade 3-4 in. wide	6
Band saws, blade 4-5 in. wide	7
Band saws, blade 5-6 in. wide	8
Small mortisers	6
Single end tenoners	6
Double end tenoners	7
Double end, double head tenoners	10
Planers, matchers, moulders, stickers, jointers, etc.—	
With knives, 6-10 in.	5–6
With knives, 10-20 in.	6–8
With knives, 20-30 in.	6–10
Shapers, light work	4–5
Shapers, heavy work	8
Belt sander, belt less than 6 in. wide	5
Belt sander, belt 6-10 in. wide	6
Belt sander, belt 10-14 in. wide	7
Drum sander, 24 in.	5
Drum sander, 30 in.	6
Drum sander, 36 in.	7
Drum sander, 48 in.	8
Drum sander, over 48 in.	10
Disc sander, 24 in. diam.	5
Disc sander, 26-36 in. diam.	6
Disc sander, 36-48 in. diam.	7
Arm sander	4

direction and speed with which it is thrown off. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood should be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible.

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. This is of considerable value where exhaust systems adapted to particular operations have been standardized by practice. Tables 1 and 2 present the duct sizes usually employed for standard wood-working machinery and for grinding and buffing wheels. Static pressures which in practice have been found necessary to control and convey various materials, are given in Table 3. It must be remembered, however, that the *suction* is merely a rough

TABLE 2. SIZE OF CONNECTIONS FOR GRINDING AND BUFFING WHEELS

DIAMETER OF WHEELS	MAX GRINDING SURFACE SQ IN.	MIN DIAM OF BRANCH PIPES IN INCHES
Grinding—		
6 in. or less, not over 1 in. thick	19	3
7 in. to 9 in., inclusive, not over 1½ in thick	43	3½
10 in. to 16 in., " " " 2 in "	101	4
17 in. to 19 in., " " " 3 in "	180	4½
20 in. to 24 in., " " " 4 in "	302	5
25 in. to 30 in., " " " 5 in "	472	6
Buffing—		
6 in or less, not over 1 in. thick	19	3½
7 in. to 12 in., inclusive, not over 1½ in thick	57	4
13 in. to 16 in., " " " 2 in "	101	4½
17 in. to 20 in., " " " 3 in "	189	5
21 in. to 27 in., " " " 4 in "	338	6
27 in. to 33 in., " " " 5 in. "	518	7

measure of the air volume handled and consequently of the air velocity at the opening of the hood. The elimination of any dusty condition requires added information concerning the shape, size and location of the hood used with regard to the operation in question.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. The static suction requirements, which range from 1½ to 5 in. water displacement in a *U*-tube, should be followed although in several instances they may appear to be excessive. Frequently, in these operations, a large part of the wheel must be exposed and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft. This tendency may be diminished by locating the connecting duct so as to create an air flow of not less than 200 fpm about the lower rim of the wheel.

Exact determinations of hood control velocities are not available, but it is safe to assume that for most dusty operations they should not be less

TABLE 3. SUCTION PRESSURES REQUIRED AT HOODS

TYPE OF INSTALLATION	STATIC SUCTION IN INCHES OF WATER
Exhausting from grinding and buffing wheels	1½–5
Exhausting from tumbling barrels	2
Exhausting from wood-working machinery—light duty	2
Exhausting from wood-working machinery—heavy duty	2–4
Shoe machinery exhaust	2–3
Exhausting from rubber manufacturing processes	2
Flint grinding exhaust	2
Exhausting from pottery processes	2
Lead dust and fume exhaust	2–4
Fur and felt machinery exhaust	2–3
Exhausting from textile machinery	2–3
Exhausting from elevating and crushing machinery	2
Conveying bulky and heavy materials	3–5

than 200 fpm at the point of origin. For granite dust generated by pneumatic devices, Hatch et al[2] give velocities from 150 to 200 fpm, depending on the type of hood used, as sufficient for safe control. Considering the character of the industry, air velocities of this order may be extended to similar dusty operations. The method for approximately determining these velocities in terms of the velocity at the hood opening is given below.

HOODS

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dusts and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many dust discharges may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, the following formula may be used to determine the air velocity at any point along the axis[3]:

$$V = \frac{0.1\,Q}{x^2 + 0.1\,A} \tag{1}$$

where

V = velocity at point, feet per minute
A = area of opening, square feet.
x = distance along axis, feet.
Q = volume of air handled, cubic feet per minute

Velocity Contours

It is possible by use of a specially constructed pitot-tube[4] to map contours of equal velocity in any axial plane located in the field of influence. It has been found that the positions of these contours for any hood can be expressed as percentages of the velocity at the hood opening and are purely functions of the shape of the hood[5].

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[2]Control of the Silicosis Hazard in the Hard Rock Industries I A Laboratory Study of the Design of Dust Control Systems for Use with Pneumatic Granite Cutting Tools, by Theodore Hatch, Philip Drinker and Sarah P Choate (*Journal of Industrial Hygiene*, Vol XII, No 3, March, 1930)

[3]The Control of Industrial Dust, by J M DallaValle (*Mechanical Engineering*, Vol 55, No 10, October 1933)

[4]Studies in the Design of Local Exhaust Hoods, by J M DallaValle and Theodore Hatch (*A S M E Transactions*, Vol 54, 1932).

[5]Velocity Characteristics of Hoods under Suction, by J M DallaValle (A S H V E TRANSACTIONS, Vol 38, 1932, p 387)

Further, the velocity contours are identical for similar hood shapes when the hoods are reduced to the same basis of comparison. These facts are applicable to all hood problems so that when the velocity contour distribution is known, the air flow required can be determined. Fig. 1 shows the contour distribution in two axial planes perpendicular to the sides of a rectangular hood with a side ratio of one-half. The distribution shown is identical for all openings with a similar side ratio provided the mapping is as shown in the figure. The contours, of course, are expressed as percentages of the velocity at the opening.

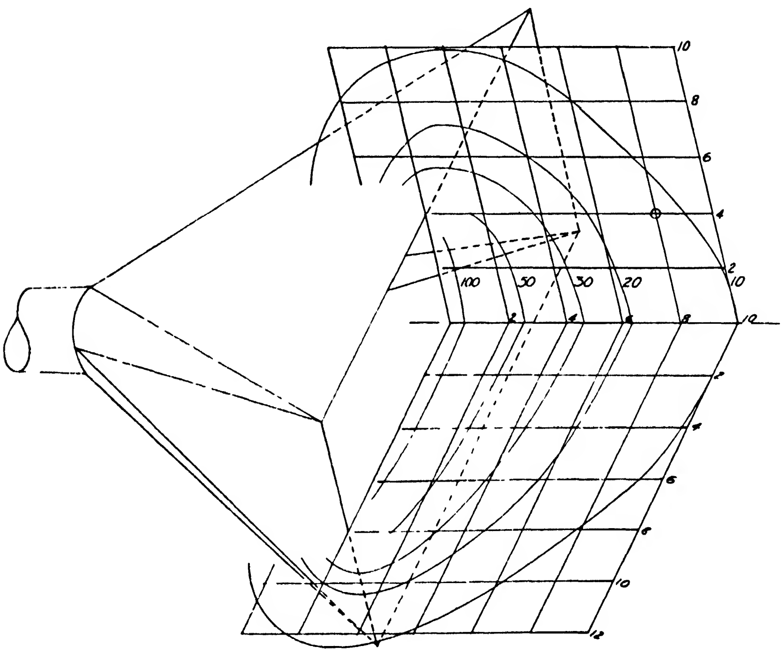

FIG. 1. VELOCITY CONTOURS FOR A RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

Air Flow from Static Readings

The volume of air flow through any hood may be determined from the following equation:

$$Q = 4005 f A \sqrt{h_t} \tag{2}$$

where

Q = volume of air flow, cubic feet per minute.

A = area of connecting duct, square feet.

h_t = static suction at throat of hood, inches of water.

f = orifice or restriction coefficient which varies from 0.6 to 0.9 depending on the shape of the hood.

An average value of f is 0.71, although for a well-shaped opening a value of 0.8 may be used. The factor f is determined from the equation:

$$f = \sqrt{\frac{h_v}{h_t}} \tag{3}$$

where h_v is the velocity head in the connecting duct.

The term *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with respect to the hood are known. This is clearly indicated by Equation 1 which shows that the velocity at any point along the axis varies inversely as the area of the opening and the square of the distance. However, this formula coupled with Equation 2 should serve to indicate the velocity conditions to be expected when operations are conducted external to the hood opening.

Large Open Hoods

Large hoods, such as are used for electroplating and pickling tanks, should be sub-divided so the area of the connecting duct is not less than one-fifteenth of the open area of the hood. Frequently, it will be found necessary to branch the main duct in order to obtain a uniform distribution of flow. *Canopy hoods* should extend 6 in. laterally from the tank for every 12-in. elevation, and wherever possible they should have side and rear aprons so as to prevent short circuiting of air from spaces not directly over the vats or tanks. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off, cyanide fumes, for example, requiring an air velocity of approximately 75 fpm on the surface of the tank and acid and steam vapors requiring velocities as low as 25 to 50 fpm. The total volume of air flow necessary to obtain these velocities may be approximately determined from the following simple formula:

$$Q = 1.4PDV \tag{4}$$

where

Q = total volume of air handled by hood, cubic feet per minute.
P = perimeter of the tank, feet.
D = distance between tank and hood opening, feet.
V = air velocity desired along edges and surface of tank, feet per minute.

Lateral Exhaust Systems

The lateral exhaust method, as developed for chromium plating[6], is applicable in many instances in preference to the canopy type hoods. The method makes use of drawing air and fumes laterally across the top of vats or tanks into slotted ducts at the top and extending fully along one or more sides of the tanks. The slots are 2 in. wide and for effective

[6]Health Hazards in Chromium Plating, by J. J. Bloomfield and Wm. Blum (*U. S Public Health Report*, Vol 43, No 26, September 7, 1928)

ventilation a 2,000 fpm exhaust air velocity at the slot face is advisable. In addition, the duct should not be required to draw the air laterally for a distance of more than 18 in. and the level of the solution should be kept 6 to 8 in. below the top of the tanks.

Flexible Exhaust Systems

The flexible exhaust tube method may be advantageously used for removing dust or fumes. Flexible tubes having one end connected to an exhaust system and a slotted hood attached to the other end may be shaped at will to fit in with industrial processes without affecting the ease of operation. Efficient dust or fume removal may be had with use of relatively small exhaust volumes. This type of system may be used on swing grinders, portable grinding wheels, soldering operations, stone cutting, rock drilling, etc.

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances spraying operations can be performed mechanically in wholly enclosed booths, the volatile vapors may reach injurious or explosive concentrations. At all times the concentrations of these vapors, and particularly those containing benzol, should be kept below 100 parts per million. Spray booth vapors are dangerous to the health of the worker and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located in a horizontal position slightly below the object sprayed. Stagnant regions within the booth should be carefully avoided or should be provided with exhaust. The air volume should be sufficient to maintain a velocity of 150 to 200 fpm over the open area of the booth, and the vapors may be discharged through a suitable stack to permit dilution, but it is better practice to pass the fumes or vapors through baffle type washers or scrubbers designed for efficient spray fume removal[7].

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment and should be well lighted. Air velocities should exceed 50 fpm when the window is opened to its maximum height.

DUCT SYSTEM DESIGN

The duct system should be large enough to transport the fumes or material without causing serious obstruction to the air flow. It is good practice to proportion the ducts to obtain the desired velocities and suction pressures at the hoods, although in many cases only an approximation to an ideal design is possible. Many exhaust hoods, and par-

[7]For a discussion of spray booths, see Special Bulletin No 16, Spray Painting in Pennsylvania, Department of Labor and Industry, 1926 Harrisburg, Pa

ticularly those used in buffing and polishing, are connected by short branch pipes to the main duct which renders proportioning impractical.

Construction

The ducts leading from the hoods to the exhaust fan should be constructed of sheet metal not lighter than is shown in Table 4. The piping should be free from dents, fins and projections on which refuse might catch.

All permanent circular joints should be lap-jointed, riveted and soldered, and all longitudinal joints either grooved and locked or riveted and soldered. Circular laps should be in the direction of the flow, and piping installed out-of-doors should not have the longitudinal laps at the bottom. Every change in pipe size should be made with an eccentric taper flat on the bottom, the taper to be at least 5 in. long for each inch change in diameter. All pipes passing through roofs should be equipped with collars so arranged as to prevent water leaking into the building.

The main trunks and branch pipes should be as short and straight as possible, strongly supported, and with the dead ends capped to permit inspection and cleaning. All branch pipes should join the main at an

TABLE 4. GAGE OF SHEET METAL TO BE USED FOR VARIOUS DUCT DIAMETERS

DIAMETER OF DUCT	GAGE OF METAL
8 in. or less	24
9 to 18 in.	22
19 to 25 in.	20
26 in. or more	18

acute angle, the junction being at the side or top and never at the bottom of the main. Branch pipes should not join the main pipes at points where the material from one branch would tend to enter the branch on the opposite side of the main.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Elbows should be made at least two gages heavier than straight pipe of the same diameter, the better to enable them to withstand the additional wear caused by changing the direction of flow. They should preferably have a throat radius of at least one and one-half times the diameter of the pipe.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely.

The passing of pipes through fire-walls should be avoided wherever possible, and sweep-up connections should be so arranged that foreign material cannot be easily introduced into them.

TABLE 5. AIR SPEEDS IN DUCTS NECESSARY TO CONVEY VARIOUS MATERIALS

MATERIAL	AIR VELOCITIES (FPM)
Grain dust	2000
Wood chips and shavings	3000
Sawdust	2000
Jute dust	2000
Rubber dust	2000
Lint	1500
Metal dust (grindings)	2200
Lead dusts	5000
Brass turnings (fine)	4000
Fine coal	4000

At the point of entrance of a branch pipe with the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always necessary and is frequently done at the expense of a reduced air velocity, it is none the less advisable where future expansion of the exhaust system is contemplated.

Air Velocities in Ducts

When the static suction has been fixed for a given hood, the air velocity in the duct may be determined from Equation 2. Air velocities for conveying a material should be moderate. Table 5 gives the velocities generally employed for conveying various substances. Equations 5 and 5a may be used as tests to determine the conveying efficiency of a system[8]. Velocities determined from these formulae should be increased by at least 25 per cent since they represent the minimum at which a stated size and density of material can be transported.

For vertical ducts:

$$V = 13{,}300 \frac{s}{s+1} d^{0.57} \tag{5}$$

For horizontal ducts:

$$V = 6000 \frac{s}{s+1} d^{0.40} \tag{5a}$$

where

V = air velocity in duct, feet per minute.
s = specific gravity of particles.
d = average diameter of largest particles conveyed, inches.

Example 1. Granular material, the largest size of which is approximately 0.37 in in diameter, with a specific gravity of 1.40 is to be conveyed in a vertical pipe the velocity of the air in which is 4100 fpm; find whether the material can be transported at this velocity.

Substitute data in Equation 5a and multiply by 1.25:

$$V = 1.25 \times 13{,}300 \times \frac{1.4}{2.4} \times 0.37^{0.57}$$

Antilog (0.57 × log 0.37) = 0.568; the required velocity is, therefore, 5500 fpm.

[8]Determining Minimum Air Velocities for Exhaust Systems, by J. M DallaValle (A S.H.V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1932, p. 639)

TABLE 6 LOSS THROUGH 90-DEG ELBOWS

ELBOW CENTER LINE RADIUS IN PER CENT OF PIPE DIAMETER	LOSS IN PER CENT OF VELOCITY HEAD
50	75
100	26
150	17
200 to 300	14

Hence, the duct velocity must be increased either by speeding up the fan or decreasing the diameter of the duct, or both.

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 3, Chapter 29. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 50 to 300 per cent of pipe diameter, the loss may be estimated from Table 6. It is sometimes convenient to express the resistance of an elbow in terms of an equivalent length of duct of the same diameter. Thus with a throat radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately 10 diameters long, while with a throat diameter radius 1½ times the diameter, the resistance is about the same as that of seven diameters of straight pipe.

COLLECTORS

The most common method of separating the dust and other materials from the air is to pass the mixture through a centrifugal or *cyclone* collector. In this type of collector the mixture of the air and material is introduced on a tangent, near the cylindrical top of the collector, and the whirling motion sets up a centrifugal action causing the comparatively heavy materials suspended in the air to be thrown against the side of the separator, from which position they spiral down to the tail piece, while the air escapes through the stack at the center of the collector.

The diameter of the cyclone should be at least 3½ times the diameter of the fan discharge duct. When two or more separate ducts enter a cyclone, gates should be provided to prevent any back draft through a system which may not be operating. Cyclones working in conjunction with two or more fans should be designed to operate efficiently at two-thirds capacity rating. The following formula is useful in computing the loss through a cyclone when the velocity of the air in the fan discharge duct is known:

$$h_c = 0.13 \left(\frac{V}{1000}\right)^2 \tag{6}$$

where

h_c = the pressure drop through the cyclone, inches of water.
V = the air velocity in the fan discharge duct, feet per minute.

If a cyclone is used to collect light dusts such as buffing wheel dusts,

feathers and lint, the exhaust vent should be large enough to permit an air velocity of 200 to 500 fpm. This will, of course, require a cyclone of larger dimensions than given for the foregoing general case.

When a high collection efficiency is desired, or the material is very fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain separation. The capacities and characteristics of this type of separator should be obtained from the manufacturers.

Cloth Filters

Filters are used when the material collected by an exhaust system is valuable or cannot be separated efficiently from the air with an ordinary cyclone. They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. *Bag houses* used in the manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen cloths are employed, the filtering capacities vary from ½ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity. The time increase of resistance varies with the amount of material permitted to build up on the surface of the filter and can be determined only by experiment. The limits of the increase may be regulated by adjustment of the shaking or cleaning mechanism. These limits may be regulated further according to the capacity of the fan and the effective performance of the hoods and the duct system.

For additional information on Dust and Cinders, see Chapter 26, Air Cleaning Devices.

RESISTANCE OF SYSTEM

The maintained resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the pipes.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. The collector drop in inches of water is given approximately by Equation 6, but where possible the resistance of the particular collector to be used should be ascertained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. Find the velocities in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 6. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

EFFICIENCY OF EXHAUST SYSTEMS

The efficiency of an exhaust system depends upon its effectiveness in reducing the concentration of dusts, fumes, vapors and gases below the safe or threshold limits[9].

Too much emphasis cannot be placed on the necessity of testing exhaust systems frequently by determining the concentration of atmospheric contamination at the worker's breathing level. Commonly accepted values of threshold limits for the usual gases and vapors are given in Table 7.

SELECTION OF FANS AND MOTORS

Manufacturers generally provide special fans for the collection of various industrial wastes. These are available for the collection of coal dust, wood shavings, wool, cotton and many other substances. For particular features concerning special fans, consult the *Catalog Data Section* of THE GUIDE and manufacturers' data. When substances having an abrasive character are conveyed, the fan blades and housing should be protected from wear. This may be accomplished by placing a collector on the negative side of the fan or by lining the housing and blades with rubber.

TABLE 7. THRESHOLD LIMITS OF COMMON VAPORS AND GASES[a]

SUBSTANCE	SPEC GRAV OF GAS OR VAPOR (AIR 1)	INFLAMMABLE LIMITS (%)	PHYSIOLOGICAL ACTION	MAXIMUM ALLOWABLE CONCENTRATION (PPM)
Chlorine	2.486	non-inflamm	irritant	0.35
Ozone	5.5	do	do	0.80
Hydrogen chloride	1.2678	do	do	10.0
Sulphur dioxide	2 2638	do	do	10.0
Carbon monoxide	0 9671	12 5–74	asphyxiant	100 0
Hydrogen sulphide	1 190	4.3–46	do	85–130
Benzene	2 73	1 4–7.0	anesthetic	100 0
Methanol	1 1	7 5–26 5	do	100 0
Carbon tetrachloride	5 3	non-inflamm	do	100.0

[a]The Prevention of Occupational Diseases, by R R Sayers and J M DallaValle (*Mechanical Engineering*, Vol 57, No 4, April, 1935)

If no future expansion of an exhaust system is contemplated, the fan motor should be chosen to provide the calculated air volume. Should, however, the exhaust system be required to handle more air in the future, the motor should be adequate for the maximum load anticipated. Further information regarding the choice of fans and motors is given in Chapters 27 and 38.

PROTECTION AGAINST CORROSION

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to

[9]Criteria for Industrial Exhaust Systems, by J J Bloomfield (A S H V E TRANSACTIONS, Vol 40, 1934, p 353)

TABLE 8. MATERIALS TO BE USED FOR THE PROTECTION OF EXHAUST SYSTEMS AGAINST CORROSION[a]

Type of Fume Conveyed	Protective Material to be Used
Chlorine	Rubber lining or chrome-nickel alloys
Hydrogen sulphide	Aluminum coated iron, aluminum, high chrome-nickel alloys
Ammonia	Iron or steel
Sulphurous gases	High chrome-nickel alloys
Hydrochloric acid	Rubber lining, chrome-nickel alloys
Nitrous gases	Nickel-chrome alloys

[a]Condensed from data given by Chilton and Huey (*Industrial and Engineering Chemistry*, Vol 24, 1932)

chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 8. Hoods and ducts when short, may frequently be constructed of wood and be quite effective. Rubberized paints are available and may be applied as protective coatings in handling such gases and fumes as chlorine and hydrochloric acid.

PROBLEMS IN PRACTICE

1 ● What is the most common method of reducing total air volumes handled in cases employing large hoods over apparatus covering a large area?

The use of the petticoats on large hoods which permits a comparatively high air velocity at the rim of the hood and controllably small velocities in the center.

2 ● Why is it not permissible to connect up emery wheels and buffing wheels to the same exhaust system?

Emery wheels and buffing wheels should be handled by separate systems because of the fire hazard, as it is possible for sparks from the emery wheels to ignite the lint and dust from the buffing wheels when both are carried through the same system.

3 ● A tank, 4 ft by 8 ft, contains a fluid which gives off injurious vapors. A large hood is located 30 in. above the top of the tank and extends slightly over its edges. Assuming that a velocity of 60 fpm is required to adequately control the vapors near the edges of the tank, calculate the air flow required.

Using Equation 4, $P = 2 \times 4 + 2 \times 8 = 24$ ft; $D = 30$ in. $= 2.5$ ft, $V = 60$ fpm.

$$\text{Hence, } Q = 1.4 \times 24 \times 2.5 \times 60 = 5040 \text{ cfm.}$$

4 ● Silica dust with a specific gravity of 2.65 is being conveyed in a duct system. The velocity measured in a vertical portion of the system is found to be 2700 fpm. What is the maximum diameter particle transported at this velocity?

$$\text{Using Equation 5a, } 2700 = 13{,}300 \times \frac{2.65}{3.65} \times d^{0.570}$$

from which

$$d = (0.28)^{1.75} = 0.11 \text{ in.}$$

5 ● What special materials may be used to resist chemical corrosion in a system exhausting gases and fumes?

Various protective materials are available for exhaust systems depending largely upon the type of fumes conveyed. Nickel-chrome alloys, aluminum coated metals and rubber linings are extensively used. Also protective rubberized paints are available which may be applied for conveying chlorine and hydrochloric acid fumes.

Chapter 35

DRYING SYSTEMS

Drying Methods, Driers, Mechanism of Drying, Moisture, General Rules for Drying, Equipment, Humidity Chart, Combustion, Design, Estimating Methods

DRYING, in its broader sense, refers to the removal of water, or other volatile liquid from either a gaseous, liquid, or solid material. In practice, the process of direct drying gaseous material is referred to generally as dehumidifying, or condensing, and in some cases chemicals are used in the adsorption or absorption of moisture. Drying a liquid is called evaporation or distillation. The common usage of the word *drying* refers to the removal of water or other liquid, such as a solvent, by evaporation from a solid material.

When the solid to be dried contains large amounts of free water, the actual drying process is frequently preceded by the removal of part of the water by some mechanical means, such as filtration, settling, pressing or centrifuging. Removal of as much water as possible by such methods is usually advisable, as the cost of these operations, per pound of water removed, is generally much less than by evaporation.

DRYING METHODS

Drying may be accomplished in any one or combination of the following methods:

1. Radiation.
2. Conduction, or direct contact
3. Convection.

Radiation

The source of heat for radiation may be either the sun, or heated surfaces. Sun drying is practiced where danger from rain is slight, and where sufficient time can be allowed. Where a strict adherence to a schedule is necessary, or where dusty atmosphere is present, this method is not in favor. Fruits are often dried in the sun.

Radiation from hot surfaces (heated by steam, electricity, or other means) furnishes generally, from one-third to one-half the total heat required for evaporation. Convection currents set up by these hot surfaces and the cooler materials carry the balance of the heat.

TABLE 1. DRIERS FOR EVAPORATION OF WATER

TYPE	KIND	MATERIALS HANDLED	MEANS OF HANDLING	TEMP RANGE DEG F	HEAT SUPPLY	USES AND REMARKS
Batch or Intermittent	Compartment	Paper, Leather, Yarns, Lumber, Foodstuffs	Suspended, Truck, Tray	80 to 180	Steam Coils, Air, Electricity	When production does not warrant continuous drier
	Agitated	Chemicals too sticky for Rotary Drier	Shoveled into Drum or Pan	100 to 330	Water, Steam Jacketed, may have Vacuum on top	Where dust must be saved
	Vacuum	Chemicals, Explosives, Pharmaceuticals, Food Products	Tray Basket. Tumbling Drum	80 to 300	Water, Steam	Cost of operation high, for expensive materials
Continuous	Tunnel	Ceramics, Chemicals, Lumber, Food Products	Truck, Tray, Belt	100 to 350	Steam Coils, Air, Electricity, Products of Combustion	For High Production
	Rotary	Bulk	Cascades through	80 to 500	Air, Steam, Products of Combustion	Where material will stand rough handling and is not subject to balling up
	Drum	Liquids, Slurries	Flowed on Drum, Dry Material Scraped off	to 310	Steam, may have Vacuum on Top	Hygroscopic materials dried with vacuum, and packed immediately
	Cylinder	Paper, Textiles, Chemicals	Continuous Sheets, Endless Chain Belt	to 350	Steam inside of Drum	Where material comes in sheets or rolls, and will stand direct contact with heating surface
	Festoon	Paper, Chemicals	Continuous Sheets, Suspended on Metal Screens	to 200	Air, Steam Coils	Where one side cannot come in contact with supports until dry
	Tower or Column	Grains, Sand	Falls through by Gravity	125 to 250	Air, Steam Coils	Where headroom is available
	Spray	Solutions over 30% Solids	Sprayed into Chamber	120 to 350	Air, Products of Combustion	Drying is almost instantaneous
	Induction	Metals, for removal of traces of Water	Placed in High Frequency Field	to 400	Electricity	Where heating of metal from inside out is important

Conduction or Direct Contact Drying

This method of drying is advantageous where the material can be flowed on to the drying surface and the dried material scraped off, or where the material to be dried can be handled in a sheet, and where there is no danger of subjecting the product to the full temperature of the heating medium. The source of heat for this method may be steam, electricity, hot oil or hot water.

Convection

The circulation of heated air or other gases about the material to be dried is generally termed convection drying. The convection may be either natural or forced. With forced circulation, the temperature of the

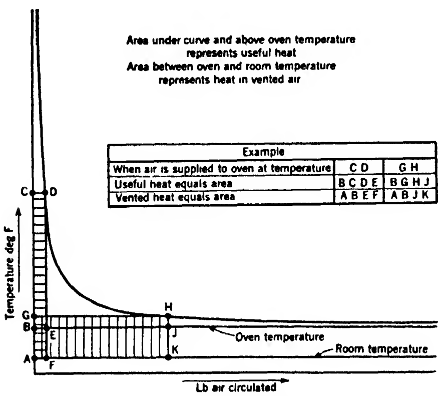

FIG. 1. RELATION BETWEEN USEFUL AND TOTAL HEAT SUPPLIED

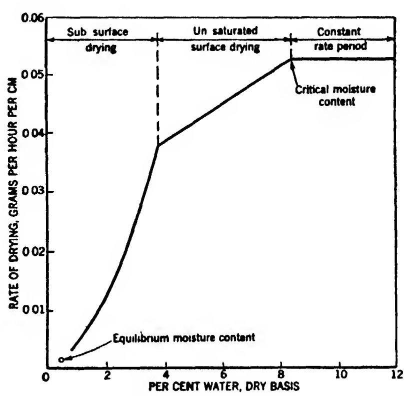

FIG. 2. RATE OF DRYING OF WHITING SLAB

drier is more uniform and the rate of drying is much higher than with natural circulation. Where humidity is used, the control is much easier, and more accurate.

The source of heat for a convection drier may be steam, electricity, hot water, oil-fired heater, gas-fired heater, or coal furnace. Where either oil, gas or coal is used, the type of heater may be direct or indirect; *i.e.*, the products of combustion may be used (direct), or the circulated air may be heated through an interchanger (indirect).

Where the direct type is used, there is naturally a higher thermal efficiency, but it can only be used where the odor, soot, or the chemical elements of the products of combustion do not affect the material being dried. When heat economy is an important consideration this method (Fig. 1) may be used, permitting a small amount of air to be circulated, if a sacrifice of accurate control of temperature and humidity can be justified.

DRIERS

The term *adiabatic drier* is applied to a drier in which all the heat is supplied by air externally heated. The temperature of the air in the

drier decreases as a transfer of heat to the material being dried takes place. Where part or all of the heat is supplied by steam coils or other means, within the drier itself, the drier is known as a *constant temperature drier.* Driers using little air for heating medium with a high temperature drop, are difficult to hold at uniform temperatures; the more air used, the easier it is to secure accurate control of temperature and humidity. Driers may be classified as shown in Table 1.

MECHANISM OF DRYING

The modern theory of drying may be summed up as follows: Assuming uniform velocity and distribution of air at a constant temperature and humidity over the surface to be dried, the drying cycle will be divided into two distinct stages:

1. Constant rate period.
2. Falling rate period.

The *constant rate period* occurs while the material being dried is still very wet, and continues as long as the water in the material comes to the surface so rapidly that the surface remains thoroughly wet, and evaporation proceeds at a constant rate, precisely as from a free water surface. The material assumes a temperature corresponding to the wet-bulb temperature of the surrounding air, or slightly higher, due to radiation and conduction from dry surfaces adjoining the material. The constant rate period continues until a time when the moisture no longer comes to the surface as fast as it is evaporated. This point is called the *critical moisture content.*

As the drying proceeds, a period of *uniform falling rate* is entered. During this period, the surface of the material is gradually drying out, and the rate of drying falls as the remaining wet surface decreases in area. This period is also known as unsaturated surface drying.

As drying continues, the surface is completely dry and the water from the interior evaporates and comes through the surface as vapor. As the plane of water recedes, the diffusion of the vapor becomes more difficult and hence the period is known as *varying falling rate period*, or sub-surface drying.

As drying progresses another point called *equilibrium moisture content* is reached, where the vapor pressure of the moisture in the air and the vapor pressure of the moisture in the material are equal, and drying ceases. The drying of a slab of whiting is shown in Fig. 2 and illustrates the principles pointed out above. The factors affecting the variations of drying rates during the above periods are pointed out in Table 2.

Omissions in the Cycle

Many solids, such as lumber, are so dry at the beginning of the drying operation that the constant rate period of free surface evaporation does not occur. Frequently the surface of the material is dry enough so that no surface drying can take place, in which case only the final stage of sub-surface drying is involved. In other instances, the critical moisture content of a wet solid is sufficiently low that sub-surface drying starts almost immediately after the conclusion of the constant rate period. Thus the

intermediate state of unsaturated surface drying does not occur and the drying is of the sub-surface type during practically the whole of the falling rate period. With other kinds of material, particularly thin sheets, such as newsprint paper, sub-surface drying may occur at such a low moisture content that it is not encountered in commercial work, the

TABLE 2. FACTORS INFLUENCING DRYING

FACTOR	DRYING PERIOD	
	Constant Rate, Unsaturated Surface	Sub-Surface
Temperature	Increase in temperature increases drying rate	Increase in temperature increases drying rate, because with decreased viscosity, diffusion increases
Humidity	Drying rate increases as humidity is decreased	No effect until equilibrium content is reached, drying then ceases
Air Velocity	Drying rate varies approximately as the 0 6 power of the velocity	No effect
Air direction	Drying rate increases the more nearly the air blows perpendicular to surface, for dead air film becomes thinner	No effect
Thickness of Material	Drying rate is not affected by the thickness	Drying rate varies inversely as the square of the thickness

falling rate period being confined solely in practice, to unsaturated surface drying.

MOISTURE

Moisture in the solid may be in either of two forms:

1. Capillary or free
2 Hygroscopic or chemically combined.

Free moisture is contained in the capillary spaces between the particles or fibers of the materials. The loss of this moisture changes only the weight of the material. *Chemically combined* or hygroscopic moisture is intimately associated with the physical nature of the material and its removal changes both the physical characteristics as well as the chemical properties. The amount of hygroscopic moisture a material can contain is limited. This limit is called the *fiber saturation point.* When material is dried below this point, care must be exercised to avoid physical changes in the material, such as shrinkage, hardening, etc. All hygroscopic materials have definite equilibrium moisture contents dependent on temperature and humidity. Materials are frequently dried to a lower moisture content than those of equilibrium conditions in use, and allowed to regain the necessary moisture after leaving the drier to equalize the moisture in the material. Fig. 3[1] shows the equilibrium moisture content of wood.

[1] *U S Department of Agriculture Bulletin*, No. 1136

GENERAL RULES FOR DRYING

Temperature

The highest temperature possible should be used because of faster drying and smaller requirements for ventilation. The amount of moisture that can be carried by a pound of air increases rapidly with rise in temperature as shown in the humidity chart of Fig. 4. Too high a temperature may cause spoilage of materials; many materials calcine or change their chemical properties if heated too hot; gypsum and glauber salts lose some of the chemically combined water, fall apart, and change their chemical properties. Too high or rapid rise in temperatures in drying lumber or ceramics may create a liquid vapor tension within the material so high that the cells explode, causing permanent injury to the fiber. If too high a temperature is used on some chemicals, they begin to react

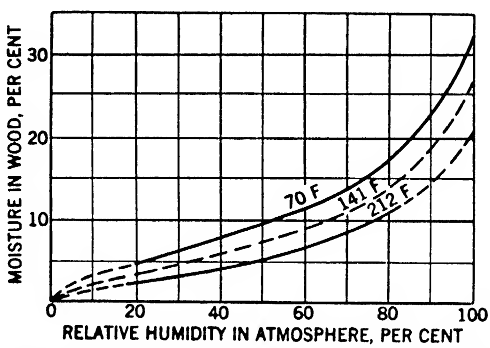

FIG. 3. RELATION OF EQUILIBRIUM MOISTURE CONTENT IN WOOD TO THE RELATIVE HUMIDITY OF SURROUNDING AIR

exothermally; a temperature rise and chemical action from within will burn the materials, *e.g.*, bakelite products, gunpowder, etc. During the constant rate period of drying, the material heats only to the wet-bulb temperature of the surrounding air, consequently high temperatures will not injure the material in this stage.

Humidity

Moisture in the drying air may be very important. Many materials tend to case-harden, dry on the outside, forming a skin which retards the moisture flow from the inside to the surface, or stops it completely, and so increases the drying time very much or causes a change of the physical properties of the material. It is often necessary to add humidity to the air in the initial stage of drying. Lumber case-hardens, cracks, and warps if the outside is dried too fast. Ceramics crack if not heated through before drying commences. Elastic materials warp while others crack if not evenly dried. Many paints case-harden if not dried under high humidity.

On the other hand, in the case of those materials whose physical or chemical properties require that they be dried at relatively low temperatures high humidity tends to retard drying in the first stage and may even stop it altogether in the final stage. Where drying temperatures

below 120 to 140 F are used the drying rate may be highly dependent on atmospheric humidity conditions. In such instances it is often desirable to dehumidify the air entering the drier during periods of high atmospheric humidity; where a high degree of uniformity is required it is often possible to secure complete independence of atmospheric conditions by recirculating the air in a closed system which includes a suitable dehumidifier. For this purpose absorptive dehumidifying systems have the advantage of accomplishing the desired reduction of humidity without appreciably elevating or lowering the dry-bulb temperature of the air; for this reason after-cooling is not required, and reheating is reduced to a minimum. Complete descriptions of such dehumidifying systems are given in Chapter 23 on Cooling and Dehumidification Methods.

Air Circulation

As noted under Mechanism of Drying, air velocity is more important in the first two stages of drying than in the last, and for this reason zone drying in continuous driers is frequently considered. It permits accurate regulation of temperature, humidity, and velocity in the different zones. High velocity results in more rapid drying, more even distribution of temperature and consequently more even drying in the first period. Too high a velocity may be detrimental because of excessive power needed for creating it, or because the material may blow away if it is light and fluffy. In the drying of paints, varnishes, and enamels, high velocity or improper distribution of the air even with the use of filters, may cause dust already in the drier, to be blown against the material, ruining the finish. Table 3 presents data on drying of various materials.

EQUIPMENT FOR DRYING

Equipment for drying may be divided into the following classes:

1. Heat and humidity supply.
2. Methods of handling.
3. Ovens.

The heat and humidity supply for low temperature work up to 250 F is often steam; steam coils either in the oven or outside, heat the air used for drying. Circulation of heated oil is used to a limited extent, but the danger of leaks is serious, for if the oil is hotter than the flash point, a fire may start if the oil is released to the atmosphere. In many cases where steam is not available, direct or indirect fired heaters are used with gas or oil as fuel. Indirect heaters should be carefully selected from a standpoint of long life and efficiency. The heat exchange surface should be adequate in area and easily accessible for cleaning and removal. For extremely high temperatures, alloy surface may be used. With direct-fired equipment care must be used in the selection of burners and sufficient combustion space allowed to insure complete combustion of fuel. Humidity can be obtained in driers by the use of steam spray, humidifiers, or recirculation.

Methods of *handling of material* have been indicated in Table 1.

For low temperature work up to 200 F *ovens* and driers are commonly built of two thicknesses of insulating board (fireproof preferred), with air space between. As the temperature increases materials better able to

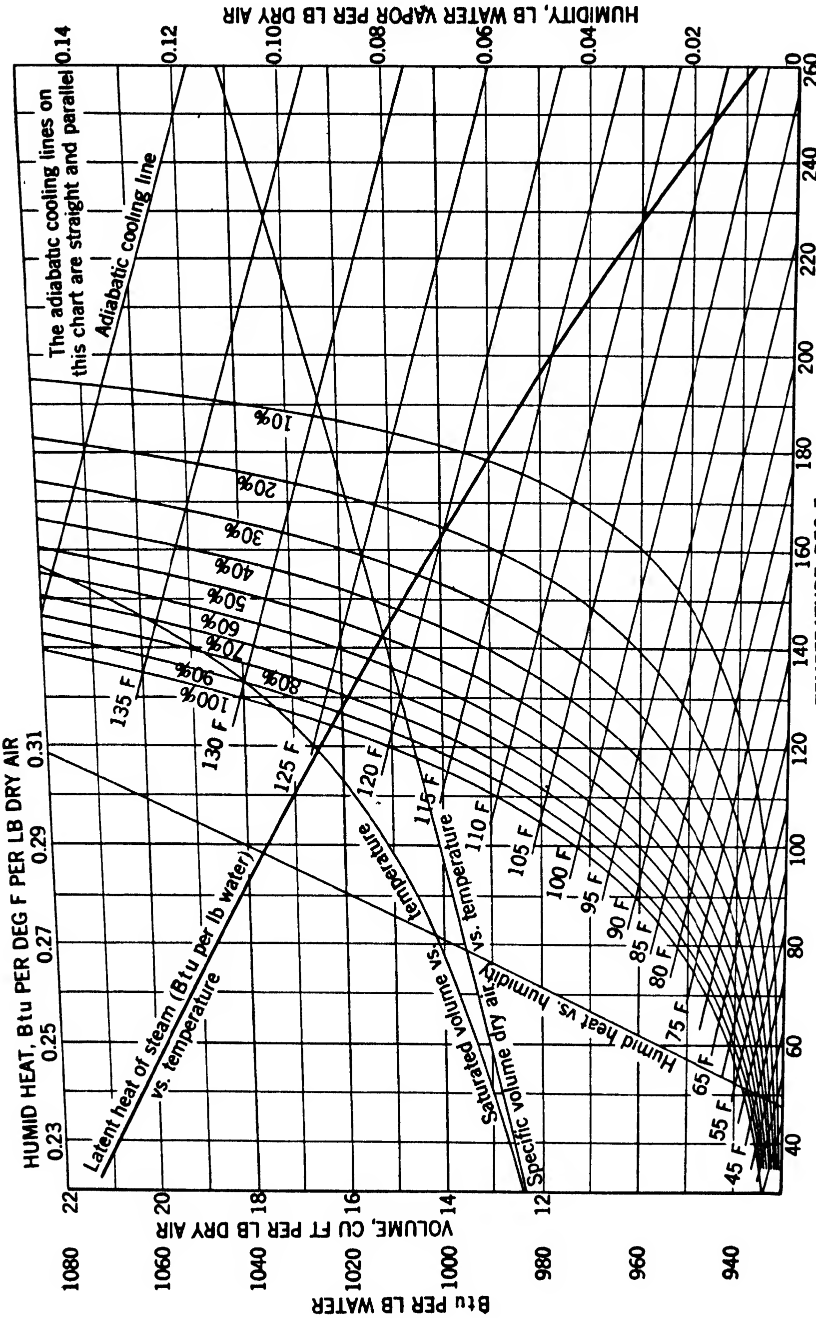

FIG. 4. HUMIDITY CHART

withstand the heat must be used. Metal lined ovens are easy to keep clean, and many high temperature driers up to 1000 F are made of metal panels with insulation between. Care should be taken to avoid *through metal* (metal extending through the oven from inside to out). Batch type ovens are entirely closed while in use and control of air leakage is easily taken care of. In the continuous drier where the ends are open, heat and air leakage becomes important. Warm air leaking out of the ends of ovens means a heat loss, and often the temperature and humidity outside the oven becomes unbearable. For this reason, inclined or bottom entry ovens are used, as the warm air leakage can be more easily controlled. See Figs. 5 and 6.

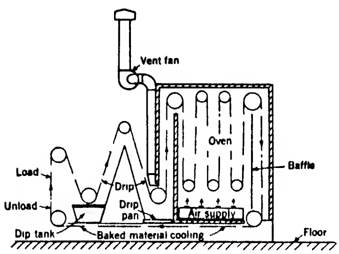

FIG. 5. SMALL PART MULTIPLE PASS OVEN

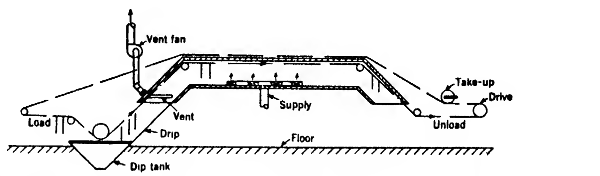

FIG. 6. INCLINED END ENAMELING OVEN

HUMIDITY CHART FOR DRYING WORK

In drying problems the chemical engineer uses different psychrometric values than those used by the heating, ventilating and air conditioning engineer. The humidity chart illustrated in Fig. 4 is based upon values determined from the following explanations:

Humidity (H) is the number of pounds of water vapor carried by one pound of dry air.

Percentage Humidity ($\%H$) is the number of pounds of water vapor carried by one pound of dry air at a definite temperature, divided by the number of pounds of vapor that one pound of dry air would carry if it were completely saturated at the same temperature.

Per Cent Relative Humidity (Φ) is the ratio of weight of water vapor contained in any given volume of air, to the weight of water vapor present in the same volume of saturated air, all values referring to the same temperature.

To convert from one relation to the other,

$$\% H = \frac{29.92 - p_s}{29.92 - p} \times \Phi \tag{1}$$

where

p_s = vapor pressure of water, inches mercury; at dry-bulb temperature, degrees Fahrenheit.

$p = \Phi\, p_s$.

COMBUSTION

Where products of combustion are used directly in the oven, a knowledge of their formation and heat values is important. The properties of

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]

MATERIAL	TEMPERATURE DEG F	PER CENT RELATIVE HUMIDITY	DRYING TIME
Apples	140–180		6 Hrs
Armatures Varnish	200		2.5 Hrs
Banana Food ¼ in. Thick	140		4–6 Hrs
Barrels	300		15 Min
Beans	140		18 Hrs
Bedding	150–190		
Blankets	120		40 Min
Brake Lining	325		12 Min
Brick continuous	350 to 90		24 Hrs
Briquets	1100		108 Min
Cabbage Raw	150		4.5 Hrs
Candied Peel	165		2 Hrs
Casein	180		5 Hrs
Cereals	110–150		
Ceramics before firing	150	70 to 20	24 Hrs
Chicle	95–100		
Coco-fiber mats	170–210		10 Hrs
Cocoanut	150–200		4–6 Hrs
Coffee	160–180		24 Hrs
Conduit (Enamel)	400 Max		2 Hrs
Cores, Oil sand for molding ½—1 in. thick	300		30 Min
Black sand with goulic binder about 0.6 of time 3 in. thick	480		2.5 Hrs
Black sand with goulic binder about 0.6 of time 8 in. thick	480		4.5 Hrs
Black sand with goulic binder about 0.6 of time 16 in. thick	700		10 Hrs
Cores, Crank case (in continuous ovens)	525–600		2–3 Hrs
Cores, Radiator (in continuous ovens)	275–450		1.5 Hrs
Cornstalk Board	150		2 Hrs
Cotton Linters	180		
Enamels synthetic			
Finish coat on autos	225		2 Hrs + Air Dry
Ice boxes all metal (white)	290–425		1 Hr
Ice boxes wood inside (white)	225		3 Hrs
Enamel not synthetic			
Fence posts green	200		1 Hr
Golf balls (white)	90–95	40–50	18–36 Hrs
Small parts (auto) black	450		1 Hr
Steel furniture	225–300		30–350 Min
License plates	250		1 5 Hrs
Feathers	150–180		
Films, Photographic	85–110		20–30 Min
Fruits and Vegetables	140		2–6 Hrs
Furs	110		
Gelatin	110		
Glue bone, thin sheets on wire trays	70–90		6–9 Days
Glue skin	70–90		2 Days
Glue size on furniture	130		4 Hrs
Gut	150		
Gypsum board ⅜ in. thick Start Wet	350		60 Min
Gypsum board ⅜ in. thick Finish	275		
Gypsum block	350–190		8–16 Hrs
Hair felt	180–200		
Hair goods	150–190		1 Hr
Hanks on poles	120		2 Hrs
Hats felt	140–180		
Hides thin leather	90		2–4 Hrs
Hides heavy	70–90		4–6 Days

[a]See references at end of chapter.

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]—Con.

Material	Temperature Deg F	Per Cent Relative Humidity	Drying Time
Hops	120–180		
Ink printing	70–300		
Japan beds	300		1.5–2 Hrs
Japan cash register	300–450		1 5 Hrs
Japan metal shelving	200		30 Min
Knitted fabrics	140–180		
Leather mulling	78–95	85	
Leather thick sole	90	70	4 Days
Leather uppers	80		2–3 Days
Linoleum varnish	110–145	10–30	6–10 Hrs
Lithographing on tin color work	250–270		18–25 Min
Lithographing on tin Japan	350		
Lumber green hardwood	100–180		3-180 Days
Lumber green soft wood	160–220		2–14 Days
Macaroni	90–110		7.5–8 Hrs
Matches	140–180		
Matrix	350		15 Min
Milk and other liquid foods spray dried	135–300		Instantaneous
Millboard sheets	95		10 Hrs
Moulds green sand C I. flasks (one surface only exposed) 8 in. thick	600		6 Hrs
Moulds green sand C I. flasks (one surface only exposed) 13 in. thick	700		13 Hrs
Motors, field coils	180		6 Hrs
Motors, stators	250		6.5 Hrs
Noodles	90–95		
Nuts	75–140		24 Hrs
Oil cloth	150		
Paint, wood wheels	150	35	8–24 Hrs
Paint, on sheet metal	350–140	22–30	2 5 Hrs
Paper, machine dried	180		
Paper, air dried	90–200		
Paper wall, ground coat	140		3 Min
Paper wall, varnished	140–160	45	15 Min
Paper cardboard, spirit varnish	150		1–2 Min
Peaches	135		26 Hrs
Pears	140		24 Hrs
Peas	150		6 Hrs
Potatoes sliced	85		4 Hrs
Potatoes steamed	170		6 5 Hrs
Prunes	140		
Rags	180		
Ramie fiber	140		10 Hrs
Rice	150		
Rock wool insulation	300		8 Hrs
Rubber	85–90		6–12 Hrs
Rubber reclaimed	140–200		1–2 Hrs
Rugs	190		4–8 Hrs
Salt	350		Rotary Drier
Sand loose 1 in. deep	300		10–15 Min
Sausage casings	110		5 Hrs
Shade cloth	240		1–2 Hrs
Shirts	120		20 Min
Soap	100–125		12–72 Hrs
Starch	180–200		1–4 Hrs
Stock feed mixed	180–220		20–30 Min
Storage battery plates	100–110	90 for	24 Hrs
	250	Low for	6 Hrs
Sugar	150–200		20–30 Min

[a]See references at end of chapter

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]—Con.

MATERIAL	TEMPERATURE DEG F	PER CENT RELATIVE HUMIDITY	DRYING TIME
Tanin and other chemicals (spray dried)	250–300		Instantaneous
Terra Cotta (air drying in conditioned room)	150–200		12–96 Hrs
Tobacco leaves	85–130		12 Hrs
Tobacco stems	180–200		12 Hrs
Varnish refrigerator boxes	110	35	5–7 Hrs
Varnish steering wheels	110–140	25–35	Overnight
Veneer ¼ in. 3-ply	120–130	35–40	6–8 Hrs + 2 Hrs acclimation
1³⁄₁₆ in. 5-ply	120–130	35–40	16–18 Hrs + 4 Hrs acclimation
1¼ in. 5-ply	120–130	35–40	20–24 Hrs + 4 Hrs acclimation
Vitreous Enamel sheets before firing	170		
Wallboard pasted plywood	300		15–20 Min
Wallboard fiber insulating, roller type drier	300–385		2½–3 Hrs
Wallboard fiber insulating, truck type drier	300–385		24–48 Hrs
Walnuts	100		24 Hrs
Wheat, corn, oats, rice, barley	180		
Wire cloth Japan	200		20 Min
Wool	105		

[a]See references at end of chapter

the common constituents of fuel are shown in Table 4. The heating values of oils are shown in Fig. 7. The sensible heat in Btu contained in the products of combustion of an average fuel oil and various gases is given in Fig. 8. The problem of securing complete combustion in a heater is important, in order to secure efficiency and the absence of soot formation, but unlike the ordinary power or heating boiler, excess air need not be maintained at a minimum in most cases. Excess air is generally admitted either in the heater or before the products go into the drier.

DESIGN

In all drying problems, data regarding temperatures, time, and humidity must be obtained by experiment or previous experience. Experiments are best performed at the temperatures, humidities, and velocities to be actually used in the full sized drier, and with full size samples.

The following nomenclature and explanation of terms will be used in the discussion of drying calculations:

H = humidity of air, pounds of water vapor per pound of dry air.
G = pounds of dry air supplied to the drier per unit of time.
S = pounds of stock dried per unit of time in a continuous drier.
S' = pounds of stock charged per batch to a discontinuous drier.
Θ = time.
Q = total heat supplied to the drier.

t = air temperature.
t' = stock temperature
t'' = average stock temperature over short time interval, in a batch drier.
t_w = wet-bulb temperature.
s' = specific heat of the stock.
B = total radiation and conduction losses per unit time.
w = pounds of water per pound of dry stock.
r = heat of evaporation of water.
s = humid heat of air, *i.e.*, heat necessary to raise 1 lb of dry air + H lb of steam 1 F.

Subscript (1) designates conditions at the point where the material in question (air or stock) enters and (2) where it leaves the drier.

Air driers may be divided into two classes, those in which *all moisture* evaporated from the stock *leaves the drier as vapor* in the effluent air, and those in which *part or all* of the moisture *is condensed* from the air *in the drying equipment itself*. In any continuously operating drier of the first type the relation between moisture content of the stock and quantity of air required for the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S(w_1 - w_2) \tag{2}$$

TABLE 4. GAS COMBUSTION CONSTANTS[a]

Gas	Chemical Formula	Molecular Weight	Cu Ft per Lb	Heat of Combustion, Btu per Lb, Gross	Heat of Combustion, Btu per Lb, Net	Lbs per Lb of Combustible, Required for Combustion, O_2	Required for Combustion, N_2	Required for Combustion, Air	Flue Products, CO_2	Flue Products, H_2O	Flue Products, N_2
Carbon	C	12 000	....	14,140	14,140	2 667	8 873	11 540	3 667	.. .. .	8 873
Hydrogen	H_2	2 015	187 723	61,100	51,643	7 939	26 414	34 353		8 939	26 414
Oxygen	O_2	32 000	11 819	...	...	.	...	...	...		...
Nitrogen	N_2	28 016	13 443	...	...	...	...	...	...	...	...
Carbon Monoxide	CO	28 000	13 506	4,369	4,369	0 571	1 900	2 471	1 571	. . .	1 900
Carbon Dioxide	CO_2	44 000	8 548	...	...	...	...	...	...	...	...
Methane	CH_4	16 031	23 565	23,912	21,533	3 992	13 282	17 274	2 745	2 248	13 282
Ethane	C_2H_6	30 046	12 455	22,215	20,312	3 728	12 404	16 132	2 929	1 799	12.404
Propane	C_3H_8	44 062	8 365	21,564	19,834	3 631	12 081	15 712	2 996	1.635	12 081
Sulphur Dioxide	SO_2	64 060	5 770	. .	...	...	...	...	...	...	...
Water Vapor	H_2O	18 015	21 017	...	...	...	...	...	...	...	...
Air	...	28 900	13 063	. ...	...	...	...	...	...	...	...

[a]All gas volumes corrected to 60 F and 30 in. mercury barometric pressure dry.

In discontinuous driers, *e.g.*, compartment driers, the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S'\,\frac{dw}{d\Theta} \tag{2a}$$

In the continuous drier, the heat consumption per unit time is:

$$\frac{Q}{\Theta} = Gs_1(t_2 - t_1) + G(r_2 + t_2 - t'_2)\,(H_2 - H_1) + S(t'_2 - t'_1)\,(s' + w_1) + B \tag{3}$$

Equation (3) assumes continuity of operation. For charge or batch operations, the total time of the drying cycle may be broken up into a number of periods, sufficiently short so that over each period average

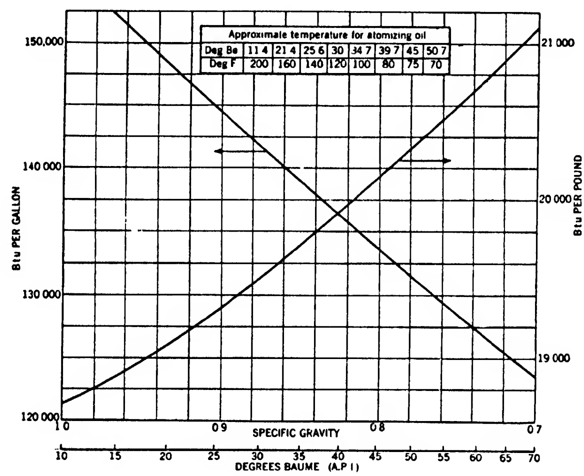

FIG. 7. HEATING VALUES OF FUEL OIL, BTU GROSS

values of t, t' and H may be employed provided the third term of the right hand member of the equation is modified to read:

$$S'\,(t''_1 - t''_1)\,(s' - w_1)$$

and in the second term t'_2 be replaced by

$$\frac{t'_1 + t''_2}{2}$$

Theoretically these periods should be very short and the equation integrated. Practically the error introduced by using a small number of long periods and employing average values of the variables over each, rarely introduces serious error. The evaluation of equation (2a) may be approximated in a similar manner.

The first term of the right hand member of equation (3) represents heat lost as sensible heat in the effluent air. In many drying operations this becomes excessive. Each pound of air supplied should remove the maximum amount of moisture. This is best accomplished by bringing the air

into contact with the stock with sufficient intimacy so that the air leaving the drier is saturated, or nearly so. Counter-current as against parallel flow of air and stock gives rise to optimum operating conditions, resulting in a minimum quantity of air required (G), and a corresponding minimum loss, as sensible heat, in the exit air. Similarly, continuous operation is superior to intermittent operation.

Despite the fact that the sensible heat loss increases with the rise in temperature of the air, the percentage of heat lost from this source decreases, provided the increase in moisture carrying capacity of the air,

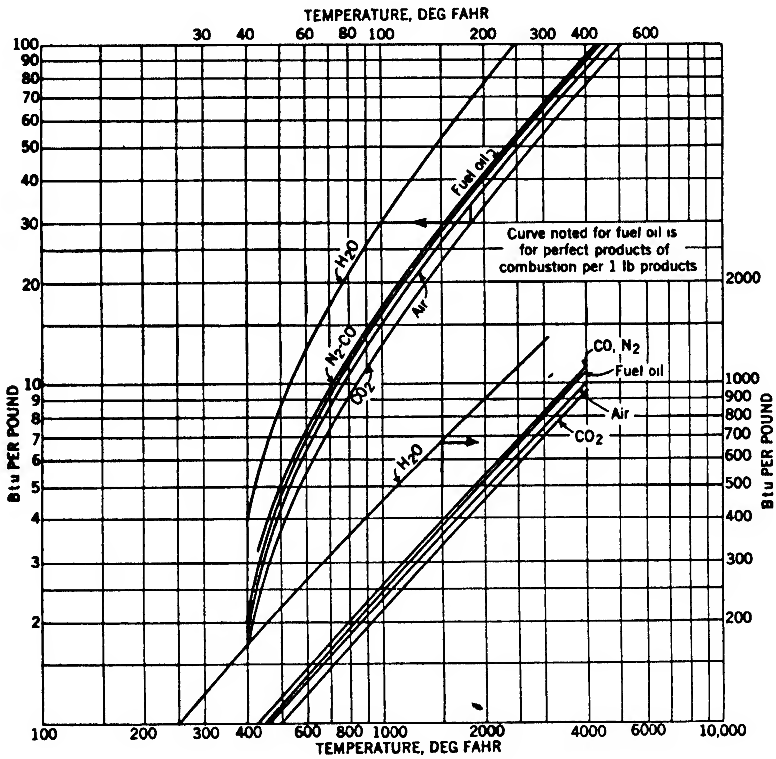

FIG. 8. HEAT CONTENT OF GASES ABOVE 32 F IN BTU PER POUND

due to high temperature, is actually utilized. To secure maximum thermal efficiency in drying, a high drying temperature and high saturation of the outlet air is imperative.

Ventilation Phase

The technique of attack of the *ventilation phase* of a drying problem is best made clear by an illustration. Assume that a material containing 40 per cent moisture is to be dried until this quantity of moisture is reduced to 5 per cent by weight. The material will stand an air temperature of 150 F and it is possible to provide sufficiently good contact between the material and the drying air so that the effluent air can be

brought up to 50 per cent humidity at 150 F. The drier is to use room air, the temperature and humidity of which may be assumed to average 70 F and 50 per cent. A counter-current drier will be employed and the air in this drier will be kept at a substantially constant temperature of 150 F by heaters thermostatically controlled. The stock enters at 70 F, rises quickly to the wet-bulb temperature of the air, with which it is in

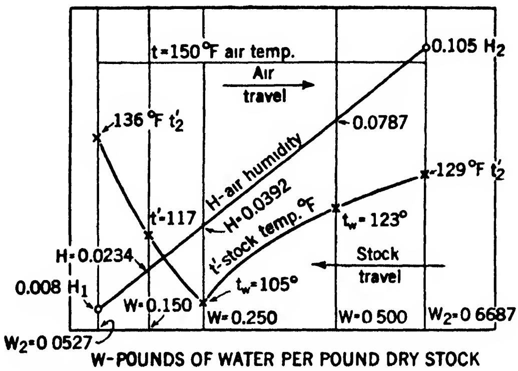

FIG. 9. TEMPERATURE HUMIDITY RELATIONS IN A DRIER

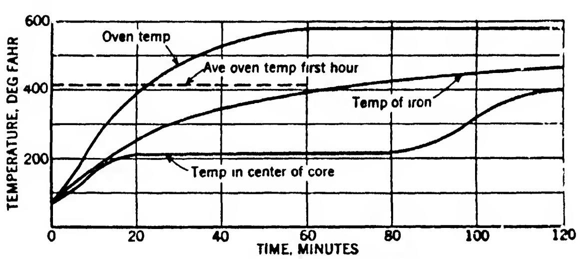

FIG. 10. CORE DRYING TIME TEMPERATURE RELATIONS

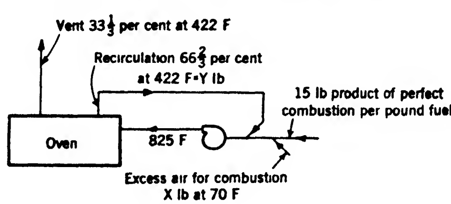

FIG. 11. CORE DRYING DIAGRAM OF COMBUSTION PRODUCTS AND AIR

contact, and is found experimentally to maintain wet-bulb temperature until the moisture content has fallen to 20 per cent. From this point its temperature rises progressively as it dries. In this range the difference in temperature between stock and air, divided by the wet-bulb depression, may be assumed proportional to the moisture content.

The moisture content of the entering stock, in the units here employed, is:

$$w_1 = \frac{40 \text{ per cent water}}{60 \text{ per cent dry stock}} = 0.6667: \; w_2 = \frac{5 \text{ per cent water}}{95 \text{ per cent dry stock}} = 0.0527$$

$w_1 - w_2 = \Delta w = 0.614$ lb water evaporated per pound of dry stock. Since the air leaving the drier is 50 per cent saturated at 150 F from Fig. 4, $H_2 = 0.105$. Similarly, $H_1 = 0.008$, corresponding to 50 per cent humidity at 70 F. Consequently $H_2 - H_1 = \Delta H = 0.097$ lb water evaporated per pound dry air.

Inspection of equation (2) shows that (H) is linear in w. Hence, one can construct on Fig. 9, the line marked (H) being drawn connecting the initial and final points just computed.

Since the air leaving the drier has a temperature of 150 F and a humidity of 0.105, Fig. 4 shows that its wet-bulb temperature is 129 F. This is plotted at the right hand side of Fig. 9. Since the stock maintains a wet-bulb temperature down to 20 per cent moisture, where $w = 0.25$, the corresponding humidity can be computed by the use of equation (2) or by reading directly from the diagram, the value being 0.0392. Fig. 4 shows that the corresponding wet-bulb temperature is 105 F. Any intermediate point on the wet-bulb temperature curve can be calculated similarly. The points for $w = 0.5$ are shown in Fig. 9.

Below the point, $w = 0.25$, the temperature of the stock begins to rise appreciably above the wet-bulb temperature. Its temperature at any given point in this range, for example at $w = 0.15$, may be computed as follows: At this point, $H = 0.0234$ (from equation (2)) and from Fig. 4, $t_w = 95$ F. Hence the wet-bulb depression, $t - t_w = 150 - 95 = 55$ F. The assumption made regarding the relation between stock temperature and moisture content in this range may be formulated:

$$\frac{\Delta t'}{t - t_w} = \frac{w}{0.25}$$

At the point $w = 0.15$, $\Delta t' = 33$ F, $t' = 117$ F. The temperature of the stock leaving the drier, similarly computed, is 136 F.

Fig. 9 thus computed gives in graphical form the information as to the temperature humidity relationships in the drier. The air requirements can be computed by equation (2). Thus, per 100 lb of dry stock, it is necessary to supply 633 lb of dry air. Furthermore, since from Fig. 4 it is seen that the volume of 50 per cent saturated air at 70 F, is 13.55 cu ft per pound; 8580 cu ft of room air must be supplied per 100 lb dry stock. Similarly, since the volume of 50 per cent saturated air at 150 F is 18.0 cu ft per pound, the volume of hot wet air discharged from the drier is 11,400 cu ft per 100 lb of dry stock. Finally, the heat necessary to supply to the drier, as a whole, or to any section of it, may be computed from equation (3).

High Temperature Drier

In the design of a high temperature drier unit a method of approach to the necessary calculations involved are outlined as follows:

Example 1. Cores 4 and 5 in. thick are to be dried by heating to a temperature at 400 F. An intermittent type box oven is to be used, size 12 x 14 x 10 ft with 856 sq ft surface having an average heat transfer of 0.3 Btu per square foot per degree per hour. Drying time as determined by test is 2 hr (Fig. 10) Cores weighing 6 tons, and 15-ton steel plates, trucks etc. are delivered to the drier at 70 F. The oven is heated by an external heater; the products of combustion and 66⅔ per cent recirculated air will be delivered to the oven at 825 F. Fuel oil of 19,980 Btu gross and 18,830 Btu per pound net heating value, weighing 6.75 lb per gallon and having 15 lb product per pound fuel

for perfect combustion. Cores consist of 91 per cent sand, 3 per cent oil binder, and 6 per cent water.

Solution. Heat required per ton of cores:

	Lb Material × Temp. Rise × Sp. Ht.	=	Btu
Sand	0.91 × 2,000 × (400 − 70) × 0.2	=	120,120
Binder	0.03 × 2,000 × (400 − 70) × 0.4	=	7,920
Water heating	0.06 × 2,000 × (212 − 70) × 1.0	=	17,040
Water evaporation	0.06 × 2,000 × 970 (Fig. 4)	=	116,520
Water superheating (approx. 50 per cent reaches 575 F)	= 0.5 × 0 06 × 2,000 × (575 − 212) × 0.45	=	9,800
Total Heat			271,400 Btu

HEATING LOAD FIRST HOUR

	HEATED TO			BTU
Sand	212 F	$\frac{142}{330}$ × 120,120	=	51,688
Binder[a]	212 F	$\frac{142}{330}$ × 7,920	=	3,408
Water	212 F		=	17,040
Evaporation	66.7%	0 667 × 116,520	=	77,680
Superheat	66.7%	0.667 × 9,800	=	6,530
Total Per Ton				156,346
For 6 ton		6 × 156,346	=	938,076
Steel plates	390 F	320 × 30,000 × 0.12	=	1,152,000
Radiation[b]	422 F Avg.	352 × 856 × 0 30	=	90,394
Total				2,180,470

HEATING LOAD SECOND HOUR

Sand	400 F	$\frac{188}{330}$ × 120,120	=	68,432
Binder[a]	400 F	$\frac{188}{330}$ × 7,920	=	4,512
Water				
Evaporation	33 3%	0 333 × 116,520	=	38,840
Superheat	33.3%	0.333 × 9,800	=	3,270
Total Per Ton				115,054
For 6 ton		6 × 115,054	=	690,324
Steel plates	460	70 × 30,000 × 0.12	=	252,000
Radiation[b]	575	505 × 856 × 0.30	=	129,684
Total				1,072,008

[a]Binder oxidizes and liberates heat, which is neglected in this calculation.

[b]Average value of coefficient is less than 0 3 because oven is not up to 575 F. This is neglected. 422 F is arrived at by taking area under curve as compared to area under 575 F ordinate.

Heat in 1 lb fuel oil	=	18,830 Btu
Heater Loss (10 per cent)	= 1883	
Duct Loss (5 per cent)	= 942	2,825 Btu
		16,005 Btu available to heat oven.

Heat content of gases in 1 lb fuel oil at 825 F is 205 Btu (Fig. 8)

15 lb × 205	=	3,075 Btu sensible heat in products of perfect combustion.
		12,930 Btu to heat air X and Y (Fig. 11).

$$Y\,(S_{825} - S_{422}) + X\,(S_{825} - S_{70}) = 12930 \qquad (4)$$

$Y = 2\,(X + 15)$ for 66.7 per cent recirculation.

where

S = heat content of air at temperature noted taken from Fig. 8

(Recirculation and exhaust contains water vapor, products of combustion, and a greater portion of air. Heat capacities of all vary so little that they have all been assumed to be air).

$S_{825} - S_{422} = 190 - 91 = 99$

$S_{825} - S_{70} = 190 - 8.6 = 181.4$

Substituting values of Y, H, etc. in Equation 4,

$(2\,X + 30)\,99 + 181.4\,X = 12{,}930$

$X = 26.3$ lb excess air.

$Y = 82.6$ lb recirculating air.

Total = 26.3 + 82.6 + 15 = 123.9 lb air and products of combustion circulated per pound fuel burned.

Heat in air exhausted from oven at 422 F per pound fuel burned = $0.333 \times 123.9 \times (S_{422} - S_{70}) = 41.3\,(91 - 8.6) = 3{,}400$ Btu.

Btu available for heating material = 16,005 − 3,400 = 12,605 Btu per pound fuel.

Fuel used in first hour = 2,180,470 ÷ 12,605 = 173 lb = 25.6 gal.

During the second hour the heater capacity will be much greater than required. If an automatic oven temperature control operates on the oil supply, the delivery temperature of the air entering the oven and the quantity of oil burned will decrease, the air supply being constant.

Heat in air exhausted = $41.3\,(S_{675} - S_{70}) = 41.3\,(127 - 8.6) = 4{,}880$ Btu per pound fuel.

Heat available for heating material = 16,005 − 4,880 = 11,125 Btu.

Fuel used in second hour = 1,072,008 ÷ 11,125 = 96.5 lb oil = 14.3 gal

Total oil used per load = 25.6 + 14.3 = 39.9 gal.

ESTIMATING METHODS

Values based on practical experience are available for rough estimating of drying problems. The temperature will drop approximately 8.5 F per grain of water evaporated per cubic foot of air (measured at 70 F) or approximately 0.62 F per pound of air at any temperature. Air will drop 55 F per cubic foot for each Btu extracted. Generally air will absorb from 2 grains to 5 grains per cubic foot of air in one passage through an air drier, depending on the temperature and the degree of contact with the material. The amount of steam required to evaporate a pound of water will vary from 1.5 lb to a more usual figure of from 2.5 to 3 lb of steam per pound of water evaporated.

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PROBLEMS IN PRACTICE

1 ● What makes a commercial adiabatic drier differ from a theoretical one?

The word *adiabatic* means no heat lost to the outside and that the sensible heat lost by the air is equal to the latent heat of the water evaporated. In an actual drier, the solid containing the water, and the water itself must be heated to the temperature of evaporation, before evaporation can begin. Radiation losses from the drier enclosure is the other factor causing deviation from the theoretical adiabatic process.

2 ● What is a zone drier?

This term refers to a continuous drier where the drying medium is divided into two or more sections, in order to have better control of the temperature and humidity gradients through the drier, and often different velocities.

3 ● If a material enters a drier containing 70 per cent water and 30 per cent solids, and leaves the drier with 10 per cent water and 90 per cent solids, (a) What is the evaporation per pound of dried product? (b) What is the evaporation per pound of bone dry material?

a. $\frac{90}{30} - 1 = 2$ lb water per pound dried product.

b. Water entering $= \frac{70}{30} = 233$ per cent on bone dry basis.

Water leaving $= \frac{10}{90} = 11$ per cent on bone dry basis.

Water evaporated 222 per cent on bone dry basis.

Evaporation $=$ 2 22 lb water per pound bone dry material.

4 ● What items must be included in a calculation of the drier heat requirements?

a. Heating water to be evaporated from the entering temperature to the temperature of evaporation.

b. Evaporating water to be removed.

c Superheating evaporated water from the temperature of evaporation to the exit temperature of the air.

d. Heating material from entering to leaving temperatures

e. Heating residual water from the entering to the leaving temperatures.

f. Heating conveyor or other supporting materials.

g. Radiation losses through the enclosure.

h. Sensible heat in the exit air.

5 ● The following conditions prevail in a drier; 250 lb water evaporated per hour. Air enters heater at 80 F dry-bulb and 65 F wet-bulb. Air exhausted from drier at 130 F dry-bulb and 100 F wet-bulb. Stock enters drier at 70 F. Heat required for warming stock and radiation losses are not considered. Fan is located ahead of heater. Find conditions of air entering and leaving drier, volume handled by fan, and temperature of air entering drier to supply the necessary heat, using Humidity Chart in Fig. 4.

Entering Air: Humidity, $H = 0.01$ lb water vapor per pound dry air.
Dew-point $=$ 57 F.
Per Cent Humidity, $\% H = 46$.

Leaving Air: Humidity, H = 0.0355 lb water vapor per pound dry air.
Per Cent Humidity, $\% H$ = 32.

Water pick up = 0.0355 − 0.01 = 0.0255 lb per pound bone dry air.

Bone dry air circulated per hour = 250 ÷ 0.0255 = 9800 lb.

Volume of air circulated at 80 F dry-bulb, and 46 per cent humidity
14.1 − 13.6 = 0.5 cu ft vapor (Fig. 4).

Volume = 13.6 + (0.46 × 0.5) = 13.87 cu ft = 1 lb dry air + vapor.

Volume handled by fan at 80 F = $\frac{9800 \times 13.87}{60}$ = 2260 cfm.

Btu received by water = (130 − 70) × 1.0 = 60
Latent heat of steam at 130 F (Fig. 4) = 1019
Total = 1079 Btu per pound.

Heat used for evaporation per pound dry air = 1079 × 0.0255 = 27.43 Btu.

For entering air: Humidity, H = 0.01, Humid Heat = 0.2425 Btu per pound (Fig. 4)

$$(t_1 - t_2) \times S = \text{Btu for evaporation}$$
$$(t_1 - 130) \times 0.2425 = 27.43$$
$$t_1 = 247 \text{ F}$$

6 ● Given the following conditions, air 160 F dry-bulb, 49.6 per cent relative humidity (Φ), 29.92 in. Hg, barometric pressure, find the per cent H, absolute humidity.

For 160 F, p_s = 9.65 in. Hg (From Table 6, Chapter 1)

$$p = \Phi p_s = 0.496 \times 9.65 = 4.78$$

$$\frac{29.92 - 9.65}{29.92 - 4.78} \times 0.496 = 0.40 \text{ or 40 per cent absolute humidity.}$$

Chapter 36

NATURAL VENTILATION

Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces, supplemented in certain cases by wind-actuated devices finds application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for the displacement of air in buildings are, (*a*) wind forces, and (*b*) the difference in temperature between the air inside and outside the building, or a combination of the two. The results that are obtained by natural ventilating systems are variable, as they depend on wind action and temperature difference. The arrangement and control of ventilating openings should be such that the two forces act cooperatively and not in opposition.

WIND FORCES

In considering the use of natural wind forces for the operation of a ventilating system, account must be taken of (1) average and minimum wind velocities, (2) wind direction, (3) seasonal, daily and hourly variations in wind velocity and direction, and (4) local wind interference by buildings, trees, etc.

Table 1, Chapter 8, gives values for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 2, Chapter 7, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While average wind velocities are seldom below 5 mph, there are many hours in each month during which the wind velocity is from 3 to 5 mph, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the hourly wind velocity falls much below 3 mph for more than 10 daylight hours per month. Usually a natural ventilating system should be designed to operate satisfactorily with a wind velocity of 3 to 6 mph, depending on locality.

The following formula may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings:

$$Q = EAV \tag{1}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlet (or outlet) openings, square feet
V = wind velocity, feet per minute,
= miles per hour $\times$ 88.
E = effectiveness of openings.

(E should be taken at from 50 to 60 per cent if the inlet openings face the wind and from 25 to 35 per cent if the inlet openings receive the wind at an angle)

If outlet openings, where air leaves a building, are smaller than inlet openings, where air enters a building, the air will be less effective than indicated by the constant E.

The accuracy of the results obtained by the use of Formula 1 depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edge orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less, and if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the following four places:

1. On the side of the building directly opposite the direction of the prevailing wind
2. On the roof in the low pressure area caused by the jump of the wind (see Fig 1).
3. In a monitor on the side opposite from the wind
4. In roof ventilators or stacks exposed to the full force of the wind[1].

Forces Due to Stack Effect [2]

The stack effect produced within a building when the outdoor temperature is lower is due to the difference in weight of the warm column of air within the building and the cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\,A\,\sqrt{H\,(t - t_o)} \tag{2}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlets or outlets (assumed equal), square feet.
H = height from inlets to outlets, feet.
t = *average* temperature of indoor air in height H, degrees Fahrenheit.
t_o = temperature of outdoor air, degrees Fahrenheit.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

The height between inlets and outlets should be the maximum which the building construction will allow.

[1]Airation of Industrial Buildings, by W C Randall (A S H V E TRANSACTIONS, Vol 34, 1928, p 159).
[2]Neutral Zone in Ventilation, by J E. Emswiler (A.S.H V E TRANSACTIONS, Vol 32, 1926, p 59)
Predetermining Airation of Industrial Buildings, by W C Randall and E W Conover (A S H V E. TRANSACTIONS, Vol. 37, 1931, p 605)

TYPES OF OPENINGS

Types of openings may be classified as: (1) windows, doors, monitor openings and skylights, (2) roof ventilators, (3) stacks connecting to registers, and (4) specially designed inlet or outlet openings.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in

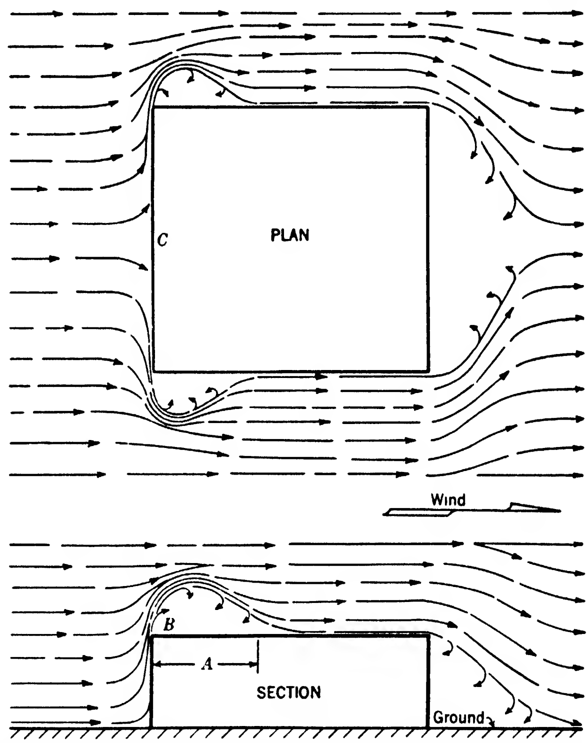


FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (*A*—LENGTH OF SUCTION AREA; *B*—POINT OF MAXIMUM INTENSITY OF SUCTION; *C*—POINT OF MAXIMUM PRESSURE)

various ways; they may open by sliding as in the ordinary double-hung windows, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top, bottom or side.

The proper distribution of the air in spaces to be ventilated is as important as that of sufficient air quantity. Advantageous pivoting of sash is very useful for securing good air distribution. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet. If it is of a type which is sensitive to wind action addi-

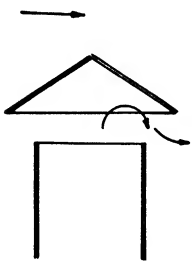


FIG. 2

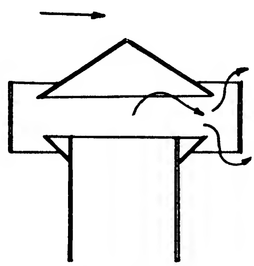


FIG. 3

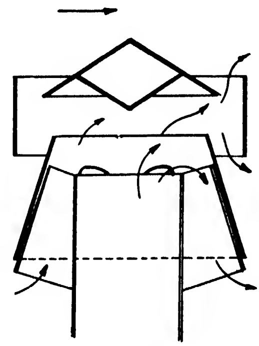


FIG. 4

TYPES OF STATIONARY VENTILATORS

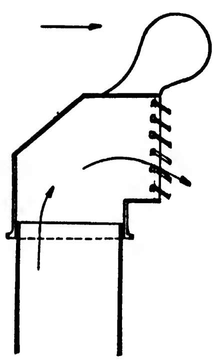


FIG. 5

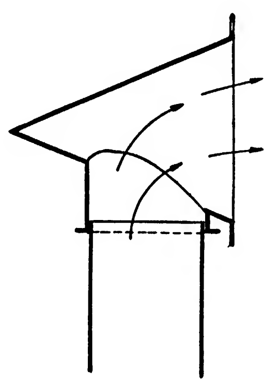


FIG. 6

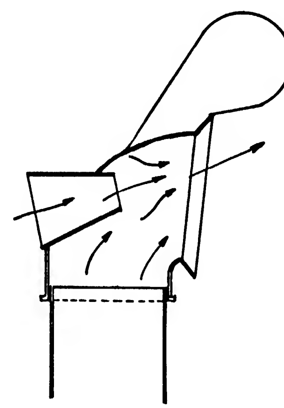


FIG. 7

OSCILLATING VENTILATORS

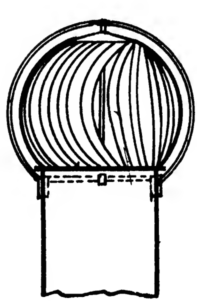
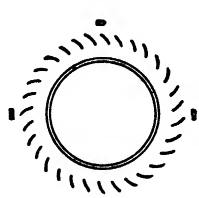


FIG. 8



Section

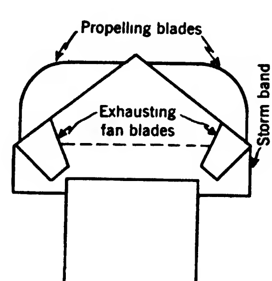


FIG. 9

ROTATING VENTILATORS

tional flow capacity will be produced. The capacity of a ventilator at a constant wind velocity and temperature difference, depends upon four things: (1) its location on the roof, (2) the resistance it and the duct work offers to air flow, (3) the height of draft, and (4) the efficiency of the ventilator in utilizing the kinetic energy of the wind for inducing flow by centrifugal or ejector action.

For maximum flow induction, a ventilator should be located on that part of the roof which receives the full wind without interference. (See Fig. 1.) This does not mean that any ventilators are to be installed within the suction region created by the wind jumping over the building, or in a light court, or on a low building between two high buildings. Ventilators are highly effective in such low-pressure areas, but their ejector action, caused by wind velocity, is of little value in these locations, and hence their size should be increased proportionally.

The base of the ventilator should always be provided with a taper-cone inlet in order to produce the effect of a bell-mouth nozzle (flow coefficient 0.97) rather than that of a square-entrance orifice (flow coefficient 0.60). If a grille is provided at the base of a ventilator it should be oversized as compared with the ventilator size.

Air inlet openings located at lower levels in the building should be at least equal to, and preferably larger than the combined throat areas of all roof ventilators. The air discharged by a roof ventilator depends on wind velocity and temperature difference, but due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Several types of roof ventilators are shown in Figs. 2 to 9. These may be classified as *stationary*, Figs. 2 to 4, *pivoted* or *oscillating*, Figs. 5 to 7, or *rotating*, Figs. 8 and 9. When selecting roof ventilators, some attention should be paid to ruggedness of construction, storm-proofing features, dampers and damper operating mechanisms, possibilities of noise from dampers or other moving parts, and possible maintenance costs.

Natural ventilation units may be used to supplement power-driven supply fans, and under favorable weather conditions it may be possible to shut down the power-driven units. Where low operating costs are very important, such a combination has great advantages.

Controls

Gravity ventilators may have dampers controlled by (1) hand, (2) thermostatic, and (3) wind velocity, in combination with a fan. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks or vertical flues are really chimneys and utilize both the inductive effect of the wind and the force of temperature difference (the so-called

gravity action). Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction.

With little or no wind, chimney effect depending on temperature difference and lower outdoor temperature will produce a removal of air from the rooms where the inlet openings are located.

HEAT REMOVAL

In problems of heat removal, knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain this temperature difference can be determined by means of the following equation:

$$Q = \frac{VH}{c\,60\,(t - t_o)} \tag{3}$$

where

c = 0.24 = specific heat of air.
V = specific volume of the air, cubic feet per pound, about 13.5. (See Chapter 1.)
H = heat to be carried off, in Btu per hour.
Q = air flow in cubic feet per minute.
t = inside temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.

For disposing of odors or other air impurities, the amount of outside air to be introduced must be of such quantity to dilute the impurities to a degree that they are no longer objectionable. See Chapter 3 for the minimum of outside air necessary for ventilation. For garage ventilation, sufficient air must be admitted to dilute the carbon monoxide content of the indoor air to 1 in 10,000 (see Garage Ventilation in this Chapter).

Suggested methods for estimating the air flow due to temperature difference alone and to wind alone have already been given. It must be remembered that when both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The same openings have been assumed in both cases, and since the resistance to flow through the openings varies approximately with the square of the velocity[3], this resistance becomes a limiting factor as the flow through the openings is increased.

Recent investigations[4] show that the total flow is only 10 per cent above the flow caused by the greater force when the two forces are nearly equal, and this percentage decreases rapidly as one force increases above the other. Tests on roof ventilators indicate that this is too conservative in the direction of low total flow quantities, but there is in any case a large judgment factor involved. The wind velocity and direction, the outdoor temperature, or the indoor activities cannot be predicted with certainty, and great refinement in calculations is therefore not justified. When designing for winter conditions, an added variable is the heat lost by direct flow through walls and windows and by infiltration.

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[3]Loc Cit Notes 1 and 2

[4]This is true for *turbulent* flow only It would be more correct to state that the resistance varies approximately with V^2 for high to moderate velocities, with $V^{1.8}$ for moderate to low velocities, and with the first power of the velocity for very low velocities through small openings

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per lb is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Temperature differences are 10 F in summer and 30 F in winter, and the wind velocity is 5 mph in summer and 8 mph in winter. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution. The system must be designed for the summer conditions as these are the more severe. The heat to be removed per hour is:

$$H = 15 \times 7.75 \times 18{,}000 = 2{,}092{,}500 \text{ Btu.}$$

By Equation 3, the air flow required to remove this heat with a temperature difference of 10 deg is:

$$Q = \frac{VH}{c\,60\,(t - t_o)} = \frac{13.5 \times 2{,}092{,}500}{0.24 \times 60 \times 10} = 196{,}172 \text{ cfm.}$$

This is equal to 19 6 air changes per hour The assumption is made that the average temperature difference between indoors and outdoors is the same as the temperature rise of the air from the inlet opening to the outlet opening Actually, the latter difference is larger and so the value of 19.6 air changes per hour is conservative as it allows for more cooling than is necessary for an *average* temperature difference of 10 deg.

If 196,172 cfm are to be circulated by the force of the temperature difference alone, the area of opening would be, by Equation 2:

$$A = \frac{Q}{9.4\sqrt{H\,(t - t_o)}} = \frac{196{,}172}{9.4\sqrt{30 \times 10}} = 1{,}205 \text{ sq ft.}$$

If this area of openings were provided, a wind velocity of 5 mph, acting alone, would produce a flow according to Equation 1, of:

$$Q = EAV = 0.50 \times 1{,}205 \times 5 \times 88 = 265{,}100 \text{ cfm.}$$

If the inlet openings do not face the wind, but are at an angle with it, about half this amount may be considered to flow.

A factor of judgment must now be exercised in making the selection of the area of openings to be specified. Apparently 1205 sq ft are a very generous allowance because either a direct wind of 5 mph or an average temperature difference of 10 deg acting alone will more than suffice to carry away the heat, and when the two forces are acting together, the system may have an excess capacity of 25 per cent to 50 per cent, especially if the outlets are made up partially of roof ventilators which employ the force of the wind for producing a suction effect. On the other hand, the wind may at times come from an unfavorable direction, or its velocity may fall below 5 mph or the building construction may not permit a full 2400 sq ft of inlet window area and an equal amount of monitor or roof ventilator outlet area. In case the two sets of openings are not equal, their effectiveness is reduced.

From this example, it must be apparent that while formulas may furnish a reliable guide, the final solution of a problem of natural ventilation requires a common sense analysis of local conditions to supplement and to modify the dictates of the formulas.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1. Inlet openings in the building should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top Outside air will then be supplied to the zone to be ventilated.

2. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

3. Roof ventilators should be located 20 to 40 ft apart each way and preferably on the ridge of the roof. The closer spacings are used when ventilating rooms with low ceilings.

4. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

5. In an industrial building where furnaces, that give off heat and fumes, are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas laden air.

6. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

7. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.

8. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.

9. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.

10. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building together with amount and grouping of ventilation openings can be readily arranged to take full advantage of the force of the wind. On the other hand, where the direction of the wind is quite variable, it may be stated as a general principle that windows should be arranged in sidewalls and monitors so that there will be approximately equal area on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force of the wind, and others opposed to a suction force, and effective movement through the building will be assured.

11. The intensity of suction or the vacuum produced by the jump of the wind is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when changes in occupancy are possible, or to provide for extremely hot days. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. Special consideration should be given to the possibility of sidewall or monitor windows being closed on account of weather conditions. Such possibilities favor roof ventilators and specially designed stormproof inlets.

MEASUREMENT OF NATURAL AIR FLOW

The determination of the performance of any ventilating system involves measurements which are not easy to make. The difficulties are increased in the case of natural ventilation, since the motive forces and the air velocities are very small. The measurements necessary for giving the *capacity* of a system are (1) velocity of the wind, (2) velocity of the air through inlet and outlet openings, (3) outdoor air temperature, and (4) average indoor air temperature. (See Chapter 44).

DAIRY BARN VENTILATION[5]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume is within the limit that can be heated by the stabled animals, the air supply need not be heated. The air should be supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cfh of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per hour per cubic foot of barn space, and 0.197 to 0.305 Btu per hour per square foot of barn exposure.

GARAGE VENTILATION

On account of the hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means particularly during the mild weather when doors and windows can be kept

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[5]Dairy Barn Ventilation, by F. L. Fairbanks (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 181).

Cow Barn Ventilation, by Alfred J. Offner (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 149).

For additional information on this subject refer to *Technical Bulletin, U. S. Department of Agriculture* (1930), by M. A. R. Kelley.

open. However, the A.S.H.V.E. Code for Heating and Ventilating Garages, adopted in 1929 and revised in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.[6]

Research

Research on garage ventilation undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, Lawrence, Kans., in cooperation with the A.S.H. V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory has resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed in the following statements:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than is had with complete mixing of the exhaust gases and the air supplied However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cfh, with an average rate of 35 cfh.

5. An air change of 350,000 cfh per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

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[6]Code for Heating and Ventilating Garages (A S H V E. TRANSACTIONS, Vol 35, 1929, p. 355), (A.S. H V E Reprint, January, 1935).

Airation Study of Garages by W C Randall and L. W. Leonhard (A S H V E TRANSACTIONS, Vol 36, 1930, p 233)

A.S H.V E RESEARCH REPORT No 874—Carbon Monoxide Concentration in Garages, by A S Langsdorf and R R Tucker (A.S H V E TRANSACTIONS, Vol 36, 1930, p 511).

A S H V E RESEARCH REPORT No 935—Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A S H V E TRANSACTIONS, Vol. 38, 1932, p 439).

A S H V.E RESEARCH REPORT No 934—Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A S H V E TRANSACTIONS, Vol 38, 1932, p. 424).

A.S H V E. RESEARCH REPORT No 967—Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F C Houghten and Paul McDermott (A S H V E. TRANSACTIONS, Vol 39, 1933, p. 395).

Carbon Monoxide Surveys of Two Garages, by A H. Sluss, E. K. Campbell and Louis M Farber (A.S.H V.E. TRANSACTIONS, Vol. 40, 1934, p. 263).

PROBLEMS IN PRACTICE

1 ● What factors may make the adoption of a system of ventilation depending upon wind movement inadvisable in new construction?

a. Variation in direction of wind.

b. Variation in wind velocity.

c. Inability to clean incoming air.

d. Inability to control location, size and shape of buildings on adjacent property.

e. Unsatisfactory warming of incoming air during cold weather.

2 ● a. What factors are important in the location and control of ventilating openings?

b. What types of ventilating openings are best suited to a proper distribution of the air supplied?

a. The proper distribution of air as required by the occupants and usage, and the best utilization of natural ventilating forces. The general rules referred to in this chapter apply particularly to these factors.

b. Windows with swinging sash and openings with deflectors may be used to direct air to the points desired.

3 ● a. What is the best location for ventilating openings?

b. How are the sizes of ventilating openings determined for proper air supply?

a. Inlet openings should be low and facing the prevailing winds where possible. Outlet openings should be high and on the side opposite the prevailing winds.
b. For simple openings use Formula 1:

$$Q = EAV$$

and for stacks use Formula 2:

$$Q = 9.4\,A\,\sqrt{H\,(t - t_o)}$$

The use of these formulae is illustrated in Example 1 of the text of this chapter. Inlet and outlet areas should be approximately the same for best results.

4 ● a. What are the advantages of roof ventilators?

b. How are proper sizes determined for roof ventilators?

a. Roof ventilators offer the best utilization of the inductive force of the wind, and they may be very economically fitted with built-in fans to supply the necessary circulation when the force of the wind is not sufficient.

b. Because of the many factors affecting the flow through roof ventilators no accurate formula can be given. It is usual practice to make the combined throat area of all roof ventilators between one-half area and full area of the air inlets as determined by Formula 1.

5 ● What methods of control are used in ventilating systems?

Hand control, control by a thermostat located in the ventilated space or in the ventilator, or wind velocity control designed to keep the air discharge constant regardless of wind velocity.

6 ● How is the quantity of air required for a building determined?

Sufficient air must be supplied to carry away the heat and impurities generated within a building. The temperature rise and concentration of impurities in the exhaust air must be held within specified limits. (See Example 1 in the text of this chapter).

7 ● What measurements are necessary to determine the capacity of a ventilating system?

Wind velocity and air velocities through openings, determined by suitable anemometers; outdoor air temperatures, measured by a shaded thermometer not near objects

heated by the sun or near exhaust air openings; indoor air temperatures, measured at various heights to secure a good average.

8 ● How much air must be supplied for dissipating the heat generated in a dairy barn housing 100 cows if the outside temperature is 20 F and the inside temperature is to be maintained at 45 F?

The total heat generated is 100 × 3000 = 300,000 Btu per hour. Then from Formula 3,

$$Q = \frac{VH}{c\ 60\ (t - t_o)}$$

$$= \frac{13.5 \times 300{,}000}{0.24 \times 60\ (45 - 20)}$$

$$= 11{,}250 \text{ cfm.}$$

This amount of air should also keep down humidity and odors.

9 ● a. What precaution is necessary in the ventilation of garages using natural ventilation?

b. How much window area is required for a garage with 50 x 100 sq ft floor area if natural ventilation is used?

a. The carbon monoxide content of the air should be kept below 1 part in 10 000 and windows should be kept open at all times.

b. The window area should aggregate 5 per cent of the floor area

$$0.05 \times 50 \times 100 = 250 \text{ sq ft of window area.}$$

This area should be evenly distributed along two sides of the building.

Chapter 37

AUTOMATIC CONTROL

Purpose of Automatic Control, Definitions of Control Units and Terms, Types of Control, Central Fan Systems, Unit Systems, Control of Automatic Fuel Appliances, Residential Control Systems, Control of Refrigeration Equipment, Industrial Processes

THIS chapter is prepared with the purpose of acquainting the engineer with the principles underlying the use of automatic control, the general types and varieties of control equipment available and their application.

Automatic control, properly applied to heating, ventilating and air conditioning systems, makes possible the maintenance of desired conditions with maximum operating economy. A properly designed and complete control system has the ability to interlock and coordinate the various functions of heating, ventilating and air conditioning in a manner impossible to accomplish with manual regulation.

Automatic control is an integral and essential part of a heating, ventilating or air conditioning installation and cannot be regarded as an accessory. In order to insure satisfactory results, the control should be designed with and incorporated in the heating, ventilating or air conditioning system. The control equipment should be given careful consideration in the planning of any installation in order that the entire system may operate together with satisfactory results.

In order that proper selection and application of controlling devices may be made it is important that a broad understanding exist as to the types of control available and their principles of operation. Improper selection and application of control equipment will result in unsatisfactory and inefficient operation. Specific control devices and systems are described in the *Catalog Data Section.*

PURPOSE OF AUTOMATIC CONTROL

Automatic control is normally applied to heating, ventilating or air conditioning systems:

1. To insure the maintenance of certain desired or required conditions of temperature, pressure, humidity, air motion or air distribution.

2. To serve a safety function, limiting pressures or temperatures within predetermined points, or preventing the operation of mechanical equipment unless it may function without hazard.

3. To produce economical results and thereby insure operation of the system at a minimum of expense.

DEFINITIONS OF CONTROL UNITS AND TERMS

Controlling devices and terms commonly used in the automatic control of heating, ventilating and air conditioning systems are:

Thermostats: Thermostats are defined as temperature sensitive devices reacting to temperature changes. There are four major types of thermostats.

A *Room Thermostat* is normally installed on the wall of the room whose temperature it is to control, and in reacting to rising or falling temperatures, the thermostat causes the operation of heating or cooling equipment so that desired temperatures will be maintained.

The temperature sensitive element will usually consist of a bi-metal strip or coil, or a vapor-filled bellows as illustrated in Fig. 1.

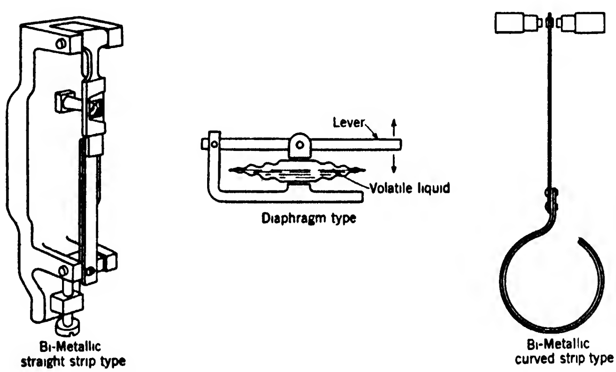


FIG. 1. TYPICAL THERMOSTATIC ELEMENTS

Immersion Thermostats are used for controlling liquid temperatures. The sensitive element will normally be encased in a protective well which is inserted in the liquid, the temperature of which is being controlled.

The temperature sensitive element will usually consist of a bi-metal coil, thermal expansion rod, or a vapor-filled system. If the latter is used the temperature sensitive bulb may be connected to the case of the instrument by either a flexible or rigid tube.

Insertion Thermostats are similar to immersion thermostats except that they are for use in controlling the temperature of a gas such as air. The sensitive element will often be encased in a protective well which prevents mechanical damage but which permits the gas to come in direct contact with the element.

Surface Thermostats include those devices which measure surface temperatures. These surface temperatures will often be an indirect measure of the temperature of a gas or fluid as in the case of a pipe within which water is flowing. The sensitive element will usually be placed in direct contact with the surface of the object whose temperature it is to measure and may consist of a bi-metal spiral or vapor-filled bellows.

Humidity Controls: Humidity controls are defined as automatic devices reacting to changes in relative humidity. Within this group, the devices which operate in controlling humidity supplying equipment are regulating devices and when operating only to prevent relative humidity from exceeding a predetermined maximum are a form of limit control.

The humidity sensitive element may consist of hair, paper, wood, skin or any other material which changes its dimensions with changes in humidity.

Controls are available provided with both temperature and humidity sensitive elements, which operate to maintain definite relations between dry-bulb temperature and relative humidity.

Pressure Controllers: Pressure controllers are defined as devices reacting to pressure and pressure changes. Examples of such devices are the pressure controls governing the operation of refrigeration equipment from either head or suction pressure, devices reacting to steam or water pressure or the pressure of air in the distribution systems.

Damper Motors: Damper motors are defined as specialized power units, the purpose of which is to position outdoor air, face, by-pass or distribution dampers, regulating the flow of air through the system. Connected by suitable linkages, these damper motors react at the command of thermostats, humidity controllers and pressure controllers to adjust the air flow to the needs of the system.

Control Valves: Control valves are defined as steam valves, water valves or air valves which may be adjusted at the command of controllers to regulate the flow of the medium passing through them to the needs of the system. Such control valves are usually constructed with an electric or pneumatic power unit connected to the valve stem so that the movement of the power unit will react to position the valve as conditions demand.

Self-contained valves are also included under this classification. Their application is principally limited to the regulation of the steam supply to individual radiators in two-pipe low pressure steam heating systems, and the temperature of hot water supply tanks.

Solenoid Valves: Solenoid valves are, as their name implies, valves actuated by the magnetic effect of an electric solenoid built within them. While normally these valves are opened when the solenoid is energized, they are sometimes built in a reverse acting manner and closed when energized. In heating, ventilating and air conditioning systems, they are normally adapted to the control of oil or gas burners as fuel valves, as water valves on humidifiers, or as refrigerant valves in refrigeration systems.

Relays: A relay is defined as a unit installed between a controller and the device under control, for purposes of amplifying the capacity of the controller or performing an auxiliary control function. For example, a thermostat, in order to preserve its sensitivity may be constructed so that it is not capable of handling the power required of a motor. A relay is, therefore, installed between the two. The thermostat actuates the relay and the relay, in turn, actuates the motor. Motor-driven switching devices are also often used as relays.

TYPES OF AUTOMATIC CONTROL

Operating Media or Source of Power Supply

Automatic control systems may be classified in three broad groups based upon their primary operating media or source of power, as follows:

1. *Electric Control Systems.* In such control systems the primary medium utilized to provide for the operation is electricity, and the basic function of these controls consists of switching or otherwise adjusting electric circuits to govern electric motors, relays or solenoids. The individual units of this type of system are interconnected by line voltage or low voltage wiring, and this wiring serves to complete the circuits carrying the commands of the controllers to the controlled valves or damper motors.

2. *Pneumatic Control Systems.* In the pneumatic control systems, the primary source of operation is obtained through a medium of compressed air, the pressure of which is varied by the controlling devices. In these systems one or more centrally located air compressors furnish a supply of compressed air which is distributed in special piping to the various controlling and controlled devices. By means of leak ports or orifices, the pressure of the air is varied in the branch lines and the changing pressures are utilized in air operated damper motors or valves to obtain the movement necessary to the operation of valves and dampers.

3. *Self-Contained Control Systems.* In self-contained control systems, the primary source of operation is the vapor pressure of a volatile liquid in the closed thermal system of the controller, which is increased or decreased in direct proportion to variation of the temperature in the controlled medium. These pressure changes are transmitted directly to the control valve or damper motor. Applications consist of valves or dampers to regulate the flow of heating or cooling media to coils, radiators, or liquid tanks, as determined by the controller element.

Motion of Controlled Equipment

Automatic control equipment can also be classified into two general types with respect to the characteristics of the motion imparted by the controls to the controlled equipment, such as two position or positive-acting control and modulating or graduated-action control.

In any control system it is necessary to choose the type of equipment whose characteristics permit the type of control operation desired and in many cases both types of control are used in the same system to best meet various requirements.

1. *Two Position or Positive-acting Control.* This type of control operates positively between two positions such as *on and off* or *open and closed* with no intermediate positions or degrees of motion between the two extremes of operation. A simple thermostat which starts and stops an oil burner or a unit heater motor is an example of this type. As applied to a valve or a damper, the action of the controlling device would serve to fully open or fully close the valve or damper.

In some applications of this type of control, artificial heat is applied to the sensitive element of the room thermostat at the same time that heat is being added to the space under the control of the thermostat in order to

increase its sensitivity. This usually results in more accurate control and more frequent operation of the heat source.

2. *Modulating or Graduated-acting Control.* This type of control causes motion in the controlled device in proportion to motion caused in the controller by fractional degree variations in the medium to which the controller is responsive. After a fractional change has been measured at the controller and has effected a new position of the valve or damper in proportion to the amount of such change, the system stands by awaiting further change at the controller before any additional motion occurs. The extent of the motion is limited only by the limits of the controller and by the intensity of the change of conditions as measured. With this type of control, the damper or control valve may be operated in intermediate positions between its extreme limits in order to properly modulate or proportion the flow of air, steam or water, reacting with changes of conditions at the controller. Various modifications of this type of control are available, designed to meet special requirements and conditions, all based on operation of the controlled equipment in intermediate positions.

This type of control motion cannot be used on valves of one-pipe steam systems as the partial opening of the valves will not permit the condensate to escape against the flow of incoming steam. Where this type of control is used to control the flow of steam to a heater coil of a fan system which is in the direct path of untempered outdoor air at temperatures below freezing, care should be taken that the control point and operating characteristics of the regulator are such that the valve is fully open at temperatures below freezing, to avoid the possibility of freezing condensate in the bottom of the coil.

Division of Space under Control

Control systems vary considerably with the type and size of the building, occupancy of the building, and with the heating or cooling system, humidity supplying equipment and ventilating means available for control. In the following paragraphs the general requirements of various phases of these different buildings will be discussed.

1. *Individual Room Control.* The most accurate and flexible form of control for any structure is that calling for the regulation of each individual room by control equipment reacting to conditions in that room only. Such control necessitates a thermostat in each room, located to properly measure the conditions of the room, controlling the radiator, unit heater, unit ventilator or other heating source supplying heat to that room only in which the thermostat is located. This arrangement permits the maintenance of any desired conditions in any room, entirely independent of any other room. In the case of large rooms, where one thermostat location will not serve to properly measure the conditions throughout the room, and where two or more sources of heat supply are provided in the room, additional thermostats may be used, each controlling its respective section of the heating source. This form of control, due primarily to the number of control devices required over the entire building, normally is the most expensive type of control system. However, where maximum flexibility and the most accurate control is desired, individual room control can be depended on to furnish the desired results.

2. *Single Thermostat Control.* Probably more widely used than any

other form of control is the type of automatic system regulated entirely from a single room thermostat. The wide use of this particular means of control is primarily due to the fact that it is the form of regulation best adapted to residences and small buildings, which far out-number the larger structures. In larger buildings, this form of control has definite shortcomings. In the small buildings and average size residences it is possible to select a location and install a thermostat of suitable characteristics which, in controlling from the surrounding air temperature, will hold the temperature of the entire building within entirely satisfactory limits. It must be recognized that the thermostat reacts to and controls from the temperatures to which it is subjected and that, therefore, the position selected for the thermostat must be representative of general conditions throughout the structure. It must further be recognized that if certain areas or rooms of a structure are not properly balanced as regards heating or cooling capacity and distribution, the control as dictated by the thermostat will not produce satisfactory results in these unbalanced areas.

3. *Zone Control.* As the size of buildings increases, it becomes increasingly difficult to provide proper regulation for the entire structure from a single thermostat control. In such instances, where the advantages of individual room control are not obtainable by reason of its cost, an intermediate form of control system is available, commonly described as *zone control.* In this form of control system a building is divided into areas or zones such that the general requirements and the general conditions through the areas are relatively constant as to exposure and occupancy, and then each zone is provided with control equipment which functions to regulate the conditions in that particular zone. As in the case of individual room control, each zone may be regulated to its own needs which may vary from the needs of other zones within the same structure.

Variations of the usual zone control methods by the use of recently developed special devices have been quite successful in obtaining greater economy from heating systems. Frequently these use an outside thermostat or group of thermostats which adjust the operation of the controls to conform to variations in weather conditions.

CENTRAL FAN SYSTEMS

A central fan system includes any conditioning system by which either outdoor air, return air, or combinations of outdoor and return air, are conditioned at a central point and then distributed through duct work to the various sections of the space being conditioned.

Heating Cycle

Central fan ventilating systems may be sub-divided, first into split systems, by which air is supplied for ventilating purposes only and heat is supplied in winter from another source such as direct radiation; and second, into combined systems, in which the functions of ventilation and heating are both performed by the central fan system.

A control system for a central fan ventilating system using all outdoor air and discharging air at a predetermined temperature is illustrated in Fig. 2. Thermostat T_1 located in the outdoor air intake is set just above

freezing, and controls valve V_1 on the first heating coil. This valve must be completely open or completely closed to avoid danger of freezing. The by-pass damper around the heaters and the other two valves V_2 and V_3 are controlled by thermostat T_2 located in the discharge duct from the fan. If the temperature of the discharge air increases, through the action of T_2 the damper is moved automatically to admit more cold air. Should this not reduce the temperature sufficiently, the valves V_2 and V_3 on the heating coils will be closed gradually and in sequence until the correct temperature is reached. The control of the damper and valves V_2 and V_3 must be gradual or there will be a wide fluctuation in temperature.

In ventilating systems it is customary to supply air to the ventilated spaces at an inlet temperature approximately equal to the temperature maintained in the rooms. The radiators therefore are designed to take care of all the heat losses from the room and in order to maintain controlled room temperatures it is necessary to control the radiators independently of the ventilation control.

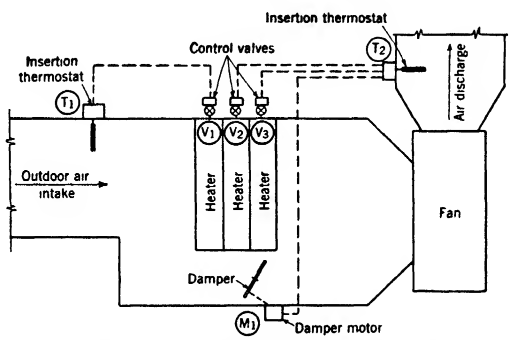


FIG. 2. CONTROL OF A SPLIT SYSTEM OF VENTILATION

In some installations, such as theatres and auditoriums, it is difficult to install sufficient direct heating surface to offset the heat losses from the room. There are also installations where a short heating-up period is allowed before occupancy of the room, and in these cases it is necessary to use the entire heating capacity of the ventilating system for this purpose. An additional thermostat may be installed in the room which will take the control away from the fan discharge thermostat (T_2 in Fig. 2) and utilize the full heating capacity when the room is below normal temperature.

In central fan systems, air washers are often used and in such cases, due to the effect of temperatures on humidity, additional control is required. An arrangement with control of the second tempering heating coil from the air washer temperature and with the usual control of the first tempering heating coil from the outside temperature is shown in Fig. 3. This permits the air to be kept cool while passing through the washer so that too much moisture will not be absorbed. Control of the reheating units and by-pass damper by an insertion thermostat in the fan discharge, and the application of a pilot thermostat to a system of this sort is illustrated in Fig. 3.

Where a number of rooms are to be heated and ventilated through one central fan system it is customary to provide tempering heating units, automatically controlled to provide a minimum temperature for ventilation only and additional heating units to supply the heating requirements. These reheating units may be located in the various branch ducts to the different rooms, each under control of its individual room thermostat, or individual ducts may be run to the various rooms from the central unit. In this case reheater coils are provided to maintain a predetermined temperature in a warm air chamber. Each room duct is connected to this warm air chamber and to the tempered air supply, and through the action of a room thermostat on a gradual-acting double-mixing damper the proper proportions of warm and tempered air are secured to maintain desired conditions in the room.

In all types of central fan systems, the outdoor air damper is usually opened and closed by a damper motor controlled from a manual switch

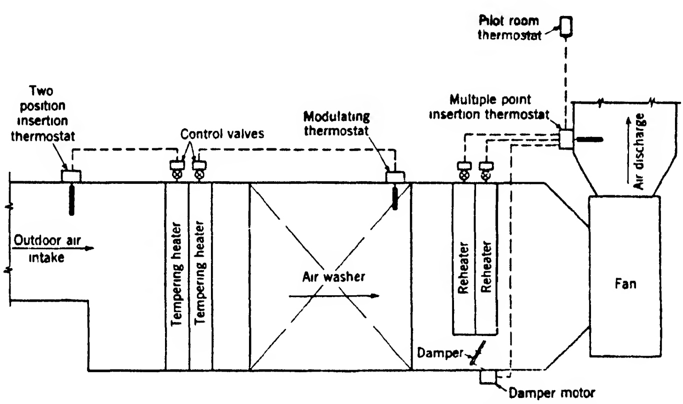


FIG. 3. CONTROL OF VENTILATING SYSTEM WITH AIR WASHER USING PILOT THERMOSTAT

or by a relay in the fan motor circuit, so arranged that when the fan motor is started, the relay causes the damper motor to open the outdoor air damper.

Recirculating and vent dampers may also be opened and closed by means of damper motors controlled from remote locations. Generally these damper motors are positive acting and are either completely open or closed. However, in some cases, where part outdoor air and part recirculated air is desired, it is advantageous to control the dampers so that definite proportions of damper opening area exist. In some installations the control of outdoor air and recirculating dampers is under the command of a thermostat at the intake to the conditioner, in which case the proportions of outdoor and recirculated air are fixed by the resultant temperature of their mixture. This arrangement tends to reduce the amount of outdoor air used as the outside temperature is lowered.

The operation of a central fan system during the heating cycle often results in unfavorably low relative humidity and the provision and control of humidity becomes an important factor of the system. If water spray humidification is used, control may be effected by a humidity con-

troller actuating a control valve in the water supply to the sprays. If steam humidification is used, either of the steam jet type or of the steam heated evaporating type, the flow of steam may be controlled from the humidity controller in the ventilated space. Where an air washer is used, approximate control of humidity may be obtained by maintaining the air temperature in the air washer at a predetermined desired dew-point temperature.

For example, the dew-point temperature at 70 F and 40 per cent relative humidity is 45 F. Therefore, if the air temperature is maintained at 45 F as it leaves an air washer (assuming it is fully saturated) and then is heated to 70 F, it will have a relative humidity of 40 per cent. If it is desired to maintain these conditions in a given space, the air temperature can be raised to any necessary point, say 120 F (at which the relative humidity will be only 9 per cent). When the heat in the air has been dissipated, the space temperature being maintained at 70 F, the relative humidity will be 40 per cent.

Whenever moisture is being added to the air during the heating cycle by the use of a spray or any other means, a considerable amount of care must be used in order to prevent frost from collecting on the windows due to the air being reduced below its dew-point at the inside surface of the windows.

Cooling Cycle

Central fan cooling systems are divided into two general groups based upon the methods employed to control the temperature and humidity of the treated space. Cooling normally involves the removal of moisture from the air, and to accomplish this end the temperature of the air must be lowered below the dew-point. The air at this low temperature must then be treated or introduced into the room in such manner as to avoid uncomfortable cold drafts.

In the first group the air is supplied from the conditioner after being cooled and dehumidified to a fixed temperature and humidity and then before entering the treated space is reheated. This is accomplished either by passing the air through coils heated with steam, hot water, or other heating medium, or the air from the conditioner is mixed with recirculated air before entering the conditioned space.

In the second group are those systems which use the treated space as a mixing chamber, the air being supplied to it at the temperature and humidity leaving the conditioner and depending upon diffusion in the conditioned space to give ultimately the correct conditions. In these systems the temperature and humidity of the treated space are measured and govern, through control of the cooling means, the temperature and the humidity of the air leaving the conditioner.

In Fig. 4 is represented one of the most simple central fan types of cooling system. Thermostat T measures the temperature within the treated space and operates to start and stop the refrigeration compressor or to control the supply of refrigerant to the cooling unit as required to maintain a fixed temperature in the space.

There are three general methods for the control of relative humidity in central fan cooling systems, which are:

1. By provision for limiting the relative humidity in addition to the temperature at a definite point. When this method is used, either temperature or humidity may demand operation of the cooling source regardless of whether or not the other factor has been exceeded. The use of a high limit humidity control in this manner is desirable during conditions of high relative humidity but its operation may cause excessive cooling unless some method of reheating is employed.

2. By the maintenance of a fixed effective temperature. By this method, a definite relation is maintained between temperature and humidity, and sensible cooling is done whenever possible instead of the removal of latent heat in the form of moisture.

3. By the maintenance of a fixed dew-point in the air discharge. This method usually provides for the control of relative humidity within the space being conditioned between reasonable limits, but does not take into consideration any change in the latent heat load, as compared to the sensible heat load.

The necessity for varying inside temperature conditions in accordance with changes in outdoor conditions on many types of installations is important. A control system is shown in Fig. 5 where the temperature of the treated space is adjusted according to the outdoor temperature.

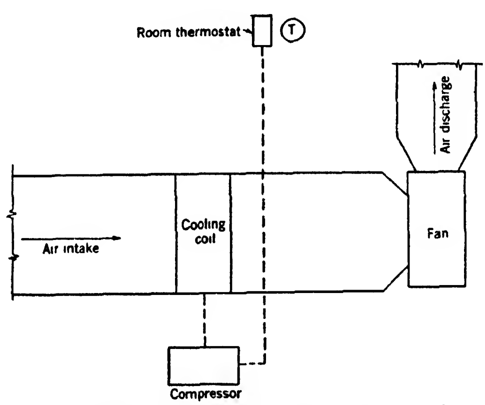


FIG. 4. DIAGRAM OF SIMPLE COOLING SYSTEM CONTROL

Thermostat T_1 measures the outdoor temperature and thereby automatically determines the inside dry-bulb temperature control point. Thermostat T_2 in the conditioned space measures the temperature of that space and controls the refrigerant to the cooling coil so as to maintain the temperature in the space being conditioned at the point which has been set up by thermostat T_1.

It is usually found desirable to adjust the indoor temperature between available limits with the outdoor temperature all of which is fully described in Chapter 3. Various combinations of control may be applied to cooling systems to secure desired relationship between outdoor temperature and resultant indoor temperature and humidity.

All Year Systems

An all year central fan conditioning system consists of the combination of a ventilating system and a cooling system.

During certain seasons of the year, it is sometimes possible to control

the dew-point of the air discharged from an air washer by regulating the relative quantities of outdoor and return air. The use of this method for controlling the outdoor and return air dampers may also provide for automatic change-over from the heating to cooling cycles, providing thereby for the maintenance of a fixed dew-point temperature in the air-discharge during both cycles.

Complete automatic control of all year systems incorporates an automatic change-over between the cooling and heating cycles. If the installation necessitates operation of manual switch or other device to change over between the heating and cooling cycles, then the control system is semi-automatic. The full automatic change-over between cycles becomes particularly desirable in the early and late portions of the cooling and heating seasons when heating is required during the early and late portion of the day and cooling may be required during the middle of the day.

A system for the control of an all year conditioning system providing for automatic change-over from the cooling to heating cycles is illustrated in Fig. 6.

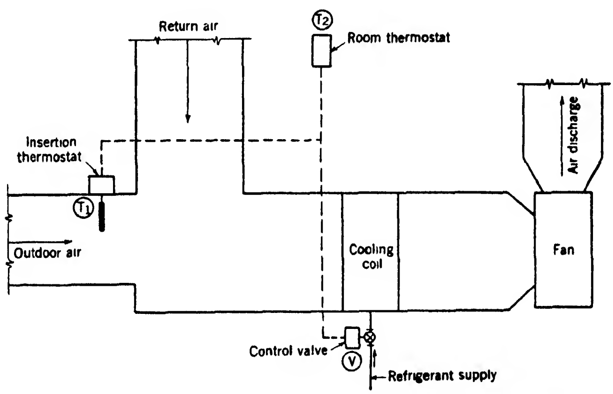


FIG. 5. DIAGRAM OF COMPENSATED COOLING SYSTEM CONTROL

During the heating cycle, thermostat T_1 in the return air or room measures the temperature of the conditioned space and modulates control valve V_1 which, in turn, modulates the flow of steam to the heating coil so as to maintain a fixed temperature in the space. Humidity control H_1 measures the relative humidity in the space being conditioned and opens control valve V_2 so as to admit water to the sprays whenever moisture is required in the air.

During the cooling cycle, thermostat T_2 in the return air measures the temperature in the space being conditioned and modulates control valve V_3 which, in turn, modulates the flow of water to the cooling coil so as to maintain a fixed temperature in the space. Humidity control H_2 measures the relative humidity in the space being conditioned and then assumes command of control valve V_3 whenever the relative humidity exceeds a predetermined amount.

During the heating cycle thermostat T_3 acts as a low limit. It assumes command of control valve V_1 whenever it is necessary to prevent the air

discharge temperature from falling below a minimum point. Thermostat T_3 may also be arranged to act as a low limit during the cooling cycle if the conditions of the installation make it desirable.

Thermostat T_4 installed in the inlet to the conditioner controls damper motor M_1 which in turn regulates the relative quantity of outdoor and return air admitted to the system. This damper action may be provided with a minimum setting of the outdoor air damper so that a minimum fixed requirement of outdoor air will be insured for ventilating purposes.

Humidity control H_3 measures the outdoor air relative humidity and prevents the outdoor air damper from opening beyond its minimum

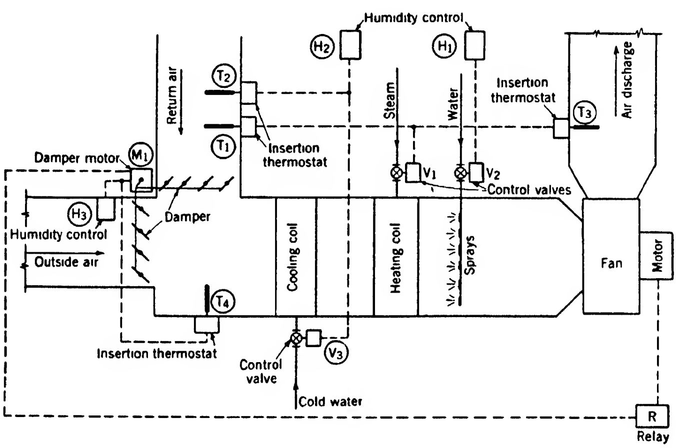


FIG. 6. DIAGRAM OF COMPLETE AUTOMATIC CONTROL ALL YEAR AIR CONDITIONING SYSTEM

position whenever the outdoor air relative humidity exceeds a predetermined point.

When the fan is stopped, relay R positions damper motor M_1 so as to close the outdoor air damper.

Thermostat T_1 must be set at a lower temperature than thermostat T_2 in order that each may assume command upon the fall or rise respectively of the temperature of the return air. As an example, T_1 might be set at 72 F and T_2 at 76 F. When the temperature of the return air approaches 72 F, it would indicate that a change had taken place from the cooling to the heating cycle and when the return air approaches 76 F, it would indicate that a change has taken place from the heating to the cooling cycle.

UNIT SYSTEMS

A unit system provides for the same functions as a central fan system except that the actual conditioning is usually done within the space being conditioned instead of at some central location outside of the space. The

automatic control problems, therefore, become exactly the same as for central fan conditioning systems except that compactness, ease of installation and control cost often assume somewhat more importance.

Because of the usual segregated location of unit equipment throughout a building and its consequent lack of competent supervision, complete automatic control is essential to its satisfactory operation.

Unit Heaters

In its simplest form, unit heater control consists of a room thermostat the function of which is to start the unit heater motor when heat is required and shut it off when the demand is satisfied. With this limited control, it is possible in some instances that, with no steam available at the heater, the operation of the fan at the command of the thermostat would cause objectionable drafts. To prevent this occurrence, limit controls are available which will prevent the operation of the fan at the command of the room thermostat except when steam is available, as determined by the temperature of the steam or return pipe or the pressure of the steam supply.

In some cases it is desirable to operate the unit heaters continuously for circulation of air where, due to the type of installation, cold drafts will not result therefrom. In such instances the room thermostat regulates the supply of steam to the unit through a control valve in the steam supply line and the unit heater motor operation is manually controlled.

Where several unit heaters serve a limited area, they may be grouped for purposes of automatic control, and several heaters placed in operation at the command of one thermostat. By properly grouping the units which will operate together, the benefit of zone control can often be obtained with a minimum of control equipment. Where such group operation is utilized, the thermostat and limit control usually function through a relay, as the combined load of the several motors may exceed the current capacity of the thermostatic control device.

Cooling Units

The recommended form of temperature control for the cooling unit contemplates the continuous operation of the cooling unit fan with automatic two position regulation of the compressor or cooling coil as determined by a room thermostat or by a temperature controller measuring the temperature of the return air as it is taken into the cooling unit. Such operation insures continuous circulation of the air in the room served by the cooling unit, and in addition to providing the cooling effect due to the moving air, this circulation overcomes the tendency of air to stratify. Thus, as this temperature tends to rise, the temperature controller will open the valve supplying either refrigerant or cold water to the cooling unit coil or start the compressor.

Cooling units may also be controlled by arranging the room thermostat to start and stop the fan motor or by a combination of motor and refrigerant control.

A humidity controller may be used in conjunction with the thermostat as a high limit control to permit the cooling and dehumidifying of the air

whenever the relative humidity rises above some predetermined point such as 60 per cent even though the thermostat is satisfied. This control is desirable on damp days or in conditions where the humidity load may become excessive, but its operation will result in excessive cooling unless some means of reheating is provided.

Unit Ventilators

There are various types of unit ventilators available but in general all types are designed to draw air from the outside or to mix outside and recirculated air, heat it and introduce it into the room under control of a thermostat.

In the application of control to unit ventilators the essential requirement is that the action be graduated to prevent sudden changes in the temperature of the discharged air and where direct radiation is used in conjunction with the unit that the cycle of control be so arranged that steam will be admitted to the direct radiation only when the unit is unable to carry the heating load. This arrangement prevents the unit from delivering air at low temperatures to offset the overheating effect of the direct radiation and results in the delivery of a higher percentage of tempered air.

There are two general types of control applied to unit ventilators as follows:

1. The mixing or by-pass damper type of unit is provided with a damper, equipped with a damper motor, which, under control of the thermostat, passes air through and around the heating element in such proportion as to maintain a uniform room temperature, the two streams of cold and tempered air being mixed and diffused at the ceiling A control valve may also be used on the steam supply to the heating element of the unit and should be arranged to throttle the steam supply when the damper approaches a position to by-pass all of the air.

The outside air damper of this type of unit is usually provided with a damper motor and controlled by a remote manual switch to assume either a fully open or fully closed position.

2. The recirculating type of unit ventilator is equipped with a control valve on the steam supply to the heating element of the unit and with a damper motor on the outside air-recirculating air damper, both under the control of the room thermostat. Some units are so arranged that a mixture of outside air and recirculated air passes through the heating element and others so that only the recirculated air is heated

The fundamental requirements of control as applied to this type of unit is that the steam supply to the direct radiation, the steam supply to the unit ventilator and the mixing of outside and recirculated air be accomplished in a definite cycle or sequence to meet the requirements of the particular unit used and differs from the mixing damper type of unit in that the percentage of outside air and recirculated air delivered by the unit is determined by room temperature. The damper motor is sometimes arranged so that a fixed minimum quantity of outside air is delivered continuously as soon as the room has reached a predetermined temperature. A limit thermostat, either in the mixing chamber or in the discharge of the unit, is sometimes used in conjunction with the room thermostat, so arranged that the action on either the control valve or the dampers, or both, is stopped when a predetermined minimum temperature has been reached in the unit discharge, to prevent delivery of air at a lower temperature.

For additional information on the control of unit ventilators when installed and operated under various types of applications refer to Chapter 22.

All Year Conditioning Units

It is desirable to provide for automatic change-over between the cooling and heating cycles in the control system for all year conditioning units because of the probable necessity of changing over a large number of units if done manually.

A control system for an all year conditioning unit providing for the automatic change-over is shown in Fig. 7. Operation of the control equipment is as follows:

1. *During the Heating Cycle.* Combination controller T_1 measures the temperature in the space being conditioned and opens control valve V_2 so as to admit steam to the heating coil whenever heat is required so as to maintain a fixed temperature in the space. Combination controller T_1 also measures the relative humidity in the conditioned space and opens

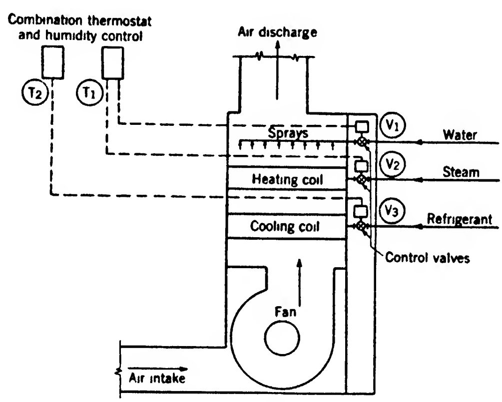


FIG. 7. ALL YEAR AIR CONDITIONING UNIT WITH COMPLETE AUTOMATIC CONTROL

control valve V_1 so as to admit water to the sprays whenever moisture is required in the space.

2. *During the Cooling Cycle.* Combination controller T_2 measures the temperature and humidity in the conditioned space and opens refrigerant control valve V_3, thereby admitting refrigerant to the cooling coil whenever cooling is required to maintain the temperature or relative humidity within predetermined maximum limits.

The temperature control point of controller T_1 must be set at a lower point than that of controller T_2 in order to provide for the automatic change-over between the cooling and heating cycles. As an example, controller T_1 might be set at 72 F and controller T_2 at 76 F. As the temperature in the space approaches 72 F, it would indicate a change from the cooling to the heating cycle and when the temperature in the space approaches 76 F, it would indicate a change from the heating to the cooling cycle, and the corresponding controllers would assume command. In the same way, the relative humidity control point of controller T_1 would be set at a lower point than that of controller T_2. As an example, T_1 might be set at 35 per cent and T_2 at 60 per cent.

CONTROL OF AUTOMATIC FUEL APPLIANCES

It is essential that automatic controls be used with oil burners, gas burners, and stokers in order to maintain even temperatures and provide safe and economical operation of the heating plant. There are many types of burners and many types of automatic control, and it is essential that the proper type of control equipment be selected to fulfill the requirements of the burner equipment and its application.

Combustion regulation equipment should be used on the larger commercial and industrial applications to control the secondary air supply and thereby provide for economical operation. This type of control will usually consist of a pressure regulator which measures and controls the pressure over the fire and which thereby indirectly regulates the carbon dioxide percentage in the flue gas.

On all automatically-fired steam boilers it is advisable to provide control equipment which will stop the burner operation in case the boiler water line falls below a predetermined level of safety.

Thermostats used to control automatic fuel appliances may be provided with clock mechanisms which will operate to maintain lower temperatures during night hours for economy of fuel.

Oil Burner Controls

In the normal oil burner installation as encountered in residential and small commercial installations, the burner operation is frequently regulated by electric controls and primarily governed by a room thermostat. It is essential that a limiting control be incorporated in the control system to prevent the temperature of the heating medium from exceeding any predetermined safe maximum. The type of limit control selected will depend on the type of the heating system. In a warm air furnace installation, a limit control would be used, reacting to the temperature of the heated air in the bonnet of the furnace; in a hot water system a control reacting to the temperature of the water in the boiler; and in a steam system a control reacting to the pressure of the steam in the boiler.

In addition to the normal control of the burner from the room thermostat and limit control, it is necessary that a combustion safety device be used to prevent operation of the burner under hazardous conditions. The oil fire is automatically ignited by means of gas, electric spark or incandescent element and the combustion safety control acting through a sequence device permits the burner operation only when the fire is properly established as the burner starts up. A further function of the combustion safety control is to react to any major disturbance in the flame during the running operation, shutting down the burner and preventing the discharge of unburned fuel if for any reason the flame is extinguished.

Gas Burner Controls

In the case of the domestic burner, full automatic operation is the normal requirement and the burner is started and stopped at the command of a room thermostat which, in turn, opens and closes a control valve in the gas supply line. Modulating controls and controls providing a high and low fire are also available for gas burners. For purposes of preventing abnormally high temperatures in the bonnet of gas-fired

furnaces or in the temperature of the water in gas-fired hot water heating boilers or excessive pressures in gas-fired steam boilers, temperature and pressure limit controls are used. Ignition is normally secured through the use of a gas pilot flame and a safety device is provided, utilizing the heat of the pilot flame in such a manner that if the pilot light is extinguished for any reason, the main gas valve cannot be opened. For satisfactory and economical operation, all automatically-fired gas burners should be equipped with pressure regulators on the gas supply line.

Stoker Controls

Domestic stokers are normally placed under command of a room thermostat for primary operation subject also to the command of a limit control to prevent their operation when conditions in the boiler or furnace exceed predetermined safe maximums. Utilizing coal as fuel, automatic ignition is not provided and the stokers, once ignited, maintain their fire, merely changing the rate of combustion by changing the draft and the rate at which the coal is fed. Thus, at the command of the room thermostat the stoker motor is started, driving a forced draft fan and fuel feeding mechanism. The rate of combustion is thus increased and this operation continues until the thermostat has been satisfied when the motor is stopped and the fuel in the combustion chamber continues to burn at a slow rate with reduced draft.

At certain seasons of the year, the operation of the stoker under the requirements of the thermostat may be so infrequent that there is a possibility of the fuel in the combustion chamber burning out or the fire going out between operations. To prevent this occurrence, automatic controls may be utilized to operate the stoker independently of thermostat requirements, sufficiently to sustain the fire either through a timing device functioning for short periods at predetermined intervals or through a temperature control device reacting to minimum stack or boiler temperatures. Control may also be utilized to prevent stoker operation and the delivery of coal into the combustion chamber in the event that the fire has gone completely out. This control is governed normally by the stack temperature and shuts down the stoker after a predetermined minimum stack temperature is reached.

RESIDENTIAL CONTROL SYSTEMS

The control installation in a residence may vary from the simple regulation of a coal-fired heating plant to the completely automatic all year air conditioning system. Residential installations with automatic fuel burning appliances, such as oil burners, gas burners or stokers, are normally equipped with single room thermostat, limit and safety controls as outlined above under Control of Automatic Fuel Appliances.

Coal-Fired Heating Plant

Control in the normal coal-fired domestic heating plant consists of regulating the combustion rate in accordance with requirements. This function is accomplished by a spring or electric-driven damper motor which under the command of a room thermostat and through chain linkage, operates the draft and check dampers of a boiler or warm air

furnace. Such installation should be protected against excessive temperature or pressure by means of a limit control serving to check the fire when temperature or pressure conditions at the boiler or furnace reach a predetermined maximum.

All Year Domestic Hot Water Supply

Hot water or steam heating boilers with automatic fuel burning appliances can be used for all year heating of domestic water supply. The fuel burning appliance in this case is controlled from the temperature of water or pressure of steam in the boiler to maintain uniform boiler conditions and domestic hot water is heated by means of an indirect heater. The heating of the residence is normally governed by means of a thermostat which operates a control valve in the flow line of a gravity hot water or a steam system, or controls the operation of a circulating pump in a forced circulation hot water system.

Air Conditioning Systems

Residential air conditioning systems are of various types normally including a heating source and a motor-driven fan for circulating air. In addition, such installations may involve spray-head equipment, the purpose of which may be only to supply humidity, or which, in some instances, are of greater capacity and serve not only to humidify but to wash the air passing through them. It is also common practice to include dry filters to aid in air cleaning. Such installations distribute suitably heated and humidified air during the heating cycle, and during the summer or cooling cycle may be used effectively as conditioners if the washer unit is supplied with water at suitable temperature or if such an installation is equipped with other refrigeration means.

During the heating cycle the regulation of temperatures is normally one or the other of the problems previously discussed in connection with the various types of heating sources described, such as the oil burner, gas burner, stoker or the coal-fired heating plant under automatic control. Regulation of the humidity during the heating cycle is normally accomplished by opening and closing a solenoid water valve supplying water to the spray-heads, the solenoid valve being under control of a room type humidity control. In the average installation the fan is permitted to run only during such intervals as the thermostat is calling for heat or at the command of a limit control to prevent the overheating of the bonnet of a warm air furnace. The limit control should also prevent the operation of the fan at the command of the thermostat until the circulating air temperature has increased to a predetermined point.

When cooling equipment is provided in such installations, control during the cooling cycle will be an adaptation of the control principles described for central fan systems selected for the type of cooling equipment utilized.

The selection of automatic control equipment for residential air conditioning systems is just as important as for commercial installations. Fewer controls are generally used and systems are usually less complicated except in the case of a very large residence installation when the control system may become as complete as the commercial installation.

CONTROL OF REFRIGERATION EQUIPMENT

The most common means of providing cooling for air conditioning may be divided into four general classifications as follows:

Compressor Type Refrigeration

Refrigeration compressors may furnish refrigerant to direct expansion cooling coils through which air is being passed, or to coils in cooling tanks through which water is passed which is then pumped to air washers or cooling coils through which the air is passed.

In either case the compressor motor may be started and stopped in order to meet the demand for refrigeration or a pressure controller may be used to regulate the low side or suction pressure of the compressor. When the latter method is used, the flow of refrigerant to cooling coils may be regulated by the opening and closing of a solenoid refrigerant valve at the command of a temperature controller or thermostat.

A high pressure cutout as an individual unit or in combination with either a temperature or pressure controller provides a safety feature against the development of excessive pressures on the high side of the compressor.

Refrigeration by Ice

When ice is used for the cooling or dehumidification of air, it is usually placed in bunkers and water is sprayed over it. This water, after being cooled, may be used in air washers or surface cooling coils and is usually returned to the bunker for additional cooling after being used.

Control of the water temperature leaving the cold water tank may be maintained by a temperature controller, which measures the temperature of the water in the tank and modulates a control valve in a by-pass which permits a portion of the return water to return directly to the tank instead of passing through the sprays.

Vacuum Refrigeration

A vacuum refrigerating system consists of an evaporator, compressor, condenser and auxiliaries. The refrigerant used is water, and water vapor (steam) is the power medium.

Water which has been passed through an air washer or cooling coil is sprayed directly into the evaporator or water cooler where it is cooled by its own evaporation. A condenser is attached directly to the compressor discharge and its function is to recondense the water vapor drawn from the evaporator, plus the steam which supplies the energy for compression.

The temperature of the cold water leaving the flash chamber should be measured by a temperature controller which will in turn operate a two-position or positive-control valve installed in the steam line to the jet so as to permit steam to flow only when cooling is required. If city water is used in the condenser, the amount of water should be modulated according to the demand as measured at the condenser outlet by means of a temperature controller and control valve.

Refrigeration by Well Water

When well water is available in sufficient quantities at low temperatures during the cooling season, it may be pumped directly to air washers or cooling coils. Control is usually effected through control valves on the water supply to the cooling unit actuated by temperature or humidity controllers, or both, located either at the outlet of the conditioner or in the conditioned space.

INDUSTRIAL PROCESSES

There are many industrial processes requiring automatic temperature and humidity regulation. The control equipment operates on the same principles that have been described, but it is often especially designed for each particular process. Each installation, or the installation for each process, is likely to be a problem peculiar to that process.

PROBLEMS IN PRACTICE

1 ● What important functions of heating, ventilating, and air conditioning systems do automatic controls fulfill?

Controls are applied to maintain adequate requirements for human comfort and efficiency, to maintain requirements for industrial processes, to obtain economy in operation, and to provide necessary safety measures.

2 ● How may temperature control be obtained in a room heated by a unit heater?

With constant steam supply, the unit heater motor may be started or stopped by a thermostat, either directly or through a relay. With intermittent steam supply, operation of the motor by thermostat can be limited to the time that steam is available, by using a reverse-acting temperature or pressure limit switch.

3 ● How may temperature control be obtained in a room cooled by a self-contained mechanical unit?

The fan operation may be controlled by a manual switch, while a room thermostat in conjunction with a solenoid valve may regulate the flow of the refrigerant to the coil. The thermostatic circuit might be operative only when the fans are running, and the compressor might be controlled by refrigerant pressure.

4 ● How may temperature control be obtained in a room heated by an automatically-fired warm air furnace?

A room thermostat might control the combustion unit, and a limit switch in the top of the furnace unit, when at a low setting of its control might operate the fan whenever there is a rise of temperature, and when at a high setting of its control it might shut off the combustion unit. A room humidity control operating a solenoid valve on the water supply to the humidifier, or operating a relay on the recirculating pump motor to the humidifier, may be connected in parallel with the fan motor Humidification may be supplied only when heat is supplied and when the humidity control acts in conjunction with a time switch.

5 ● How may humidity be controlled in a unit humidifier for a steam or hot water heating plant?

Since heat is required for evaporation, a temperature limit switch, preferably of the immersion type, may be placed in the heating supply riser to cause the unit to be inoperative when heat is not available. A room humidity control will operate a solenoid valve on the water supply to the sprays Both the solenoid valve and the humidity control may be electrically wired in parallel with a fan motor, and be subject to the temperature limit switch.

Chapter 38

MOTORS AND CONTROLS

Direct Current Motors, Alternating Current Motors for Single Phase and Polyphase, Special Applications, Classification of Motors, Manual Control, Automatic Control, Pilot Controls, Direct Current Motor Control, Squirrel Cage Motor Control, Multispeed Motor Control, Slip Ring Motor Control, Single Phase Motor Control

THE electric motor, available in many different types suitable for various services, is now the most widely used form of prime mover. The equipment for starting, controlling and protecting these motors varies with the type and with the functions it is desired to attain. Motors used for heating, ventilating and air conditioning applications may be divided into two general classifications as follows:

1. For use with direct current.
2. For use with alternating current.

DIRECT CURRENT MOTORS

There are three types of direct current motors available:

1. Shunt Wound.
2. Compound Wound.
3. Series Wound.

Shunt Wound motors being suitable for application to fans, centrifugal pumps, or similar equipment where the amount of starting torque required is relatively small, are used for the majority of applications in the field of heating, ventilating and air conditioning. They may be used on reciprocating pumps and compressors, if started under unloaded conditions.

Compound Wound motors are required for application to compressors, stokers, reciprocating pumps when started under loaded conditions, and also when applied to similar equipment where high starting torque is required. Whenever frequent starting makes high starting and accelerating torque desirable, or where sudden changes of load are encountered, compound wound motors are used.

Series Wound motors find only limited application in a few special cases and are available in only a limited range of sizes.

Speed Characteristics

Direct current motors are available with speed characteristics of four types:

1. Constant speed.
2. Adjustable speed.
3. Adjustable varying speed.
4. Varying speed.

Constant Speed motors may be shunt wound or compound wound. Shunt wound motors have a nearly flat speed-load characteristic, with a regulation of 15 per cent for up to ¾ hp, 12 per cent for one to 5 hp and 10 per cent for 7½ hp and larger, based on full load speed.

Compound wound motors have a speed regulation over the range from full load to no load of not more than 25 per cent, based on full load speed.

Adjustable Speed motors are usually shunt wound since it is impractical to maintain the proper relation between the shunt and series fields of compound wound motors when wide variations of the field strength are required to obtain the speed adjustment.

Adjustment of the speed of shunt wound motors is obtained by field control on motors rated at ¾ hp and larger, with the minimum or base speed at full field strength and higher speeds at reduced field strength (obtained by adding resistance in the field circuit). The speed regulation from no load to full load will not exceed 22 per cent for 2 to 5 hp; nor 15 per cent for 7½ hp and larger. Below 2 hp, the regulation may exceed 22 per cent. If closer speed regulation is required, specifically wound motors must be obtained.

Practically constant horsepower output is obtained at all speeds up to a ratio of 2 to 1. For higher speed ratios, the horsepower rating at the minimum speed is less than at the maximum speed, this difference varying with the speed ratio. High efficiency is maintained over the entire speed range. Most listed constant speed motors are suitable for operation up to a speed ratio of 2 to 1 by the use of proper control equipment.

Adjustable Varying Speed motors may be either shunt or compound wound and speed adjustment is obtained by adding resistance in series with the armature. The speed thus obtained is always below the rated full-field speed. Any standard shunt or compound wound constant speed motor may be used in conjunction with the proper armature resistor. The usual range of speed reduction is 50 per cent. The speed obtained for any setting of the resistor depends on the load of the motor and will vary with this load.

The speed regulation at high speed is comparable to a constant speed motor, but becomes poorer as the speed is decreased.

When operating at reduced speed, an increased torque requirement which the motor could easily handle at rated speed is easily sufficient to stall the motor; for example, a motor operating at two-thirds speed would be stalled by a torque about 50 per cent in excess of the normal requirement.

The efficiency of the motor is reduced as the speed is reduced, since the

loss in the resistor is greater at lower speeds. Speed reduction by armature control is usually selected where:

1. A wide speed range is not required.
2. Close speed regulation is not necessary.
3. Operating time at reduced speed is short.
4. Operating load at reduced speed is small so that the reduced efficiency can be ignored.
5. The rating is less than 1 hp.

Varying Speed motors are series wound and the speed varies with the load on the motor. They should be used where:

1. The load is practically constant or increases with speed.
2. The motor can easily be controlled by hand.

They should not be used where there is a possibility of operation without load or at a reduced load, as the speed of the motor may become dangerously high.

For shunt wound motors with full field strength, the starting torque varies almost directly with the starting current, which is dependent on the resistance in the armature circuit. With varying positions of the starting rheostat, it is possible to obtain a wide range of starting torque, within the limits of starting current permitted by the power company.

A compound wound motor requires somewhat less current for the same starting torque. The maximum torque of shunt, series, and compound wound motors is limited by commutation.

ALTERNATING CURRENT MOTORS

Alternating current motors may be divided into two main groups, namely, (1) those operating on single phase current, and (2) those operating on polyphase current.

1. Single phase motors are available in four common types:
   a. Capacitor motors.
      1. Full capacitor.
      2. Capacitor start-induction run.
   b. Repulsion induction motors.
   c. Repulsion start, induction run motors.
   d. Split phase motors.
2. Polyphase (2 or 3 phase) motors are available in four common types:
   a. Squirrel cage induction motor.
   b. Automatic start induction motor.
   c. Slip ring, wound rotor induction motor.
   d. Synchronous motor.

Where the public utility supplying the current determines that a particular installation should be served with polyphase current, it is generally understood that the major portion of the motors will be for polyphase current, although it is commonly acceptable for the smaller motors to be single phase. This will limit the use of single phase current to the smaller motor ratings and the polyphase to the larger motors. Domestic and semi-commercial installations will invariably be single phase.

Single Phase Motors

Capacitor type motors are available in ratings up to 10 or 15 hp for general purposes. These motors are recommended for pumps, compressors and fan duty including housed centrifugal fans and propeller fans. The general purpose motor is commonly known as a high torque capacitor motor having approximately 300 per cent starting torque with normal current and having a different value of capacitance for starting and running which is automatically changed over by a mechanical or electrical means.

Capacitor motors for *fan duty* are usually divided into the open high torque type for belted fans and the totally inclosed non-ventilated low torque type for propeller fans mounted directly on the motor shaft. The open low torque capacitor motor may be used with small centrifugal fans mounted on the motor shaft.

Although the motors for *belted fans* are called high torque, the available starting torque is somewhat less than the torque of the general purpose motor and the slip at full load is approximately 8 per cent. With this larger amount of slip, adjustable speed down to 60 or 70 per cent of rated speed may be obtained by line voltage variation. Motors for *propeller fan* drive may be supplied with sleeve bearings to obtain greater quietness in the smaller sizes where the fan thrust does not exceed approximately 25 lb. For larger fans, thrust ball bearing motors should be used. Low torque capacitor motors have approximately 50 per cent starting torque and do not change the value of capacitance from start to run.

Capacitor motors with *high slip* may have taps brought out from the main winding which when connected to the line, give a second speed of from 65 to 70 per cent of the normal speed. This type of motor must be specially designed for the individual fan, otherwise the correct low speed will not be obtained. Care should be exercised in applying it to centrifugal fans where restriction to the air flow through the use of adjustable dampers changes the motor load and consequently the speed. This same effect is also found in transformer speed controllers, however, a series of transformer taps allow for a selection which partially overcomes the effect of change in motor load.

Capacitor start-induction run motors are usually confined to the smaller horsepower ratings and differ from the capacitor motors by having no running capacitor. The value of starting capacitance used may vary with the different types of applications involved. These motors may be used for practically any of the applications met in air conditioning. However, consideration should be given to the fact that they are not as quiet as a capacitor motor.

Repulsion induction motors start as repulsion motors and operate under full speed as combined repulsion and induction motors through the inherent characteristics of the motor which has, in addition to the wire winding with commutator, a buried squirrel cage winding. No additional switching devices are required to change over from start to run. This and the repulsion motor described below may be used for constant speed drives where high starting torque is required and where commutator and brush noise is not a factor.

The *repulsion start-induction run* motor starts as a repulsion motor, has a switching means for transferring from start to run which short circuits the commutator and permits operation under full speed as a wound induction motor. This motor is suitable for applications similar to those for which the repulsion induction motor is used.

The *split phase* motor has a high resistance auxiliary winding in the circuit during starting which is disconnected through the action of a centrifugal switch as the motor comes up to speed. Under running conditions, it operates as a single phase induction motor with one winding in the circuit. These units are available for the lower horsepower ratings and when equipped with a high slip rotor may be used for adjustable varying speed through line voltage control.

Polyphase Motors

Squirrel cage induction motors are available in three types and a full range of sizes:

1. The normal torque, normal starting current squirrel cage motor has close speed regulation, high efficiency, high power factor, medium starting torque, high pull-out torque, and is suitable for general purpose applications. This motor has a large current inrush and a low starting current power factor It operates with these characteristics only when started directly across the line on full voltage. When central stations require current limiting starting equipment on such motors, the starting torque is less. Current limiting hand operated starters are standard equipment.

2. The normal torque, low starting current squirrel cage motor has approximately the same torque as the normal current motor, but the starting current is about 20 per cent less than the normal torque motor on full voltage and ordinarily within the *National Electric Light Association* locked rotor current limits on sizes up to 30 hp.

   This motor lends itself to automatic or remote control because no current limiting starting equipment is necessary up to and including 30 hp. A magnetic starter with low voltage and thermal relay overload protection gives the most satisfactory service.

3. The high torque, low current squirrel cage motor has a starting torque approximately 25 to 50 per cent greater than the normal torque motor on full voltage with starting current approximately 10 per cent less than the normal torque motor started on full voltage, but within the required limits on 30 hp sizes and smaller These motors are also started directly across the line on full voltage through a magnetic starter or other approved starting device.

These three types of motors are also available in two, three, or four speed designs with variable torque, or constant torque characteristics. Two speed motors may be either single, or two winding; three speed motors are single, two, or three winding; and four speed motors are two, three, or four winding. When a motor is wound with a winding for each speed, better operating characteristics may be obtained because no sacrifice is made for the other speed and operating characteristics approaching single winding motors may be expected.

Frequently, multispeed motors lend flexibility to an installation that cannot be obtained in any other way.

Multispeed motors are started directly across the line through magnetic starting equipment with overload and low voltage protection and compelling relays to insure starting on low speed regardless of the ultimate running speed. Starting on low speed limits the starting current to the starting current of the low speed winding and consequently lowers the maximum demand.

TABLE 1. CLASSIFICATION OF MOTORS

CURRENT	TYPE	SPEED CHARACTERISTICS	FULL VOLTAGE STARTING TORQUE	FULL VOLTAGE STARTING CURRENT	HP RANGE	TYPE OF APPLICATION SEE FOOTNOTE*
		Constant Speed Drives				
DIRECT	1. Shunt	Constant	Medium	Medium	All	(*a*) Fans and (*c*) Centrifugal Pumps
	2. Compound	Constant or Variable	High	Medium	All	(*b*) (*c*) (*e*) Reciprocating Pumps and frequent or hand starting
	3. Series	Variable	High	Medium	Small	(*d*) Fans direct connected
POLY-PHASE	4. Squirrel Cage General Purpose	Constant	Normal	High 6-8 Times	All	(*a*) Fans and (*c*) Centrifugal Pumps
	5. Squirrel Cage Medium Torque	Constant	Normal	Medium 5-6 Times	Medium Small	(*a*) Fans and Centrifugal Pumps
	6. Squirrel Cage High Torque	Constant	High	Medium 5-6 Times	Medium Small	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
	7. Automatic Start High Torque	Constant	High	Low 3 Times	Medium	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
	8. Slip Ring Wound Rotor	Constant	High	Low 1-3 Times with secondary control	All	(*a*) and Hoists (*b*) Reciprocating Pumps (*c*) and Frequent (*e*) or Hand Start
	9. Synchronous High Speed	Constant	Medium	Medium 5-7 Times	Medium Large	(*a*) Fans and Centrifugal Pumps
	10. Synchronous Low Speed	Constant	Low	Low 3-4 Times	Medium Large	(*a*) Reciprocating Compressors Starting Unloaded
SINGLE PHASE	11. Capacitor	Constant	High	Normal	Medium Small	(*b*) Pumps and Compressors

*Applications·

a. Drives having medium or low starting torque and inertia (WR^2) such as fans and centrifugal pumps or reciprocating pumps and compressors started unloaded

b Drives having high starting torques, such as reciprocating pumps and compressors started loaded

c. Similar to (*a*) except where frequent or hand starting (large WR^2) requires a higher starting and accelerating torque

d. Fans direct connected.

e. Stoker drives

TABLE 1. CLASSIFICATION OF MOTORS—(Continued)

CURRENT	TYPE	SPEED CHARACTERISTICS	FULL VOLTAGE		HP RANGE	TYPE OF APPLICATION SEE FOOTNOTE*
			STARTING TORQUE	STARTING CURRENT		
SINGLE PHASE	12. Capacitor Fan	Constant	High	Medium	Medium Small	(*a*) Fans—belted
	13. Capacitor Fan	Constant	Low	Medium	Medium Small	(*d*) Fans—direct
	14. Capacitor Start Induction Run	Constant	Any	Medium	Medium Small	(*a*) Fans (*b*) Pumps and Compressors
	15. Repulsion Induction	Constant	High	Medium	Medium Small	(*a*) Fans (*b*) Pumps and Compressors
	16. Repulsion Start Induction Run	Constant	High	Medium	Medium Small	(*a*) Fans (*b*) Pumps and Compressors
	17 Split Phase	Constant and Adjust-table	Medium	Medium	Frac-tional	(*a*) Fans (*b*) Pumps and Compressors
		Adjustable Speed Drives				
DIRECT	18. Shunt Field Adjustment	Constant	Medium	Medium	All	(*a*) Fans and (*e*) Centrifugal Pumps
	19. Shunt Armature Resistor	Variable	Medium	Medium	All	(*a*) Fans and (*e*) Centrifugal Pumps
POLY-PHASE	20. Squirrel Cage High Slip, Tapped Winding	Variable	Medium	Medium	Medium Small	(*a*) Fans
	21. Squirrel Cage High Slip, Trans-former Adjust-ment	Variable	Medium	Medium	Medium Small	(*a*) Fans
	22. Squirrel Cage Separate Wind-ing or Regrouped Poles	Constant Multi-Speed	Medium or High	Low	All	(*a*) Fans (*b*) Pumps and (*c*) Compressors

TABLE 1. CLASSIFICATION OF MOTORS—(Concluded)

CURRENT	TYPE	SPEED CHARACTERISTICS	FULL VOLTAGE		HP RANGE	TYPE OF APPLICATION SEE FOOTNOTE*
			STARTING TORQUE	STARTING CURRENT		
POLY-PHASE	23 Wound Rotor, Slip, Ring, External Secondary Resistance	Variable	High	Low	All	(*a*) Fans and (*b*) Centrifugal Pumps
SINGLE PHASE	24. Capacitor High Torque Tapped Winding	Variable	High	Normal	Medium Low	(*a*) Fans, belt
	25. Capacitor Low Torque Tapped Winding	Variable	Low	Medium	Medium Low	(*d*) Fans, direct
	26. Capacitor High Torque Transformer Adjustment	Variable	Low	Low	Fractional	(*d*) Fans
	27 Capacitor Low Torque Transformer Adjustment	Variable	Low	Low	Fractional	(*d*) Fans
	28 Split Phase Regrouped Poles	Constant	Normal	Normal	Fractional	(*d*) Fans

Often where the central station requires current limiting starting equipment for the normal torque, normal starting current motor, it is advisable to use the normal torque low starting current multispeed motor.

High slip polyphase motors may be used for adjustable varying speed drives in a manner similar to that described for capacitor motors, with either a transformer speed regulator or tapped motor windings.

It is apparent from these motor characteristics that a squirrel cage motor may be selected for operating any air conditioning and allied equipment.

Automatic start induction motors are constructed with two windings on the rotor, one of which is a high resistance, squirrel cage winding used in starting and gives a high starting torque approximately the same as the high torque, squirrel cage. A centrifugal mechanism within the motor switches to the second low resistance winding when the motor comes up to speed, thus obtaining running characteristics equal to the normal torque, normal current squirrel cage motor. The power factor of the starting current is high.

Slip ring wound rotor motors are built for two classes of service, constant speed and adjustable variable speed. The motors are identical in each case and use the same primary control, the only difference being in the secondary control.

Slip ring motors for constant speed service are used where high starting torque with low starting current is required for bringing heavy loads up to speed. The resistance is in the secondary or rotor circuit, only when starting, and is short circuited when the motor is up to speed.

For adjustable varying speed service, part or all of the secondary controller resistance is in the circuit whenever the motor is operating below full speed. The speed obtained with a given resistance in the secondary circuit is dependent on, and changes with the load on the motor. The horsepower developed by the motor is approximately proportional to the speed, whereas the power required by the motor is practically the same at reduced speed as at full speed, hence the efficiency at reduced speeds is much lower than at full speed.

Synchronous motors are ordinarily used only where there is a need for, or advantage in, obtaining power factor correction. It is necessary to consider each application as a special case which must be individually engineered, since for satisfactory operation, the combined moment of inertia of the compressor fly wheel and motor rotor must be correctly established.

The general classification of motors used for heating, ventilation and air conditioning is shown in Table 1.

SPECIAL APPLICATIONS

A few applications of motors may require special constructions such as splash proof, explosion proof, fully enclosed, and self-ventilated to meet hazardous or special duty conditions. These requirements are frequently encountered in certain industrial applications, in which cases it is necessary to select the motors from the viewpoint of service conditions, as well as the required operating characteristics to meet the demands of the machines being driven.

CONTROL EQUIPMENT FOR MOTORS

In selecting control for alternating and direct current motors it is necessary to determine whether the installation is to be operated by manual or automatic control. The available controls and the function of each group of apparatus may be outlined as follows:

1. Manual Control
   a. To establish current
      (1) Snap switch.
      (2) Knife switch
      (3) Manually operated contactor.
      (4) Drum switch.
   b. Establish current and add overload protective device.
      (1) Snap switch with overload element.
      (2) Knife switch with fuse or thermal cutout.
      (3) Manual contactor with overload protective device, also reduced voltage starting compensator.
      (4) Drum switch with overload protection.
   c. Establish current and add overload and low voltage protective devices.
      (1) Not used
      (2) Not used

(3) Manual contactor or reduced voltage compensator with overload and low voltage release.
(4) Drum switch equipped with latch coil to give low voltage release.

2. Automatic Control:
   a. To start on full voltage.
      (1) Without overload device
      (2) With overload device.
      (3) With combination overload device and knife switch.
   b. Reduced voltage starting.
      (1) Primary resistance type starter.
      (2) Auto compensator type.
      (3) Reactance type.

PILOT CONTROLS

In selecting pilot control devices to operate in conjunction with either manual or automatic motor control, it is necessary that they be classified as follows:

1. *Two Wire Control.* Most thermostats, float switches, and pressure regulators, provide two wire control which gives low voltage release. A three position pilot switch can be used in connection with this method and thus provide manual control. With a low voltage (12 or 20 volt) control circuit it is desirable to use a low voltage thermostat. When this type of thermostat is used it will be found that a saving in the wiring cost results. When using the low voltage thermostat on a control circuit a relay and transformer panel should be used instead of the low voltage coil on the starter.

2. *Three Wire Control.* Momentary contact start and stop push button stations are usually furnished as standard accessories with automatic starters, which gives low voltage protection. This control cannot be used in combination with two wire pilot devices.

In selecting manual control for an alternating or a direct current motor, the common practice is to locate the control near the motor. When the control is installed at the motor, an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Frequently manual control is employed only as a device to give overload protection and another device is employed to start and stop the motor. Manual control is used particularly on small motors which operate unit heaters, small blowers, and room coolers in an air conditioning system. In other cases manual control in the form of drums, when used with multispeed motors, is only used as a speed setting device with the starting and stopping functions operated automatically through thermostats, and pressure switches.

Because of the increasing complexity of air conditioning systems, heating, ventilating and air conditioning equipment is being operated on automatic control with less dependence on manual operation and regulation.

Automatic control of motor starters may be accomplished by the use of remote push button stations, by a thermostat, float switch, pressure regulator or other similar pilot devices. An added advantage of automatic control is that the main wiring for the starter may be installed near the motor, while the starter may be operated by a control device located elsewhere. In the majority of air conditioning installations, requiring motors 1 hp and larger, two or three phase alternating current is usually supplied.

DIRECT CURRENT MOTOR CONTROLS

Air conditioning installations using direct current power are now only used where alternating current is not available. Direct current motors are always started through starters, which are devices using a resistance to be put in series with the armature circuit during starting only, the resistance being gradually cut out as the motor comes up to speed. The starting current is held within safe limits by the use of the resistance.

The speed of a direct current motor may be regulated by the following methods:

1. Speed regulation by field control—by using a device with resistance to be put in series with the field winding. After the motor has been started to be used to increase the speed of the motor above full field speed.
2. Speed regulation by armature control—by using devices with resistance to be put in series with the armature circuit to be used to reduce the speed of the motor below full field or normal speed.
3. Combinations of field and armature control, so that the starting, field control, or armature control may be combined in a single unit

Field control is usually preferred, depending on the size of the installation. For example, if a direct current motor were required with speed regulation between 1200 and 600 rpm, a choice of supplying a 1200 rpm motor with armature control or a 600 rpm motor with field control, both giving the same speed variation would be possible. While the 1200 rpm motor with armature control is lower in first cost than the 600 rpm motor with field control, the cost of operating the 600 rpm motor with field control is less and will save the difference in first cost over a period of time depending on the size of installation. A wide speed variation can be easily obtained in a direct current motor by using a combination of field and armature control.

SQUIRREL CAGE MOTOR CONTROL

To meet the requirements of various drives of an air conditioning system, three types of squirrel cage, two or three phase motors may be used:

1. Normal torque, normal starting current
2. Normal torque, low starting current.
3. High torque, low starting current.

Because of the large current inrush of the normal torque, normal starting current motor, central stations usually require current limiting starting equipment on such motors above 5 hp. To meet the starting current requirements, manual or automatic current limiting starting compensators are used. These compensators are equipped with 50, 65 and 80 per cent voltage taps, the 65 per cent tap being regularly furnished when the compensator leaves the factory. Motors 5 hp and smaller have starting currents within the requirements of central stations and manual or magnetic, full voltage control may be used.

The normal torque, low starting current motor has a starting current which is approximately 20 per cent less than the normal current motor on

full voltage and well within the required current limits on 30 hp sizes and smaller. This motor, therefore, lends itself to across-the-line control because no current limiting equipment is necessary. In selecting motors for fans, pumps, or blowers, it should be noted that while the cost of the normal starting torque, low starting current motor is higher, the cost of full voltage control is lower, so that the total cost of low starting current motors with across-the-line control is lower.

A magnetic starter with low voltage and thermal overload protection gives the most satisfactory service. These switches may be controlled by remote push button stations, thermostats, or pressure switches to meet the requirements of any particular installation.

The high torque, low starting current motor has a starting current approximately 10 per cent less than the normal torque, low starting current motor when started on full voltage. These motors, most commonly used on compressor drive, can be started directly across-the-line with manual or magnetic starters.

Adjustable varying speed motor control by terminal voltage regulation requires a tap-changing switch manually or magnetically operated. Such a control switch operates to alter the voltage applied to the motor by contacting different auto-transformer voltage-ratio taps or by changing the amount of resistance inserted in the primary or line circuit.

MULTISPEED MOTOR CONTROL

To make an installation more flexible, multispeed motors are available with two, three or four speed designs, with variable torque, constant torque or constant horsepower characteristics. Multispeed may be started by means of manual or magnetic starting equipment.

When using automatic magnetic control with two, three, and four speed separate winding or consequent pole motors, control is obtained from a remote point by means of a push button master switch. The various speeds of the motor are obtained from the master switch by simply depressing the correct push button, which is known as selective speed control. It is commonly used in the smaller theatre installations where the fan and motor is located backstage and the speed control is located in the lobby.

Magnetic multispeed motor controllers may also be provided with a compelling relay which makes it necessary that the operator press the first speed button before regulating the motor to the desired speed. This assures the operator that the motor is always started at low speed before the motor is adjusted to one of the higher speeds. Starting on low speed limits the starting current to the starting current of the low speed winding, and therefore, permits the use of motors in sizes larger than ordinarily permitted by central stations for full voltage starting.

Timing relays, which provide for automatic acceleration, may be used for control. With the automatic acceleration feature, it is only necessary to press the button for the desired speed. The motor will always start in low speed and automatically step up to the desired speed.

Where the change of speeds does not occur at regular intervals, and where it is only necessary to change from one speed to another to take

care of seasonal requirements, a manual drum speed selector may be used. This drum is used to select the proper motor speed while an automatic starter is used to start and stop the motor.

The smaller size speed selector drums rated 10 hp at 220 volts and smaller may also be used as a motor starter to make and break the current, as well as, serving as a speed selector device. Reversible or non-reversible drums may be supplied depending on the requirements of the installation.

In the large size drums, a separate contactor must be provided to make and break the current. The contactor may be any approved starter. Overload and low voltage protection may be accomplished by using a magnetic starter. No push button station is required, the handle switch on the drum having the same characteristics as a three wire push button station.

In selecting two speed motors for fan, pump, blower, or compressor drive it will be found that the two winding motors are more expensive than the single winding. The control for two speed, two winding motors is more economical and the combined price of the motor and contactor is only slightly higher. Because of the better performance of the two speed motor and the factor of safety in having two independent motor windings, the increased cost is considered worth the difference.

SLIP RING MOTOR CONTROL

When close speed regulation and low starting current is required slip ring or wound rotor motors are used. Slip ring motors are built for two classes of service, constant speed and adjustable varying speed. The motors for the two classes of service are identical, the only difference being in the secondary control used with the motors. Control for both primary and secondary of a slip ring motor is required.

The primary control for a constant or adjustable speed is the same type as used with squirrel cage motors. Manual or magnetic starters, across-the-line type, may be used depending on the installation.

The starting current and starting torque of a slip ring motor are almost entirely dependent on the amount of resistance in the secondary control and in the manner in which the secondary control is operated. The *National Electric Manufacturers Association* has adopted service classifications which allow a selection of resistors permitting a starting current on the first contact of resistance varying from approximately 25 per cent of full load current to approximately 200 per cent of full load current or more, and permitting the resistor to remain in the secondary circuit of the motor for a period varying from not more than 15 seconds during an interval of operation from 4 minutes to continuous.

Speed regulation of a slip ring motor is obtained by inserting resistance in the secondary circuit and usually provides for a 50 per cent speed reduction when the motor takes its full rated current at normal speed. As resistors are supplied for both fan duty and constant torque duty, care should be taken in selecting the proper resistors.

Slip ring motors when used with centrifugal pumps and fans should have fan duty resistors. Because of the low current inrush of the fan and pump

load a starting resistor *NEMA* classification No. 15 may be used. For speed regulation resistor, classification No. 93 should be selected. On a compressor drive using an unloader, a constant torque resistor classification No. 15 should be used. If the compressor is started under load, *NEMA* classification No. 56 or 76 are used. For constant torque speed regulation, resistor No. 95 is used.

SINGLE PHASE MOTOR CONTROL

Where three phase current is not available or where single phase operation is preferred, then single phase repulsion induction, capacitor type or multispeed single phase motors may be used. Since the starting currents of all single phase motors are required to be within the starting-current limits established by the local power-supply company, a suitable type of starter may be chosen from the following selection:

1. Enclosed two pole manually operated motor starters with thermal overload protection.
2. Enclosed two pole automatic motor starter operated by a push button, thermostat or similar device, with thermal overload relay and low voltage protection.
3. A manual or magnetic resistance type starter with low voltage protection.
4. A manual or magnetic control for pole changing motors and for adjustable varying speed motors using an auto-transformer or resistance in the primary circuit to obtain line (or terminal) voltage drop.

In selecting across-the-line control for single phase capacitor type motors it is usually very desirable to use three pole across-the-line starters. Control for multispeed, single phase capacitor motors may be selected from tables on three phase rating when consideration is given to the increased current and the necessary switching of connections.

PROBLEMS IN PRACTICE

1 ● When motors are being considered as prime movers, what are some of the basic considerations that determine the final selection of the correct unit?

a. The kind of current available for driving the necessary motors is a primary consideration. There are two groups of motors available for driving the equipment on any job, which are the direct current type or alternating current type. The proper group selection depends entirely on the current available

b It is also necessary to decide whether constant speed or variable speed operation is desired.

c. Consideration must also be given to the type of service required.

1. Whether variable torque or constant torque motors will be required.
2. Whether a high starting torque is required or whether a relatively small starting torque is required.

d. It is important to take into consideration the atmospheric conditions surrounding the motor location.

2 ● When using direct current motors: *a.* What three types are available as regards their windings; *b.* What four types are available with reference to their speed characteristics?

a. Shunt wound, compound wound, and series wound.

b. Constant speed, adjustable speed, adjustable varying speed, and varying speed.

3 ● With direct current motors as prime movers what type would you use: *a*. For driving a fan; *b*. For driving a compressor?

a. A fan requires a relatively small starting torque, therefore, a shunt wound motor would be ideal for this type service.

b. A compressor has a constant torque, therefore, a compound wound motor would be the proper selection for this duty.

4 ● What is one of the important factors that should be taken into account when a series wound direct current motor is being considered?

With a series wound motor the speed varies with the load, therefore this type should never be used where there is a possibility of the motor operating without being loaded. The resultant high speed may prove to be dangerous.

5 ● With the use of alternating current motors what two groups are generally considered?

Motors using single and polyphase power supply.

6 ● Under the alternating current group of motors what common types are available: *a*. For single phase duty; *b*. For polyphase duty?

a. Capacitor high torque, capacitor fan—(1) high torque, (2) low torque, capacitor start-induction run, repulsion induction, and split phase

b. Squirrel cage—(1) general purpose, (2) medium torque, (3) high torque, automatic start high torque, normal torque normal current, normal torque low current, high torque low current, slip ring wound rotor, synchronous high speed, synchronous low speed.

7 ● What is the most commonly used of the polyphase motors?

The squirrel cage induction motor is the type most generally used for ordinary application.

8 ● With the use of squirrel cage motors what speed characteristic is available and what construction is used to make these more flexible?

The squirrel cage motor is basically a constant speed motor. However both single phase and polyphase high slip motors are used for adjustable varying speed drive through the use of line voltage control When using an adjustable varying speed motor, particularly with a centrifugal fan and to a somewhat lesser extent, with a propeller fan, special care should be taken to assure that the fan is closely motored (*i e*, adequately loads the motor) in order to obtain the desired speeds under reduced speed operation. To make the squirrel cage motor more flexible, multispeed units are used quite frequently. These units may be single winding for the two speed unit or for different number of windings depending upon the number and combination of speeds required For two speed single winding units the second speed is always one-half of top speed

9 ● Differentiate between synchronous speed and full load speed of a motor.

Synchronous speed is the theoretical or no load speed. With the induction motor there is a certain amount of slip depending upon the load As a rule, at full load, the speed is approximately 96 per cent of synchronous speed, however, motor manufacturers generally list full load speeds on their motor name plates.

The synchronous type of motor has a full load speed which is the same as the synchronous.

10 ● What are the general requirements usually recommended by the power company with reference to connecting polyphase motors to the power line?

For motors up to and including 5 hp, normal torque, normal starting current type of units can be connected directly to the line.

For motors from 5 to 30 hp, both high torque and normal torque, low starting current types of units can be used with across-the-line type of control.

Above these sizes, it is necessary to furnish current limiting starting equipment

It is always advisable to check with local power companies as there are no standards for

connecting of loads on the power line and they are likely to vary with different power companies.

11 ● In controlling direct current motors what two methods are used, what speed ranges are obtained, and what is the relative efficiency of each method?

In controlling direct current motors, resistance is placed in either the armature circuit or the field circuit. For armature control, the speed is reduced with the increase of resistance. With the field control, the speed is increased with the addition of resistance in the field circuit.

For most listed direct current motors, it is possible to obtain operation up to a speed ratio of two to one with field control equipment. This type of control is used in connection with shunt wound motors for best results.

For speed adjustment by resistance in series with the armature circuit, a reduction of 50 per cent in speed can generally be obtained. This control can be used with either shunt or compound wound motors.

The field control method of changing speeds on direct current motors is the most efficient. Due to the large current in the armature circuit, this method results in a high loss when the speed is reduced any appreciable amount. It is well to remember that with field control only constant horsepower output is obtained, therefore, care should be taken that the motor at normal speed is large enough to care for any increase in load as a result of speeding up the unit.

12 ● What reduction in speed is possible and how is it obtained when alternating current slip ring motors are used?

Speed variation in slip ring motors is obtained by inserting resistance in the secondary circuit. This generally allows for a 50 per cent speed reduction when it is fully loaded at normal speed.

From 20 to 30 per cent speed reduction can be obtained through the use of line voltage control of an adjustable varying speed motor with a fan closely motored (*i e* , the fan approximately fully loads the motor).

Chapter 39

PIPING AND DUCT INSULATION

Heat Losses from Bare and Insulated Pipes, Heat Losses from Ducts, Low Temperature Insulation, Insulation of Pipes to Prevent Freezing, Economical Thickness of Pipe Insulation, Underground Pipe Insulation

INSULATION reduces the flow of heat where it is desired to maintain a temperature higher or lower than that of the surroundings. Its use contributes to the most economical operation of heating and refrigerating systems.

HEAT LOSSES FROM BARE PIPE

Heat losses from horizontal bare iron pipes, based on data obtained from tests conducted at the *Mellon Institute*, are given in Table 1. The

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE IRON PIPES

Expressed in Btu per hour per linear foot per degree Fahrenheit difference in temperature between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.543	0.573	0.605	0.638	0.656	0.742	0.796
¾	0.660	0.690	0.729	0.762	0.781	0.886	0.955
1	0.791	0.829	0.878	0.920	0.953	1.084	1.166
1¼	0.979	1.02	1.087	1.15	1.184	1.345	1.450
1½	1.09	1.15	1.220	1.29	1.335	1.520	1.640
2	1.34	1.40	1.491	1.58	1.637	1.866	2.015
2½	1.58	1.67	1.778	1.87	1.937	2.215	2.388
3	1.88	1.99	2.100	2.22	2.301	2.641	2.853
3½	2.13	2.24	2.380	2.51	2.585	2.972	3.215
4	2.36	2.50	2.650	2.78	2.873	3.312	3.582
4½	2.60	2.75	2.920	3.08	3.170	3.655	3.956
5	2.87	3.02	3.200	3.38	3.493	4.030	4.368
6	3.39	3.56	3.775	4.01	4.115	4.755	5.153
8	4.32	4.55	4.830	5.14	5.270	6.120	6.635
10	5.32	5.61	5.925	6.34	6.551	7.592	8.245
12	6.25	6.62	6.995	7.46	7.670	8.900	9.670

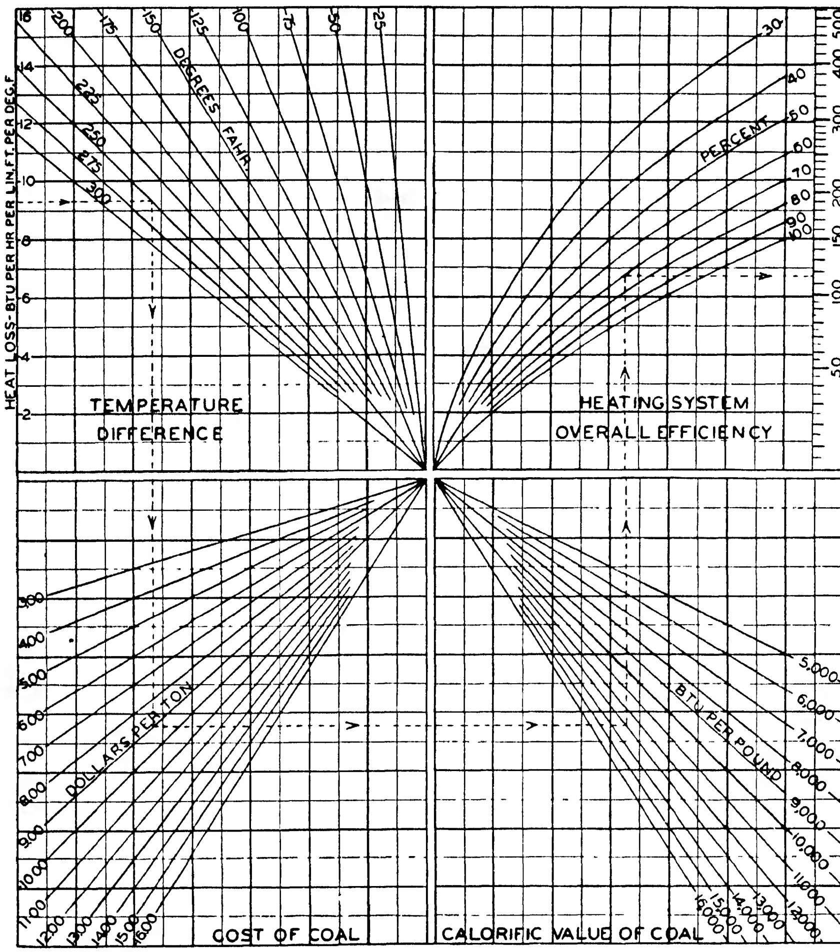


FIG. 1. CHART FOR ESTIMATING DOLLAR VALUE OF HEAT LOSS FROM BARE IRON PIPES. (SEE TABLE 1)[a]

[a]This chart is based on 100 linear feet per 1000 hours For fractions or multiples of these factors, multiply by proper percentage

monetary value of the loss of heat given in Table 1 may be obtained by means of Fig. 1 for various heating system efficiencies, temperature differences, and calorific values and costs of coal. To solve a problem, select the proper heat loss coefficient from Table 1 and locate this value on the upper left hand margin of the chart. Then draw lines in the order indicated by the dotted lines, the dollar value of the heat loss per 100 linear feet of pipe per 1000 hours being given on the upper right hand scale. In using this chart, the cost of coal should also include the labor for handling it, boiler room expense, etc.

TABLE 2. HEAT LOSS FROM HORIZONTAL BARE BRIGHT COPPER PIPE

Expressed in Btu per hour per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.180	0.210	0.218	0.229	0.299	0.338	0.355
¾	0.236	0.275	0.291	0.307	0.357	0.408	0.418
1	0.290	0.338	0.354	0.373	0.440	0.492	0.523
1¼	0.340	0.400	0.418	0.443	0.510	0.571	0.598
1½	0.390	0.463	0.473	0.507	0.598	0.671	0.710
2	0.490	0.525	0.600	0.628	0.719	0.813	0.851
2½	0.580	0.675	0.709	0.750	0.840	0.953	1.008
3	0.680	0.788	0.848	0.871	0.987	1.107	1.165
3½	0.760	0.888	0.946	1.000	1.114	1.235	1.307
4	0.940	1.000	1.045	1.107	1.210	1.361	1.456
4½					1.335	1.495	1.488
5	1.020	1.200	1.255	1.320	1.465	1.670	1.755
6	1.160	1.375	1.410	1.500	1.685	1.890	1.942
8	1.460	1.725	1.820	1.890	2.100	2.373	2.510

TABLE 3. HEAT LOSS FROM BRIGHT COPPER PIPE GIVEN ONE THIN COAT OF CLEAR LACQUER

Expressed in Btu per hour per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.240	0.265	0.282	0.307	0.401	0.461	0.478
¾	0.320	0.356	0.373	0.414	0.477	0.571	0.578
1	0.390	0.437	0.463	0.507	0.598	0.681	0.710
1¼	0.470	0.537	0.554	0.614	0.700	0.812	0.840
1½	0.540	0.612	0.645	0.714	1.208	0.966	0.990
2	0.690	0.762	0.818	0.892	1.005	1.164	1.201
2½	0.840	0.937	0.991	1.085	1.178	1.361	1.420
3	0.960	1.025	1.135	1.270	1.400	1.625	1.700
3½	1.100	1.250	1.318	1.442	1.580	1.845	1.905
4	1.241	1.400	1.480	1.556	1.750	2.040	2.130
4½					1.910	2.240	2.350
5	1.480	1.685	1.790	1.965	2.130	2.415	2.610
6	1.700	1.936	2.052	2.272	2.450	2.810	2.990
8	2.200	2.500	2.630	2.854	3.120	3.425	3.730

Heat losses from horizontal copper tubes and pipes with bright, bright lacquered and tarnished surfaces are given in Tables 2, 3 and 4[1].

In order to determine heat losses per linear foot of pipe from known losses per square foot, it is necessary to know the area in square feet per

[1]Heat Loss from Copper Piping, by R. H. Heilman (*Heating Piping and Air Conditioning*, September, 1933, p 458)

TABLE 4. HEAT LOSS FROM HORIZONTAL TARNISHED COPPER PIPE

Expressed in Btu per hour per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

NOMINAL PIPE SIZE (INCHES)	HOT WATER (Type K Copper Tube)				STEAM (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227 1 F (5 Lb)	297 7 F (50 Lb)	337 9 F (100 Lb)
	TEMPERATURE DIFFERENCE						
	50 F	80 F	110 F	140 F	157 1 F	227 7 F	267 9 F
½	0 250	0 287	0 300	0 321	0 433	0 500	0 530
¾	0 340	0 381	0 409	0 429	0 533	0 543	0 654
1	0 440	0 475	0 509	0 536	0 636	0 746	0 803
1¼	0 500	0 559	0 618	0 622	0 764	0 878	0 934
1½	0 580	0 656	0 710	0 750	0 904	1 053	1 120
2	0 730	0 825	0 890	0 957	1 101	1 273	1 364
2½	0 880	1 000	1 091	1 143	1 305	1 490	1 605
3	1 040	1 175	1 272	1 343	1 560	1 800	1 940
3½	1 180	1 350	1 454	1 535	1 750	2 020	2 170
4	1 460	1 500	1 635	1 715	1 941	2 240	2 430
4½					2 131	2 465	2 650
5	1 600	1 812	1 980	2 071	2 387	2 770	2 990
6	1 840	2 125	2 270	2 430	2 740	3 210	3 440
8	2 400	2 685	2 910	3 110	3 310	4 050	4 370

linear foot of pipe. Table 5 gives these areas for various standard pipe sizes, and Table 6 for copper tubing, while Table 7 gives the area in square feet for flanges and fittings for various standard pipe sizes.

Very often, when pipes are insulated, flanges and fittings are left bare due to the belief that the losses from these parts are not large. However, the fact that a pair of 8 in. standard flanges having an area of 2.41 sq ft

TABLE 5 RADIATING SURFACE PER LINEAR FOOT OF PIPE

NOMINAL PIPE SIZE (INCHES)	SURFACE AREA (SQ FT)	NOMINAL PIPE SIZE (INCHES)	SURFACE AREA (SQ FT)	NOMINAL PIPE SIZE (INCHES)	SURFACE AREA (SQ FT)
½	0 22	2	0 622	5	1 456
¾	0 275	2½	0 753	6	1 734
1	0 344	3	0 917	8	2 257
1¼	0 435	3½	1 047	10	2 817
1½	0 498	4	1 178	12	3 338

TABLE 6. RADIATING SURFACE PER LINEAR FOOT OF COPPER TUBING

TUBE SIZE (INCHES)	SURFACE AREA (SQ FT)	TUBE SIZE (INCHES)	SURFACE AREA (SQ FT)	TUBE SIZE (INCHES)	SURFACE AREA (SQ FT)
½	0 164	2	0 556	5	1 342
¾	0 229	2½	0 687	6	1 604
1	0 295	3	0 818	8	2 128
1¼	0 360	3½	0 949	..	. .
1½	0 426	4	1 080	.	.

TABLE 7. AREAS OF FLANGED FITTINGS, SQUARE FEET[a]

NOMINAL PIPE SIZE (INCHES)	FLANGED COUPLING		90 DEG ELL		LONG RADIUS ELL		TEE		CROSS	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2 68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5 47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15 41	20 41	19.58	26 26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a]Including areas of accompanying flanges bolted to the fitting

would lose, at 100 lb steam pressure, an amount of heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces.

HEAT LOSSES FROM INSULATED PIPES

The conductivities of various materials used for insulating steam and hot water pipes are given in Table 8. In this table the conductivities are given as functions of the mean temperatures or the mean of the inner and outer surface temperatures of the insulations. This method of stating conductivities makes it possible to readily calculate the heat loss through single or compound sections. It should be emphasized that the conductivities given in Table 8 for the various insulations are the average of

TABLE 8 CONDUCTIVITIES (k) OF VARIOUS TYPES OF INSULATING MATERIALS FOR MEDIUM AND HIGH TEMPERATURE PIPES[a]

TYPES OF INSULATING MATERIALS	MEAN TEMPERATURE				
	100 F	200 F	300 F	400 F	500 F
85 per cent Magnesia Type	0.425	0.465	0 505	0.550	0.590
Corrugated Asbestos Type (4 Plies per 1 in. thick)	0.530	0.650	0.770	0.890	
Corrugated Asbestos Type (8 Plies per 1 in. thick)	0 480	0 555	0.630	0.705	
Laminated Asbestos Type (30–40 Laminations per 1 in. thick)	0.360	0.415	0.470	0.525	0 585
Laminated Asbestos Type (20 Laminations per 1 in. thick)	0.545	0.605	0.665	0.725	0.785
Rock Wool Type	0.350	0.410	0.470	0.530	0.590
High Temperature Type (Diatomaceous Earth and Asbestos)	0.515	0.545	0.575	0.605	0.635
Brown Asbestos Type (Felted Fibre)	0.600	0.640	0.675	0.715	0 750

[a]R H Heilman, *Mechanical Engineering*, Vol 46 (1924), p 593

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH 85 PER CENT MAGNESIA TYPE INSULATION

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		120 F	150 F	180 F	210 F	227 1 F (5 Lb)	297 7 F (50 Lb)	337 9 F (100 Lb)
		TEMPERATURE DIFFERENCE						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
	½	0.744	0.754	0.764	0.774	0.779	0.802	0.814
	¾	0.672	0.681	0.689	0.697	0.701	0.721	0.731
	1	0.613	0.621	0.629	0.637	0.641	0.659	0.670
	1¼	0.562	0.570	0.577	0.585	0.589	0.606	0.617
	1½	0.532	0.539	0.546	0.553	0.557	0.573	0.582
	2	0.500	0.506	0.512	0.519	0.523	0.538	0.547
	2½	0.475	0.481	0.487	0.493	0.497	0.512	0.520
	3	0.455	0.461	0.467	0.474	0.477	0.492	0.500
1	3½	0.441	0.447	0.452	0.458	0.462	0.475	0.483
	4	0.429	0.435	0.441	0.446	0.449	0.463	0.471
	4½	0.420	0.425	0.431	0.437	0 440	0.453	0.460
	5	0.411	0.416	0.422	0.427	0.430	0.443	0.450
	6	0 402	0.408	0.413	0.419	0 422	0 435	0.442
	8	0.387	0.392	0.397	0.403	0.405	0.418	0.425
	10	0.375	0.380	0.385	0.390	0.393	0.405	0 412
	12	0.369	0.374	0.378	0.383	0.386	0.398	0.405
	½	0.617	0.625	0.633	0.642	0.646	0.665	0 676
	¾	0.550	0.558	0.566	0.573	0.577	0.596	0.606
	1	0.496	0.503	0.511	0.518	0.522	0.540	0.549
	1¼	0.453	0.459	0.465	0.472	0.475	0.490	0.498
	1½	0.424	0.430	0.436	0.442	0.445	0.459	0.467
	2	0 394	0.400	0.405	0.410	0.413	0.427	0.434
	2½	0.371	0.376	0.382	0.386	0.389	0.401	0.408
	3	0.352	0.357	0.362	0.367	0.370	0.380	0.387
1½	3½	0.339	0.343	0.347	0.351	0.354	0.364	0.370
	4	0.328	0.333	0.337	0.341	0.343	0.353	0.359
	4½	0.320	0.324	0.328	0.332	0.334	0.343	0.350
	5	0.312	0.316	0.320	0.324	0 326	0 336	0.342
	6	0.303	0.307	0.311	0.315	0.318	0.328	0.333
	8	0.287	0.291	0.295	0.299	0 301	0.311	0.316
	10	0.276	0.280	0.284	0.288	0.290	0.299	0.304
	12	0.272	0.275	0.279	0.283	0.285	0.294	0.299
	½	0.543	0.551	0.558	0.565	0.569	0.587	0.597
	¾	0.484	0.490	0.497	0.503	0.507	0.523	0.532
	1	0.433	0.439	0.445	0.451	0.454	0.467	0.476
	1¼	0.393	0.398	0 403	0.409	0.412	0.424	0.432
	1½	0.365	0.370	0.376	0.381	0.384	0.397	0.402
	2	0.338	0.343	0.347	0.351	0.354	0.364	0.370
	2½	0.316	0.320	0.324	0.328	0.331	0.341	0.347
	3	0.297	0.301	0.305	0.309	0.312	0.321	0.326
2	3½	0.284	0.288	0.292	0.295	0.297	0.306	0.311
	4	0.275	0.278	0.282	0.285	0.287	0.296	0.301
	4½	0.266	0.270	0.273	0.276	0.278	0.286	0.290
	5	0.258	0.262	0.265	0.268	0.270	0.278	0.283
	6	0.250	0.254	0.257	0.260	0.262	0.270	0.274
	8	0.236	0.239	0.242	0.245	0.247	0.255	0.258
	10	0.224	0.227	0.230	0.233	0.235	0.242	0.246
	12	0.219	0.222	0 225	0 228	0 230	0 237	0 240

Table 10. Coefficients of Transmission (U) for Pipes Insulated with Corrugated Asbestos Type Insulation (4 Plies per Inch Thickness)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297 7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227 7 F	267.9 F
	½	0.890	0.919	0.949	0.978	0.995	1.065	1.106
	¾	0.803	0 829	0.857	0.883	0.898	0 961	0.997
	1	0 731	0.756	0.780	0.804	0 818	0.876	0.909
	1¼	0 671	0.693	0.716	0.738	0.751	0.804	0.834
	1½	0.635	0.656	0.677	0.698	0.710	0.760	0.788
	2	0.595	0.615	0.635	0.656	0.667	0.715	0.742
	2½	0.567	0.586	0.605	0.624	0.635	0.680	0.705
	3	0.544	0 562	0.580	0.598	0 608	0.652	0.677
1	3½	0.527	0.544	0.561	0.578	0.588	0.631	0.654
	4	0.513	0.530	0.548	0.565	0.575	0.616	0.639
	4½	0.502	0.518	0.535	0 551	0.561	0.601	0.624
	5	0.490	0.507	0 523	0.539	0.549	0.588	0.611
	6	0 480	0.496	0 512	0 528	0.538	0.577	0.599
	8	0.462	0 477	0.493	0 508	0 517	0.554	0.575
	10	0.447	0.462	0.476	0 491	0 500	0 537	0 557
	12	0.441	0.456	0.470	0.485	0.493	0.529	0.550
	½	0.737	0.762	0 787	0.812	0.826	0.884	0 918
	¾	0.657	0 679	0.702	0 725	0.737	0.790	0.820
	1	0 594	0 614	0 634	0 654	0 666	0.713	0.740
	1¼	0.542	0 559	0.577	0.596	0 606	0.649	0.673
	1½	0.507	0 524	0 541	0.558	0.568	0.609	0.632
	2	0.471	0.487	0.503	0.519	0 528	0.565	0.587
	2½	0.443	0 458	0.473	0.488	0.497	0.533	0.553
	3	0.421	0 435	0.449	0 463	0.472	0 506	0.525
1½	3½	0.403	0 417	0.430	0.443	0 451	0 483	0 502
	4	0.393	0 405	0.418	0.432	0 439	0.471	0 489
	4½	0.383	0.394	0.407	0.420	0.428	0.460	0.476
	5	0.372	0 384	0 397	0.409	0 417	0.447	0 463
	6	0.362	0.374	0.387	0.399	0 406	0.436	0 452
	8	0.343	0.354	0 366	0 378	0 385	0 413	0.429
	10	0.328	0 339	0 351	0 362	0 369	0 397	0 413
	12	0.323	0.334	0.346	0.357	0 364	0.391	0 407
	½	0 648	0 670	0.692	0 713	0 726	0 779	0.810
	¾	0.578	0.598	0.617	0.637	0 648	0 694	0 720
	1	0.518	0.535	0.552	0 570	0 580	0.622	0.645
	1¼	0.469	0.485	0.501	0.517	0.527	0.566	0.587
	1½	0.438	0.452	0 467	0.481	0.490	0.526	0.545
	2	0.404	0.417	0 430	0.444	0 452	0.483	0.502
	2½	0.379	0.391	0.403	0.415	0.422	0.451	0.466
	3	0.356	0.367	0.378	0 390	0 397	0.425	0.440
2	3½	0.339	0.350	0.361	0.373	0.380	0.406	0.421
	4	0.328	0.339	0.350	0.360	0.367	0 392	0.406
	4½	0 318	0.328	0 339	0 350	0.357	0.381	0.395
	5	0.308	0.318	0.329	0.340	0 346	0.370	0.384
	6	0.299	0.309	0.319	0.329	0.335	0.358	0.371
	8	0.282	0.291	0.301	0.310	0 315	0.336	0.349
	10	0.267	0 276	0 285	0.294	0.299	0.319	0.332
	12	0 263	0 272	0.280	0 289	0 294	0 314	0.325

TABLE 11. COEFFICIENTS OF TRANSMISSION (*U*) FOR PIPES INSULATED WITH CORRUGATED ASBESTOS TYPE INSULATION (8 PLIES PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297 7 F (50 Lb)	337 9 F (100 Lb)
		TEMPERATURE DIFFERENCE						
		50 F	80 F	110 F	140 F	157 1 F	227.7 F	267 9 F
	½	0.801	0.820	0.838	0.857	0.868	0.913	0.939
	¾	0 723	0 739	0.756	0.773	0.783	0 824	0 847
	1	0 658	0.673	0 688	0.704	0.713	0.751	0.772
	1¼	0 606	0.619	0 633	0 647	0.655	0.688	0.707
	1½	0.573	0 586	0.599	0 612	0.619	0.652	0 670
	2	0.538	0.550	0 562	0.575	0 581	0.612	0.629
	2½	0.511	0.523	0 534	0.546	0.553	0.582	0.599
	3	0.489	0.501	0 512	0.524	0.531	0 558	0.575
1	3½	0 474	0 485	0 496	0 507	0.514	0 542	0.557
	4	0 461	0 472	0 482	0 493	0 500	0 527	0 542
	4½	0 451	0 462	0 472	0 482	0.489	0 515	0 530
	5	0.442	0 452	0 462	0 473	0 479	0.505	0.520
	6	0.432	0.442	0.452	0.463	0 468	0.493	0 508
	8	0 416	0.426	0 436	0.446	0.451	0 475	0 489
	10	0.402	0.412	0 421	0 430	0 435	0 459	0 473
	12	0.397	0.406	0.415	0.424	0.429	0.452	0 466
	½	0.664	0.679	0 695	0.711	0 720	0 759	0.780
	¾	0 593	0.607	0 621	0 636	0.643	0 677	0 697
	1	0.535	0 547	0.560	0 573	0.580	0 611	0.629
	1¼	0 488	0 499	0.510	0.522	0 528	0.556	0 572
	1½	0.457	0.467	0.478	0.490	0 496	0.522	0.537
	2	0 425	0.434	0 444	0.455	0.460	0.485	0 499
	2½	0.399	0.408	0.418	0.428	0.434	0.457	0.471
	3	0 378	0.387	0.396	0 405	0 411	0.433	0.446
1½	3½	0.363	0.371	0.380	0.388	0.393	0.415	0.427
	4	0.353	0.361	0.369	0.378	0 383	0 403	0.415
	4½	0.343	0.351	0.360	0.368	0.373	0.393	0.404
	5	0.334	0.342	0.350	0.358	0.363	0.383	0 394
	6	0.325	0.333	0.341	0 349	0.353	0.373	0.383
	8	0.309	0 316	0.324	0.332	0 336	0.355	0.365
	10	0.295	0.303	0 310	0.318	0 322	0 340	0.350
	12	0.291	0.298	0.306	0 313	0.317	0 335	0.344
	½	0.585	0.599	0.613	0 627	0.635	0.668	0.688
	¾	0.520	0.533	0.545	0 558	0.565	0.595	0 612
	1	0.465	0.476	0 487	0.498	0.504	0.532	0.547
	1¼	0.422	0.432	0.442	0.452	0.458	0.483	0.497
	1½	0.394	0.403	0 412	0.422	0.427	0.450	0.462
	2	0 364	0 372	0 380	0 388	0 393	0 415	0.427
	2½	0.339	0.347	0 355	0 363	0 367	0.387	0.398
	3	0.319	0 327	0.334	0.342	0 346	0.365	0 375
2	3½	0 304	0 311	0.318	0 326	0 330	0 349	0.358
	4	0 295	0.302	0.308	0 315	0 319	0 336	0 345
	4½	0.285	0.292	0.299	0.306	0 310	0 327	0.336
	5	0.278	0 284	0.290	0.297	0 301	0.317	0.326
	6	0.269	0.275	0.282	0.288	0 292	0 307	0.315
	8	0.253	0 259	0.265	0 270	0.273	0.288	0.296
	10	0.240	0.245	0.251	0.257	0.260	0.275	0 282
	12	0 236	0.241	0 247	0.253	0 256	0.270	0.277

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH LAMINATED ASBESTOS TYPE INSULATION (30 TO 40 LAMINATIONS PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		120 F	150 F	180 F	210 F	227 1 F (5 Lb)	297.7 F (50 Lb)	337 9 F (100 Lb)
		TEMPERATURE DIFFERENCE						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267 9 F
	½	0.605	0 620	0 635	0 650	0 658	0.695	0.716
	¾	0.546	0.560	0.573	0.586	0.594	0.627	0.645
	1	0.498	0.510	0 522	0 534	0.541	0.570	0.587
	1¼	0.457	0 468	0.480	0 491	0.497	0.525	0.540
	1½	0.432	0.442	0 453	0 464	0.470	0.496	0.511
	2	0.406	0.416	0 426	0 437	0.442	0.467	0.481
	2½	0.385	0.395	0.405	0 415	0.420	0.443	0.457
	3	0.370	0.379	0 389	0.398	0.403	0 425	0.438
1	3½	0 359	0.367	0.376	0.385	0.390	0.413	0.426
	4	0.349	0 358	0.366	0.375	0.380	0.402	0.414
	4½	0 341	0.350	0.359	0 367	0.372	0 393	0 405
	5	0 334	0.342	0 351	0 359	0.364	0.384	0.395
	6	0 327	0.335	0 343	0.351	0.356	0.376	0.387
	8	0.314	0.322	0.330	0 338	0.343	0.362	0.373
	10	0 304	0 312	0 320	0.328	0 332	0.350	0 361
	12	0.301	0.308	0 316	0.324	0.328	0.346	0.356
	½	0.502	0 514	0 526	0 539	0.546	0.577	0.595
	¾	0 450	0.461	0 473	0 484	0.490	0.517	0.532
	1	0.405	0 415	0.426	0 436	0.442	0.466	0.480
	1¼	0 369	0 378	0 387	0.396	0 401	0.423	0.435
	1½	0.343	0.352	0 361	0.370	0 375	0.397	0.409
	2	0 321	0 329	0 337	0 345	0.350	0.369	0.380
	2½	0 301	0.309	0.317	0 324	0 330	0 348	0.358
	3	0.286	0 293	0 301	0 308	0 313	0 330	0.340
1½	3½	0 274	0 281	0 288	0 295	0 300	0.316	0.326
	4	0 267	0.273	0 280	0 287	0 291	0.307	0.317
	4½	0 259	0 266	0 272	0 279	0.283	0 299	0 308
	5	0 253	0 260	0 266	0 272	0.276	0.291	0.300
	6	0 247	0 253	0.260	0.266	0.269	0.284	0.293
	8	0.234	0 240	0 246	0 252	0.255	0 270	0 279
	10	0 223	0 229	0 235	0 241	0.245	0 258	0.266
	12	0 221	0 227	0 232	0.238	0.241	0.255	0 263
	½	0 442	0 453	0 464	0 475	0 481	0.508	0.523
	¾	0 392	0 402	0.412	0 422	0.428	0.452	0 465
	1	0.352	0 360	0 369	0 378	0.383	0.405	0 417
	1¼	0 319	0 327	0.335	0.343	0.348	0 367	0 379
	1½	0 297	0.304	0 311	0.319	0 323	0 341	0.352
	2	0 274	0 280	0.287	0.294	0 298	0.314	0.324
	2½	0 256	0 262	0 269	0 275	0 279	0.293	0.302
	3	0.243	0.249	0 254	0 260	0.264	0.277	0.285
2	3½	0.231	0 236	0 242	0.248	0 251	0 265	0 273
	4	0 223	0.228	0.234	0.240	0 243	0.257	0.265
	4½	0.216	0.222	0.227	0.233	0.236	0.249	0.256
	5	0 210	0.215	0 220	0.225	0 228	0.241	0.248
	6	0.203	0.208	0 213	0 218	0 221	0.233	0.240
	8	0.191	0.196	0 201	0 206	0 209	0.220	0.227
	10	0.182	0.187	0 192	0 196	0.199	0.210	0 215
	12	0 178	0.183	0.187	0 192	0 195	0.205	0.210

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH LAMINATED ASBESTOS TYPE INSULATION (APPROXIMATELY 20 LAMINATIONS PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227 1 F (5 Lb)	297 7 F (50 Lb)	337 9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157 1 F	227.7 F	267 9 F
1	½	0.910	0.925	0.940	0.956	0.964	1.001	1.022
	¾	0.823	0.836	0.850	0.863	0.871	0.902	0.921
	1	0.748	0.760	0.773	0.785	0.792	0.823	0.840
	1¼	0.686	0.698	0.710	0.721	0.728	0.756	0.771
	1½	0.649	0.659	0.671	0.682	0.688	0.716	0.731
	2	0.610	0.620	0.630	0.640	0.647	0.671	0.685
	2½	0.581	0.590	0.600	0.609	0.615	0.638	0 651
	3	0.558	0.567	0.576	0.585	0.591	0.613	0.626
	3½	0.539	0.548	0.557	0.566	0.571	0.592	0.604
	4	0 524	0.532	0.541	0.551	0.556	0.577	0.589
	4½	0.514	0.522	0.530	0.539	0.544	0.564	0.575
	5	0.503	0.511	0.519	0.528	0.533	0.553	0.565
	6	0.492	0.500	0 509	0.517	0 522	0.542	0.553
	8	0.473	0.480	0.488	0 497	0.502	0.521	0.532
	10	0 458	0 465	0 473	0.481	0.485	0.504	0.514
	12	0.452	0.459	0.467	0.475	0.478	0.497	0.507
1½	½	0.755	0.767	0.780	0.793	0.800	0.831	0.848
	¾	0.674	0.685	0 697	0.708	0.715	0.743	0.759
	1	0.607	0.618	0.628	0.639	0.645	0.670	0.684
	1¼	0 553	0.562	0.572	0.581	0.587	0.610	0 622
	1½	0.517	0.527	0.536	0.545	0.550	0.572	0.584
	2	0.481	0.490	0.499	0 508	0.513	0.535	0.547
	2½	0.453	0.460	0.469	0.477	0.481	0.500	0.511
	3	0.429	0.436	0.444	0.452	0 456	0.475	0.485
	3½	0.412	0.419	0.427	0.434	0.438	0.456	0.465
	4	0.400	0.407	0 415	0.422	0.426	0.443	0.453
	4½	0.390	0.396	0.402	0.409	0.413	0.429	0.437
	5	0.380	0.386	0.393	0.400	0.403	0.418	0.427
	6	0.369	0.375	0.382	0.389	0.392	0.408	0.417
	8	0.351	0.358	0.364	0.370	0.374	0.388	0.397
	10	0.337	0.344	0.350	0.356	0.359	0.373	0.382
	12	0.332	0.338	0.344	0.350	0.353	0.367	0.375
2	½	0.664	0.675	0.687	0.698	0.704	0.732	0.747
	¾	0.591	0.601	0.611	0.621	0.627	0.652	0.665
	1	0.529	0.538	0.547	0.557	0.562	0.584	0.597
	1¼	0.480	0.488	0.497	0.505	0.510	0.529	0.540
	1½	0.445	0.453	0.462	0.470	0.475	0.494	0.504
	2	0.412	0.420	0.427	0.434	0.438	0.455	0.464
	2½	0.385	0.392	0.398	0.405	0.409	0.425	0.434
	3	0.364	0.370	0.376	0.382	0.385	0.400	0.408
	3½	0.346	0.352	0.358	0.365	0.368	0.382	0.390
	4	0 336	0.342	0.348	0.354	0.357	0.371	0.378
	4½	0.325	0.332	0.338	0.343	0.346	0.360	0.367
	5	0.316	0.322	0.327	0.333	0.336	0.349	0.356
	6	0.306	0.312	0.317	0.323	0.326	0.338	0.345
	8	0.288	0.293	0.298	0.303	0.306	0.317	0.324
	10	0.275	0.279	0.284	0.289	0.292	0.302	0.308
	12	0.269	0 274	0.278	0.283	0 286	0.296	0.302

TABLE 14. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH ROCK WOOL TYPE INSULATION

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		120 F	150 F	180 F	210 F	227 1 F (5 Lb)	297.7 F (50 Lb)	337 9 F (100 Lb)
		TEMPERATURE DIFFERENCE						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267 9 F
	½	0.631	0.644	0.658	0.672	0.680	0 712	0.730
	¾	0.569	0.581	0.593	0 606	0.613	0.642	0.659
	1	0.518	0.529	0.541	0.552	0.559	0.585	0.600
	1¼	0.476	0.486	0.497	0.507	0 513	0 537	0.551
	1½	0.450	0.460	0.470	0.480	0.485	0.508	0.522
	2	0.422	0.431	0.441	0.450	0.456	0.478	0.490
	2½	0.402	0.411	0.420	0.428	0.434	0.455	0.466
	3	0.385	0 394	0.402	0.411	0.415	0.435	0.446
1	3½	0.373	0.381	0.389	0.398	0.402	0.421	0.432
	4	0 363	0.371	0.379	0.387	0.392	0.411	0.422
	4½	0.355	0.363	0.371	0.379	0.383	0.402	0.413
	5	0.348	0.356	0.364	0.371	0.376	0.394	0 404
	6	0.341	0.348	0 356	0 363	0.368	0 386	0 396
	8	0 327	0.335	0.342	0 349	0.353	0.372	0.381
	10	0 317	0 324	0 331	0 338	0 343	0.360	0.369
	12	0 313	0.320	0 327	0.334	0.338	0.355	0.364
	½	0 523	0.534	0 545	0.556	0 563	0 590	0 606
	¾	0 468	0.477	0.487	0.497	0.503	0 528	0.542
	1	0.421	0.430	0 440	0.449	0.455	0 477	0.490
	1¼	0.383	0.391	0.399	0 407	0 412	0 433	0.444
	1½	0.359	0.366	0.375	0.383	0 387	0 407	0.419
	2	0.333	0.340	0.348	0.356	0.360	0.378	0.389
	2½	0.314	0.320	0.327	0 335	0.339	0.355	0.365
	3	0.296	0.302	0.310	0.317	0.321	0.337	0 347
1½	3½	0.286	0.291	0.298	0.304	0.307	0 323	0.332
	4	0.278	0.284	0.290	0.296	0.300	0.315	0.323
	4½	0.270	0.276	0.282	0.287	0 291	0.305	0.313
	5	0.263	0.269	0.275	0.280	0.284	0.298	0.305
	6	0 257	0 262	0.267	0.273	0.277	0 290	0.297
	8	0.244	0 249	0.254	0.260	0.263	0.276	0.283
	10	0.235	0 240	0.245	0.250	0 253	0.265	0.272
	12	0.230	0.234	0.239	0 245	0.247	0.260	0.267
	½	0.461	0.471	0.481	0.491	0.496	0.520	0.534
	¾	0.409	0.418	0.427	0.436	0.441	0.463	0.475
	1	0.366	0.374	0.382	0.390	0.395	0.415	0.427
	1¼	0.333	0.340	0.347	0.355	0.359	0.377	0.387
	1½	0.310	0.316	0.323	0.330	0.334	0.351	0.360
	2	0.286	0.292	0.298	0.304	0.308	0.323	0.331
	2½	0.268	0.274	0.279	0.285	0.289	0.302	0.310
	3	0.252	0.257	0.262	0.268	0.272	0.284	0.292
2	3½	0.241	0.246	0.251	0.257	0.260	0.272	0.280
	4	0.232	0.237	0.242	0.247	0 250	0.262	0.269
	4½	0.225	0.230	0.235	0 240	0.243	0.255	0 262
	5	0.218	0.223	0.228	0.233	0.236	0.247	0.253
	6	0.213	0.217	0.221	0 226	0.228	0 239	0 245
	8	0.200	0.204	0.208	0.213	0.215	0.225	0.231
	10	0.189	0 193	0.197	0.201	0.204	0.214	0 220
	12	0.185	0.190	0.194	0.198	0.200	0.210	0.216

values obtained from a number of tests made on each type of material, also that all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials, will of course, vary in conductivity to some extent from these values

The heat losses through six of the types of insulation given in Table 8 for 1, $1\frac{1}{2}$ and 2 in. thick materials, and for temperatures commonly encountered in engineering practice can be obtained from Tables 9 to 14 inclusive. The loss through other thicknesses of the materials, and for other hot water or steam temperature conditions may be obtained by interpolation. The heat loss coefficients given in Tables 9 to 14 are based on the conductivities in Table 8 and were computed from data given in Chapter 22, THE GUIDE 1931.

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of well-insulated surfaces the increases in losses due to air velocity are very small as compared with increases shown for bare surfaces, because of the fact that air flowing over the surface of the insulation can increase only the rate of heat transfer from surface to air, and cannot change the internal resistance to heat flow inherent in the insulation itself. The maximum increase in loss due to air velocity ranges from about 30 per cent in the case of 1 in. thick insulation, to about 10 per cent in the case of 3 in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface.

If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be sealed as tightly as possible. Pipe insulation out-of-doors should be provided with a waterproof jacket, and other outdoor insulation should be thoroughly weatherproofed.

HEAT LOSSES FROM DUCTS

The heat transmission through sheet metal duct walls is mainly a function of the surface character of the metal since the thickness of the metal itself is not enough to appreciably retard the flow of heat. In other words, the two surfaces provide the resistance to heat flow through the metal. The surfaces of black iron probably offer the least resistance to the flow of heat, while metals with brighter and smoother surfaces, offer greater resistance. For ducts in service at normal air velocities and temperatures, the coefficient of heat transmission for black iron is 1.6 Btu per square foot per hour per degree Fahrenheit difference between the mean temperature in the duct and the temperature of the surrounding air and for galvanized iron is 1.1 Btu per square foot per hour per degree Fahrenheit difference.

The heat loss from a given length of duct is expressed by:

$$H = k\,P\,L\left[\left(\frac{t_1 + t_2}{2}\right) - t_3\right] \tag{1}$$

The heat given up by the air in the duct is:

$$H = 0.24\,M\,(t_1 - t_2) = 14.4\,A\,V\,d\,(t_1 - t_2) \tag{2}$$

Equating 1 and 2 enables the determination of the temperature drop in the duct:

$$k\,P\,L\left[\left(\frac{t_1 + t_2}{2}\right) - t_3\right] = 14.4\,A\,V\,d\,(t_1 - t_2)$$

$$\frac{t_1 + t_2 - 2t_3}{t_1 - t_2} = \frac{28.8\,A\,V\,d}{k\,P\,L} \qquad (3)$$

where

H = heat loss through duct walls, Btu per hour.
k = overall heat transmission coefficient, Btu per square foot per hour per degree Fahrenheit temperature difference.
P = perimeter of duct, feet.
L = length of duct, feet.
t_1 = temperature of air entering duct, degrees Fahrenheit.
t_2 = temperature of air leaving duct, degrees Fahrenheit.
t_3 = temperature of air surrounding duct, degrees Fahrenheit
M = weight of air per hour through the duct, pounds
A = cross-sectional area of duct, square feet.
V = velocity of air in the duct, feet per minute, at specified temperature.
d = density of air at the specified temperature at which V is measured

In using the Formula 3 one of the duct air temperatures will be unknown and will be solved for by substitution of the other known or assumed values. The assumed values dependent upon the mean duct air temperature can be determined exactly by cut and try. A more exact formula[2] is available for determining the heat loss from a duct in case the duct is exceedingly long.

Heat losses for insulated ducts are given in the warm air column of Table 15. The losses are based on a uniform series of material conductivities at 86 F mean temperature and an air temperature of 50 F outside of the duct. The losses may be interpolated for odd material conductivities and temperatures. The conductivities of various materials will be found in Table 2 of Chapter 5. For cases where the surrounding air temperature is other than 50 F the losses may be selected on the basis of temperature difference.

Example 1. Determine the entering air temperature and heat loss for a duct 24 x 36 in cross section and 70 ft in length, insulated with ½ in of a material having a conductivity of 0.35 Btu at 86 F mean temperature, carrying air at a velocity of 1200 fpm, measured at 70 F, to deliver air at 120 F with air surrounding the duct at 40 F.

Solution Assume the entering air temperature to be 130 F Thus the mean temperature difference will be 85 F. Referring to the warm air column of Table 15 and interpolating for 90 F temperature difference, the overall heat transmission coefficient is found to be 0 516 Btu. From Table 4, Chapter 1 the density of air at 70 F and 29 92 in Hg is found to be 0 07423 lb per cubic foot. Substituting these and the other given values in Formula 3,

$$\frac{t_1 + 120 - (2 \times 40)}{t_1 - 120} = \frac{28\,8 \times 6 \times 1200 \times 0\,07423}{0\,516 \times 10 \times 70}$$

$$t_1 + 120 - 80 = 42\,62\,(t_1 - 120)$$

$$t_1 + 40 = 42\,62\,t_1 - 5114$$

$$5154 = 41\,62 t_1$$

$$123.8 = t_1$$

[2]Performance Tests of Asbestos Insulating Duct, by R H Heilman and R A MacArthur (A S H V.E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February, 1938, p 127)

Based on 123.8 F entering air temperature the new mean temperature difference will be 81.9 F and the new transmission coefficient will be 0.515. Resubstituting in Formula 3, t_1 becomes 123.9 F, which value is evidently exact within one tenth of one degree.

Substituting in Formula 1,

$$H = 0.515 \times 10 \times 70 \left[\left(\frac{123.9 + 120}{2} \right) - 40 \right]$$

$$H = 0.515 \times 10 \times 70 \times 81.95$$

$$H = 29{,}543 \text{ Btu.}$$

TABLE 15. HEAT TRANSMISSION THROUGH DUCT WALLS INSULATED WITH MATERIALS OF VARYING CONDUCTIVITIES[a]

Values are expressed in Btu per hour per square foot of flat surface per degree Fahrenheit difference in temperature between air inside and still air outside at 90 F for cold air and 50 F for warm air in ducts

Conductivity of Insulation at 86 F Mean Temp	Thickness of Insulation (Inches)	Cold Air			Warm Air			
		40 F	60 F	80 F	90 F	120 F	150 F	180 F
		Temperature Difference						
		50 F	30 F	10 F	40 F	70 F	100 F	130 F
0 200	½	0 319	0 323	0 328	0 324	0 330	0 337	0 344
	1	0 175	0 177	0 180	0 178	0 181	0 184	0 188
	1½	0 121	0 122	0 124		0 125	0 127	0 129
	2	0 092	0 093	0 095				
0 250	½	0 382	0 387	0 392	0 390	0 397	0 404	0 412
	1	0 214	0 217	0 220	0 218	0 221	0 225	0 229
	1½	0 149	0 151	0 153		0 154	0 156	0 159
	2	0 114	0 115	0 117				
0 300	½	0 440	0 445	0 450	0 448	0 457	0 466	0 475
	1	0 252	0 255	0 258	0 256	0 260	0 264	0 268
	1½	0 176	0 178	0 180		0 181	0 184	0 187
	2	0 135	0 137	0 139				
0 350	½	0 494	0 499	0 505	0 502	0 511	0 521	0 530
	1	0 286	0 289	0 292	0 290	0 295	0 300	0 306
	1½	0 202	0 204	0 207		0 208	0 211	0 215
	2	0 156	0 158	0 160				
0 450	½		0 596	0 602	0 599	0 610	0 621	0 633
	1		0 356	0 360	0 358	0 364	0 370	0 376
	1½		0 254	0 257		0 259	0 263	0 267
	2		0 198	0 200				
0 550	½		0 682	0 688	0 685	0 699	0 714	0 730
	1		0 417	0 422	0.418	0 425	0 432	0 440
	1½		0 302	0 305		0 307	0 312	0 317
	2		0 236	0 239				

[a]For round ducts less than 30 in diameter, increase heat transmission values by the following percentages:

THICKNESS OF INSULATION (Inches)	½	1	1½	2
21 to 30 in. Duct Diameter	1%	2%	3%	4%
12 to 21 in Duct Diameter	3%	5%	7%	9%

LOW TEMPERATURE INSULATION

Surfaces maintained at temperatures lower than the surrounding air are insulated to reduce the flow of heat and to prevent condensation and frost. The insulating material should absorb a minimum amount of moisture, for one reason that the absorption of moisture substantially increases the conductivity of the material. This property is particularly important in the case of surfaces to be insulated that are below the dew-point of the surrounding air. In such cases, due to vapor pressure difference, it is necessary to seal the surface of the insulating material against the penetration of water vapor which would condense within the material, causing a serious increase in heat flow, possible breakdown of the material and corrosion of metal surfaces. An insulating material with a high degree of moisture absorption might pick up moisture before application and then, when the seal is in place and the temperature of the insulated surface reduced, release that moisture to the cold surface.

The thickness of insulation which should be used to prevent condensation on pipes and flat metallic surfaces may be obtained from Fig. 2. The maximum permissible temperature drop is indicated at the point where the guide line passes through the horizontal scale at the left center of the chart. This temperature drop represents the difference between the dry-bulb temperature and the dew-point temperature for the conditions involved. (See discussion of Condensation in Chapter 7). The surface resistances used for calculating the family of curves in Fig. 2 are based on tests made on canvas covered pipe insulation surfaces at *Mellon Institute*. However, it has been found that the resistance for asphaltic and roofing surfaces is practically the same as for canvas surfaces, so that the curves may be followed with no alteration for surfaces commonly used.

Heat gains for pipes insulated with a material having a conductivity of 0.30 Btu per square foot per hour per degree Fahrenheit difference per inch thickness are given in Table 16.

Heat gains for insulated ducts are given in the cold air column of Table 15. The heat gains are based on a uniform series of conductivities at 86 F mean temperature and an air temperature of 90 F outside of the duct. The gains may be interpolated for odd material conductivities and temperatures. For cases where the surrounding air temperature is other than 90 F the gains may be selected on the basis of temperature difference.

INSULATION OF PIPES TO PREVENT FREEZING

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point. Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 17 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and

surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that an efficient insulation be utilized. This table is based on the use of a material having a conductivity of 0.30. The initial water temperature is assumed to be 10 F above, and the surrounding air tem-

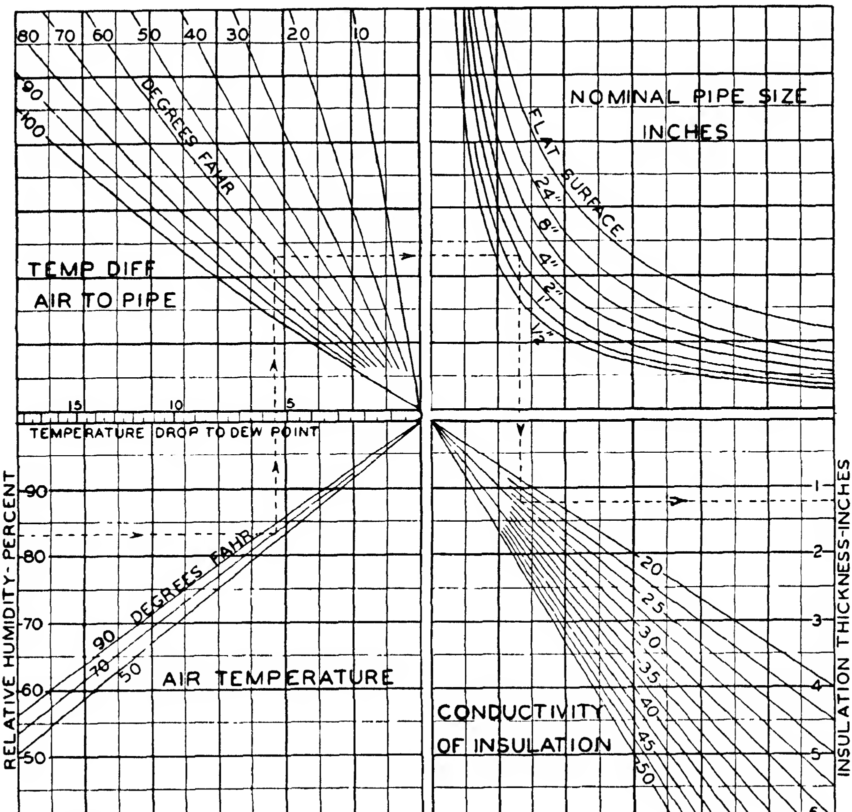


FIG. 2. THICKNESS OF PIPE INSULATION TO PREVENT SWEATING[a]

[a]Solve problems by drawing lines as indicated by dotted line, entering chart at lower left hand scale.

perature 50 F below the freezing point of water (temperature difference, 60 F).

The last column of Table 17 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is ad-

visable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions the following method of procedure may be used.

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column of Table 17. However, if the water enters the pipe at 34 F it will be cooled to 32 F in

TABLE 16. HEAT GAINS FOR INSULATED COLD PIPES

Rates of heat transmission given in Btu per hour per degree Fahrenheit temperature difference between fluid in pipe and surrounding still air

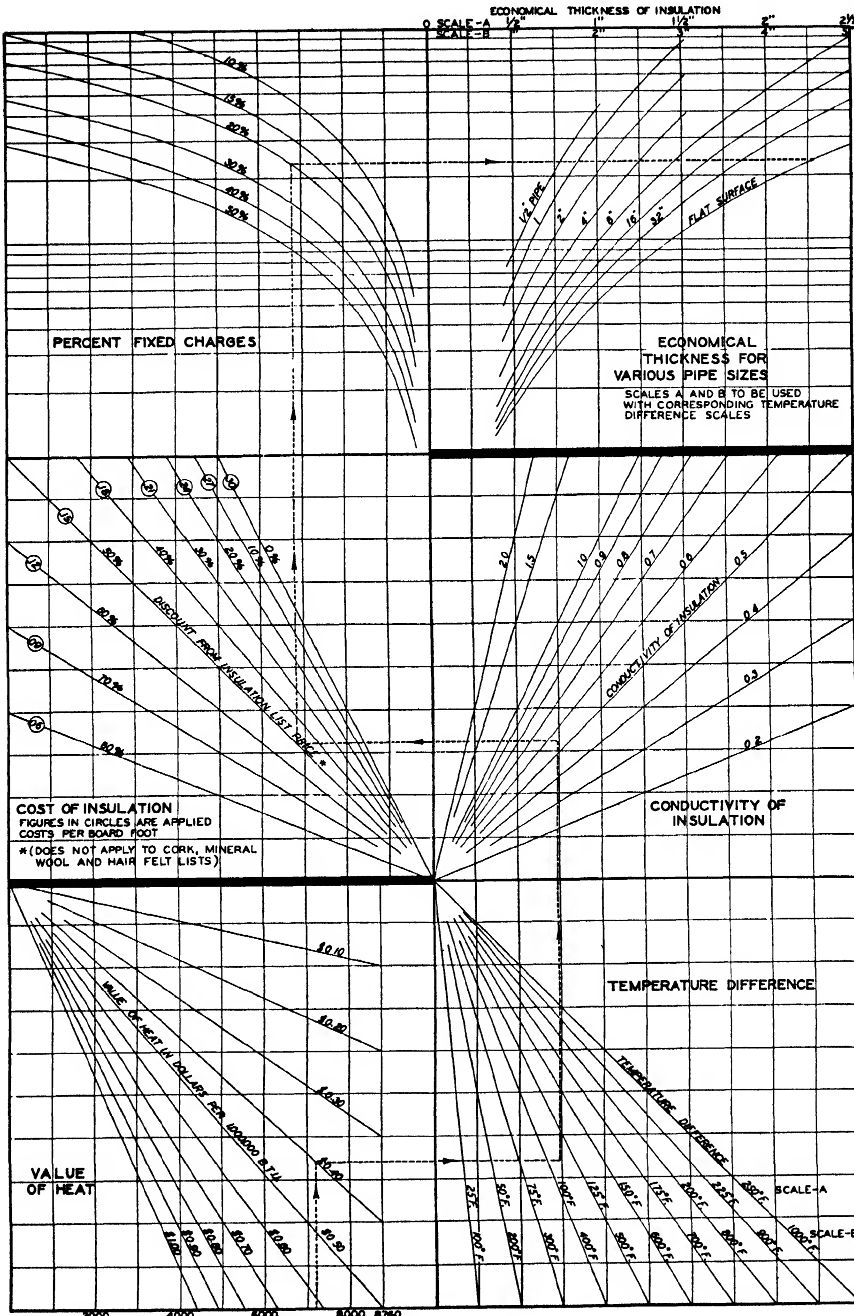
Based on materials having conductivity, k = 0 30

Nominal Pipe Size (Inches)	Ice Water Thickness			Brine Thickness			Heavy Brine Thickness		
	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface
½	1.5	0 110	0 502	2 0	0 098	0 446	2 8	0 087	0 394
¾	1 6	0 119	0 431	2 0	0 111	0 405	2 9	0 094	0 340
1	1 6	0 139	0 403	2 0	0 124	0 352	3 0	0 104	0 294
1¼	1 6	0 155	0 357	2 4	0 131	0 300	3 1	0 113	0 260
1½	1 5	0 174	0 351	2 5	0 134	0 270	3 2	0 118	0 238
2	1 5	0 200	0 322	2 5	0 151	0 244	3 3	0 134	0 214
2½	1 5	0 228	0 303	2 6	0 170	0 226	3 3	0 147	0 197
3	1 5	0 269	0 293	2 7	0 186	0 202	3 4	0 162	0 176
3½	1 5	0 295	0 282	2 9	0 191	0 183	3 5	0 176	0 167
4	1 7	0 294	0 248	2 9	0 209	0 176	3 7	0 182	0 154
5	1 7	0 349	0 239	3 0	0 241	0 165	3 9	0 202	0 138
6	1 7	0 404	0 233	3 0	0 259	0 150	4 0	0 228	0 130
8	1 9	0 455	0 201	3 0	0 318	0 140	4 0	0 263	0 116
10	1 9	0 559	0 198	3 0	0 383	0 135	4 0	0 309	0 110
12	1 9	0 648	0 194	3 0	0 438	0 131	4 0	0 364	0 108

one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is −38 F (temperature difference 80 F) instead of −18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

In making calculations to arrive at the values given in Table 17, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air have all been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.



(L. B. McMillan, *Proc. National Dist. Heating Assn.*, Vol. 18, p. 138).

FIG. 3. CHART FOR DETERMINING ECONOMICAL THICKNESS OF INSULATION

When water must remain stationary longer than the times designated in Table 17, the only safe way to insure against freezing is to install a steam or hot water line or to place an electric resistance heater along the side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

ECONOMICAL THICKNESS OF PIPE INSULATION

The thicknesses of insulation which ordinarily are used for various temperature conditions are given in Table 18. Where a thorough analysis of economic thickness is desired this may be accomplished through the the use of the chart, Fig. 3.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally to the right, to the curve representing the given pipe size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly.

UNDERGROUND PIPE INSULATION

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes. (See Chapter 42). Detailed data on commonly used forms of tunnels and conduit systems have been published by the *National District Heating Association*[3].

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

The economical thickness of insulation for underground work is difficult of accurate determination due to the many variables which have to be considered. As a result of theories[4] previously developed, together

[3]Handbook of the *National District Heating Association*, Second Edition, 1932

[4]Theory of Heat Losses from Pipes Buried in the Ground, by J. R Allen (A.S H V E TRANSACTIONS, Vol 26, 1920, p 335)

TABLE 17. DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES

Nominal Pipe Size (Inches)	Number of Hours to Cool Water to Freezing Point			Water Required to Flow to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches					
	2	3	4	2	3	4
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0 55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1 29	1 06
6	7.35	9.88	12 20	1 98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.43
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15 80	22.20	28.10	3.43	2.45	1.93

TABLE 18. THICKNESSES OF INSULATION ORDINARILY USED INDOORS[a]

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Thickness of Insulation		
		Pipes Larger Than 4 In	Pipes 2 In to 4 In	Pipes ½ In to 1½ In
0 to 25	212 to 267	1 in.	1 in.	1 in
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

[a]All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in greater than shown in this table, and covered with a waterproof jacket

TABLE 19. THICKNESS OF LOOSE INSULATION FOR USE AS FILL IN UNDERGROUND CONDUIT SYSTEMS

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In	Pipes 4 In to 10 In	Pipes Larger than 12 In	Pipes Less than 4 In	Pipes 4 In and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

with other experimental data which have been presented, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table 19 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 3. The data in Fig. 3 are based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

PROBLEMS IN PRACTICE

1 ● What precautions must be taken in selecting insulation used for covering pipe lines carrying materials at temperatures lower than the dew-point?

Materials intended for this service should be as moisture proof as possible and in addition an outer covering should be applied which is proof against diffusion of air and water vapor. If the material permits the diffusion of air, the air will reach a point in the covering where the temperature is below the dew-point. The condensed water will gradually accumulate until the covering becomes saturated, which will increase the conductivity and perhaps lower the mechanical strength of the covering until it becomes worthless.

2 ● Compute the total annual heat loss from 165 ft of 2-in. bare pipe in service 4000 hours per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained from Table 8, Chapter 1. The temperature difference between the pipe and air = 239.4 − 70 = 169.4 deg. By interpolation of Table 1 between temperature differences of 157.1 F and 227.7 F, the heat loss from a 2-in. pipe at a temperature difference of 169.4 deg is found to be 1.677 Btu per hour per linear foot per degree temperature difference. The total annual heat loss from the entire = 1 677 × 169.4 × 165 (linear feet) × 4000 (hours) = 188,000,000 Btu.

3 ● Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in Question 2. If the system is operating at an over-all efficiency of 55 per cent determine the monetary value of the annual heat loss from the line.

The cost of heat per 1 million Btu supplied to the system = 1,000,000 × 11.5 (dollars) ÷ 13,000 (Btu) × 2000 (lb) × 0.55 (efficiency) = $0.804. The total cost of heat lost per year = 0.804 × 188 (million Btu) = $151.15.[5]

4 ● If the steam line given in Question 2 is covered with 1-in. thick 85 per cent magnesia, determine the resulting total annual heat loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

By interpolation of Table 9 between temperature differences of 157.1 F and 227.7 F, the coefficient of transmission for 1-in. magnesia on a 2-in. pipe is found to be 0.525 Btu per hour per square foot of pipe surface per degree temperature difference at a temperature difference of 169 4 deg. The total hourly loss per square foot of insulated pipe will then be 0.525 × 169.4 = 89.04 Btu. From Table 5 the area per linear foot of 2-in. pipe is

[5]A closely approximate solution of this problem may be quickly made by use of the estimating chart given in Fig. 1.

found to be 0.622 sq ft. The total annual loss through the insulation = 89.04 × 0.622 × 165 (linear feet) × 4000 (hours) = 36,550,000 Btu. The annual bare pipe loss as determined in the solution of Question 2 was found to be 188,000,000 Btu. The saving due to insulation is then 188,000,000 − 36,550,000 = 151.350,000 Btu per year.

From the solution of Question 3 it was found that the heat supplied to the system cost $0.804 per million Btu; therefore, the monetary value of the saving = 0.804 (dollars) × 151.35 (million Btu) = $121.69, or 81.2 per cent of the cost when using uninsulated pipe.

5 ● The manufacturer's list price for 85 per cent magnesia insulation is $0.36 per linear foot for 1-in. (standard thick) material to cover a 2-in. pipe. Determine the period of time required for the saving found in Question 4 to pay for the cost of the insulation if it can be purchased and applied at 80 per cent of list price (20 per cent discount).

The applied cost of insulation = 165 (linear feet) × 0.36 (dollars) × 0 80 (net) = 47.52. Since the annual saving as found in Question 4 amounts to $121.69, the insulation will pay for its cost in 47.52 ÷ 121.69 = 0.3905 years; in other words, the cost will be repaid 2 56 times by the saving obtained in one heating season.

6 ● The conductivity of magnesia insulation is 0.455 at the mean temperature which will result under the conditions of Question 4. Estimate the most economical thickness of magnesia for application on the pipe when operating under the conditions which are given in the foregoing problems and when a 20 per cent return is required on the investment for insulation.

Use chart given in Fig. 3. Begin at the left bottom margin and proceed successively as shown by the dotted line example to the following essential data which are collected from the problems previously given:

4000 hours operation per year.
$0.804 value of heat, dollars per million Btu.
169 4 deg temperature difference.
0.455 conductivity of insulation.
20 per cent discount from list, cost of insulation.
20 per cent fixed charges, return on investment.
2-in. pipe size.

Solution of the problem by use of Fig 3 results in a required thickness of approximately 1.05 in. The nearest commercial thickness procurable is standard thick (1$\frac{1}{32}$ in) magnesia.

(It is of interest to note that the use of Fig 3 will generally result in solutions which, for all practical purposes, agree closely with the specifications for thicknesses given in Table 18).

7 ● Determine the minimum thickness of wool felt insulation having a conductivity of 0.30 necessary to prevent condensation of moisture on a 4-in. pipe carrying cold water at a temperature of 40 F when the surrounding air reaches maximum conditions of 90 F with a relative humidity of 90 per cent.

The difference between the temperature of the pipe and the surrounding air is 90 − 40 = 50 deg. For quick estimating purposes use the chart given in Fig 2. Enter this chart at the lower left margin on the 90 per cent relative humidity line and proceed horizontally to the right to intersect the 90 deg air temperature line. Project a line up to the 50 deg temperature difference line, and then horizontally to the right to the intersection with the 4-in. pipe size line. From this point proceed down to intersect the 0.30 line which denotes the conductivity of the insulation. Directly opposite this point of intersection the correct thickness of insulation is read from the scale on the lower right margin. This chart solution denotes that wool felt 2.4-in. thick is sufficient to prevent condensation. The nearest commercial thickness procurable is 2½ in.

For prevention of condensation as well as for protection against freezing, if the thickness determined theoretically cannot be had, it is better to apply the next greater thickness procurable rather than to use any lesser thickness because an additional factor of safety is thus obtained.

Chapter 40

ELECTRICAL HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Steam Heating, Electric Hot Water Heating, Heating Domestic Water Supply, Industrial Heating, Reversed Cycle Refrigeration, Auxiliary Electric Heating, Control, Calculating Capacities, Power Problems, Insulation

ELECTRIC heating is steadily assuming a more important place in heating, ventilating and air conditioning installations, accelerated in many territories by the load building efforts of the utilities which usually include reduced rates to encourage such installations. Electrical heating has a logical place in the heating industry because of its features of flexibility, cleanliness, safety, convenience and ease of control. Electrical heating practice has many basic principles in common with fuel heating, but there are also important differences. When heat units are delivered to each room by wire, no combustion process is necessary, either at a central plant or at the individual room units. The maximum output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air and it follows that the maximum total load on an electrical heating system is the total wattage of connected electric heaters, regardless of weather conditions. The real obstacle to the more general adoption of electric heating for buildings is the cost of the electricity itself. Because the heat units produced electrically are more costly, their conservation is of more relative economic importance than with fuel heating, so that sponsors of electric heating give greater attention to temperature-insulated building construction, and to economy by accurate controls.

All heat is a form of energy. Fuels hold stored chemical energy which is released into heat by combustion. Electrical power is a form of energy which can be released into heat by passing it through a resisting material. Both fuel and electric heating have two divisions: *first*, the conversion of energy into heat; *second*, the distribution and practical use of the heat after it is produced.

In converting the chemical energy of fuels into heat by combustion, there is necessarily a considerable variation in thermal efficiency. This is not true, however, when converting electric power into heat, because 100 per cent of the energy applied in the resistor is always transformed into heat. In electric heating practice the engineer need not be concerned about efficiencies of heat production, but rather about efficiencies of heat utilization. It is the engineer's problem to distribute the electrically

produced heat units in such manner as to obtain conditions of maximum comfort with the minimum consumption of electricity.

DEFINITIONS

Definitions of terms used in fuel heating are given in Chapter 45. The following terms apply particularly to electric heating:

Electric Resistor: A material used to produce heat by passing an electric current through it.

Electric Heating Element: A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Electric Heater: A complete assembly of heating elements with their enclosure, ready for installation in service.

RESISTORS AND HEATING ELEMENTS

Solids, liquids, and gases may be used as resistors, but most commercial electric heating elements have solid resistors, such as metal alloys, and non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by an alternating current of electricity passing through it. One of the more common resistors is nickel-chromium wire or ribbon which, in order to avoid oxidation, contains practically no iron.

Commercial electric heating elements are made in many types. Some have resistors exposed to the air being heated. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. This type of element is used extensively for operation at high temperatures when radiant heat is desired, also at low temperatures for convection and fan circulation heating, especially in large installations.

Some elements have metallic resistors embedded in a refractory insulating material, encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat-dissipating area. Elements are made in many forms, such as strips, rings, plates and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, and in some types of convection air heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters. Cloth fabrics woven from flexible resistor wires and asbestos thread, are used for many low temperature purposes such as heating pads and aviators' clothing.

ELECTRIC HEATERS

Electric heaters may be divided into three groups, conduction, radiant and convection.

Conduction electric heaters, which deliver most of their heat by actual contact with the object to be heated, are used in such applications as aviators' clothing, hot pads, foot warmers, soil heaters, ice melters, and water heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have high temperature incandescent heating elements and reflectors to concentrate the heat rays in the desired directions. The immediate and pleasant sensation of warmth which is caused by radiant heat makes this type desirable for temporary use where the heat rays can fall directly upon the body. They are not satisfactory for general air heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. A typical radiant heater is shown in Fig. 1.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in much the same manner as steam and hot water radiators or convectors, They generally have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from

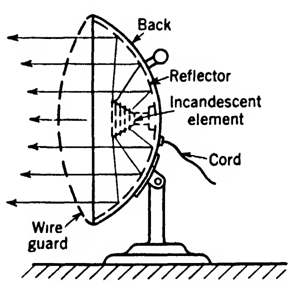


FIG. 1. TYPICAL RADIANT HEATER

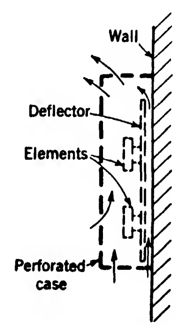


FIG 2. CONVECTOR HEATER

the floor line. See Fig. 2. The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water and steam add nothing to the efficiency of an electric heater and entail expensive construction and maintenance.

UNIT HEATERS

Unit electric heaters include a built-in fan unit which circulates room air over the heating elements. Heaters of this type are manufactured in many designs and sizes, and can be located in much the same manner as steam unit heaters.

Electric unit heaters are used in industrial plants, sub-stations, power houses, pumping stations, etc., where the power rate for electric heating is found to be favorable. The best location for the heater depends upon local circumstances as they can be mounted either on the ceiling to direct the air downward, on the side wall about 7 ft from the floor, or near the floor line. Variations in design are necessary for different locations, but typical arrangements are indicated in Figs. 3, 4, and 5.

The arrangement of the wiring circuits is very important for electric unit heaters. In principle they are all the same and include as essential elements an automatic control panel, a thermostat, and a master hand

switch. All heaters should be designed with a safety thermal trip wired in series with the magnet coil of the control panel and with the hand switch and thermostat. A typical wiring diagram is shown in Fig. 6. This applies to a single phase power supply, but for 3 phase the only difference is to have a 3-pole panel and a heater arrangement for 3-phase connection.

Portable unit heaters are useful for temporary work, such as drying out damp rooms, or for warming rooms during construction.

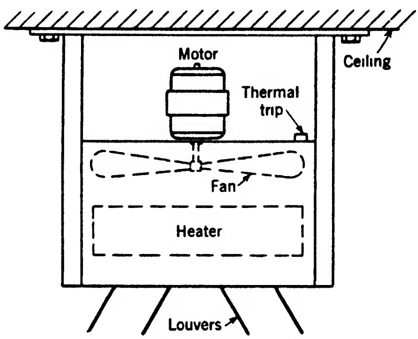


FIG. 3. CEILING MOUNTED UNIT HEATER

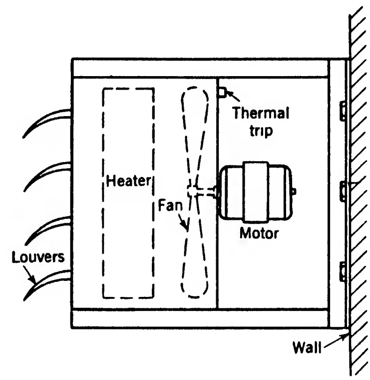


FIG 4. WALL MOUNTED UNIT HEATER

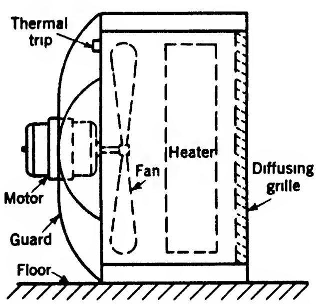


FIG. 5. FLOOR MOUNTED UNIT HEATER

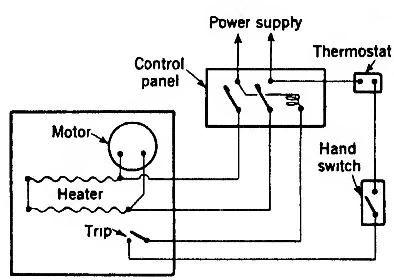


FIG 6. WIRING DIAGRAM FOR UNIT HEATER

CENTRAL FAN HEATING

Electric heating elements can be used for the prime source of heat in a central fan electric heating system or in the heating phase of a complete air conditioning system. They can be used in the same manner as steam served heating units for tempering, preheating or reheating the air at the main supply fan location and as booster heaters at the delivery terminals of the duct system. In the humidification phase of air conditioning electric heating elements can be used to provide moisture by the evaporization of water, or for controlling air washer dew-point temperatures when mounted as preheating units on the intake side of the air washer. (See Chapter 21).

In coordinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature source of heat for any given pressure and a change in air volume flowing over steam coils does not greatly affect the temperatures of the delivered air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, having a constant input of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered. This occurs because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is

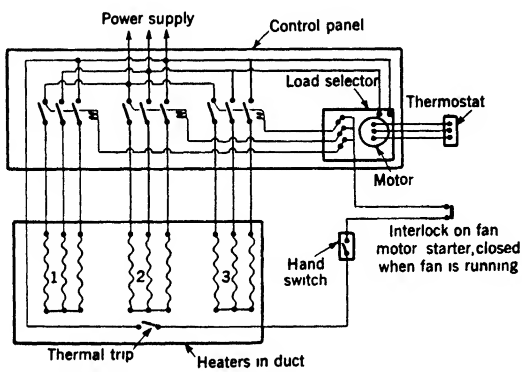


FIG 7. WIRING DIAGRAM OF A SELECTING CONTROL FOR A FAN SYSTEM

constant unless some compensating action is performed by control. Automatic variation of the electrical heat input synchronized properly with the air flow has been successfully applied to central fan systems. A typical wiring diagram of an automatic modulating system for central fan heating is indicated in Fig. 7.

Electric heaters are useful in balancing the heat distribution in central fan heating systems. Even in those instances where steam is the principal heat source, the temperature of individual rooms can be controlled locally by separate electric booster heaters. These heaters can be installed in branch ducts or behind the air outlet grilles in each room. With this arrangement, the central heating unit distributes air at an average temperature, controlled from a thermostat centrally located, such as in the main return duct. The electric booster heaters may be controlled by thermostats mounted in each individual room which permits the occupant to maintain any desired temperature independent of the rest of the building.

ELECTRIC STEAM HEATING

Electric steam heating differs from fuel heating only in the use of *electric boilers* to generate steam. Electric steam boilers are entirely automatic and are well adapted to intermittent operation. Small electric boilers

usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used on either direct or alternating current since the heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood that the heating elements will burn out, they should be of substantial construction, with a low heat density per unit of surface area. Provision should be made for cleaning off deposits of scale which restrict the heat flow. A typical

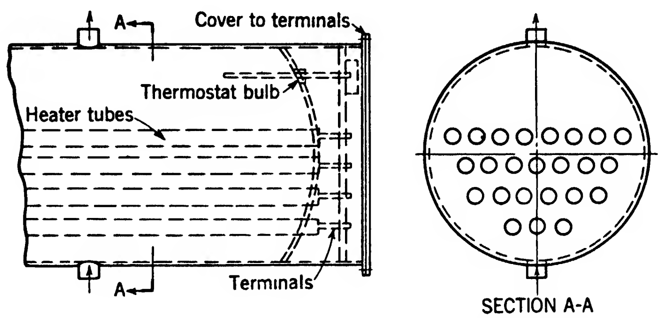


FIG. 8. RESISTANCE TYPE BOILER FOR STEAM OR HOT WATER

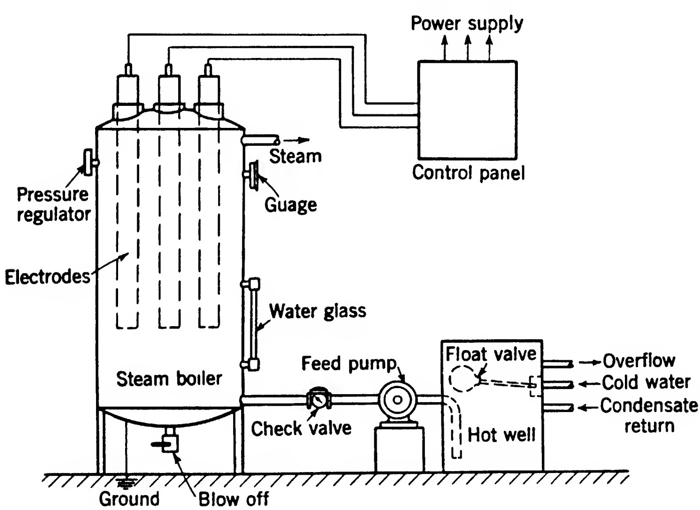


FIG. 9. DIAGRAMMATIC ARRANGEMENT OF AN ELECTRODE BOILER

resistance type of hot water or steam boiler is shown in Fig. 8. Large electric boilers are usually of the type employing water as the resistor. Only alternating current can be used, as direct current would cause electrolytic deterioration. Large boilers of this kind have electrodes immersed in the water where heat is generated directly. Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes, and for sterilizers, jacketed vessels and pressing machines which need a ready supply of steam. It sometimes is economical

to shut down the main plant fuel burning boilers when the heating season ends, and to supply steam for summer needs with small electric steam boilers located close to the operation. In general, electric steam heating is confined to auxiliary or other limited applications. If the heating system is designed to use electricity exclusively, steam generating or distributing equipment is superfluous. A diagrammatic arrangement of an electrode boiler is shown in Fig. 9.

ELECTRIC HOT WATER HEATING

Electric water heating, using an electric boiler in place of a fuel burning boiler, like electric steam heating, is generally confined to auxiliary or other limited applications. The use of insulated water storage tanks in which to store heat generated by electricity during off-peak hours at extremely low rates, is a development which has some special applications.

In this system of heating, the primary storage tank is simply a large, well-insulated, pressure type steel tank, equipped with electric heating elements connected to line with automatic time switches, which also have automatic limit controls for temperature and pressure. The heating system installed in the building may be of any standard individual radiator or fan-served indirect type or with provisions for the heating and humidification phases of an air conditioning system. A system of this kind requires very careful design to avoid excessive overall radiation losses during periods of low heat demand. It is also important to provide for sudden changes in heat demand. A typical hot water heating boiler is illustrated in Fig. 8.

HEATING DOMESTIC WATER SUPPLY [1]

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable, and in many sections of the country low electric rates have been established by the electric utilities to secure this load. In some districts, rate schedules divide the current used for water heating into two classifications, regular and off-peak. A time switch automatically limits use of the off-peak heating element to the hours of off-peak load, while the regular heating element is a stand-by at all times. Storage of this two-element type of water heater is larger than average to carry over the periods when the off-peak element is timed out, without too frequent demands on the regular heating element which takes the higher domestic lighting service rate. Some utilities now offer a schedule which, beyond a stipulated minimum, lowers the rate for all service if an electric water heater is installed.

A comprehensive survey covering United States and Canada shows a rapidly growing use of electric water heaters, although the per cent of saturation, based on the total number of domestic power customers, is still low. Public acceptance is effected by the cost of other competitive fuels, by the electric power rate, and by the temperature of the cold water supply.

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[1]Test Results of Electric Water Heaters, by C. G. Hillier (A.S.H V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1936, p. 632)

Fourth Annual Survey, by B J Martin (*Electric Light and Power*, March, 1937).

Competition with other fuels, especially gas, seems to be the major controlling factor. The first cost of electric storage heaters is also greater than for gas, owing to the need for larger tank storage due to off-peak service and slower recuperating capacity.

It is often desirable to connect an electric heater to a residential system having a coil in the fire-box of the furnace. In this case it is important to make the proper connections in order to benefit by any heat obtained from the furnace and at the same time to prevent dangerous overheating. The proper piping connections are shown in Fig. 10, and in this case the electric heater will only furnish heat when insufficient heat is supplied from the furnace. This arrangement has a further advantage in the summertime in that the bare tank through which the cold water passes on its way to the electric heater serves as a tempering tank, absorbing considerable warmth from the basement air and requiring the use of less energy in the electric heater.

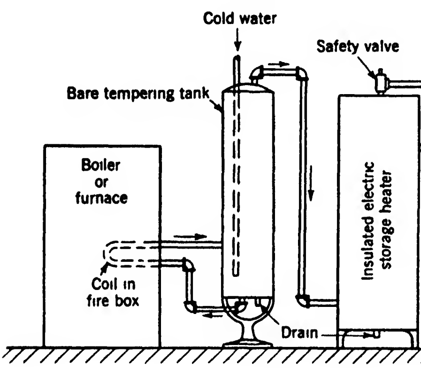

FIG. 10. PIPING ARRANGEMENT FOR CONNECTING ELECTRIC WATER HEATER TO FIRE-BOX COIL

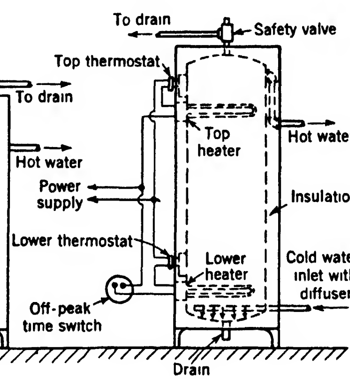

FIG. 11. DOMESTIC HOT WATER HEATER FOR OFF-PEAK SERVICE

A typical domestic hot water heater as shown in Fig. 11 is arranged with upper and lower heating elements for the usual type of off-peak heating service for which most utilities have especially attractive low power rates. The lower heating element is under the control of the off-peak time switch. However, the upper heating element is usually connected to the line so that in case the supply of hot water in the tank becomes exhausted the top thermostat can turn on the top heater and heat a small supply of water. The top heater will not heat the water in the tank below its location, but when the off-peak period arrives the lower heater is turned on and the entire tank becomes heated.

INDUSTRIAL ELECTRICAL HEATING

Electric heating elements have been successfully developed for industrial work such as annealing, brazing, carburizing, enameling, forging, ceramic firing, hardening, metal melting, nitriding and process heating. Industrial ovens and furnaces where precise control of temperature is necessary can be very successfully operated with electric resistance ele-

ments at temperatures as high as 1800 F. For higher temperatures the electric arc or high frequency induction methods are often used. Electric heaters for heating oil to high temperatures for secondary circulation in process work are used as a substitute for superheated steam. Special oil can be electrically heated as high as 700 F and pumped at a pressure just sufficient to cause flow. When used in heating coils or jacketed vessels, this gives a safe and convenient automatic system for moderate-sized installations. Pitch, waxes, and many chemicals are successfully heated by electricity, but require careful design and adequate automatic control.

REVERSED CYCLE REFRIGERATION [2]

Reversed refrigeration is frequently referred to as a *heat pump* since the electric motor driving the refrigerating compressor furnishes the motive power to transfer heat from one temperature to a higher temperature level. The compressor acts as a reversible refrigerating unit to extract heat from the outdoor air in winter and deliver it indoors for heating purposes, and, by a reversal, to extract heat from the indoor air in summer and discharge it outdoors.

In normal use a refrigerating machine is arranged to remove heat and the heat removed is dissipated to the condenser cooling water. The driving energy is converted into heat most of which is added to the heat removed and extracted. In so-called reversed refrigeration the heat removed together with the heat converted from the driving energy is utilized to heat the building. This conservation of the heat converted from the driving energy enables the reversed refrigeration to show a better performance in heating service than straight refrigeration can show in cooling service. For a detailed description of this cycle see Chapter 23.

AUXILIARY ELECTRIC HEATING

In conjunction with heating systems of other types, an auxiliary electrical heating arrangement is a convenient means of caring for mild days in the spring and fall which require little heat to make a building comfortable. Likewise, such electrical heating might be used on abnormally cold days to help out the main heating system and by this means reduce the necessary size of the system.

Because of the feeling of comfort that a radiant type heater gives, bathrooms may be heated electrically with this type of heater while the rest of the house is cared for by some other system. Offices and rooms which require heat at periods when the main heating plant is shut down

[2]Cooling Homes, A Field for Refrigeration, by A R Stevenson, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926.

The Heat Pump, An Economical Method of Producing Low-grade Heat from Electricity, by T G N. Haldane (*Electric Review*, Vol 105, p 1161-1162, December 27, 1929, and *I. E. E Journal*, Vol. 68, p. 666-675, June, 1930).

Edison Building Heated and Cooled by Electricity, by H. L. Doolittle (*Power*, Vol 74, p 384, September 8, 1931).

House Heating by Pump with 5 to 1 Pick-up Ratio, by Gilbert Wilkes and R. E Marbury (*Electrical World*, Vol 100, p. 828, December 17, 1932)

An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan (A.S H.V E TRANSACTIONS, Vol 41, 1935, p 307).

Using the Reversed Cycle Refrigerating Principle for a Self-Contained Heating and Cooling Unit, by Henry L. Galson (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1935, p. 497)

can be conveniently cared for electrically. Fan type unit heaters delivering warm air at the floor zone are very effective.

CONTROL

Because the efficiency of electric heat production is the same for large or small units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Radiant heaters are usually controlled manually but new methods for automatic control are described in Chapter 41 on Radiant Heating. For all convection and fan circulation heaters thermostatic control is essential for economical operation. For duct systems having a variable volume of air flow the electric heater control must automatically vary the heat input in coordination with the changes in air volume and demand for heat.

CALCULATING CAPACITIES

The methods of calculating heat losses outlined in Chapters 6, 7, and 8 may be used for electric heating exactly as for fuel heating. The total heat requirements in Btu per hour may then be converted into the electrical rating of an equivalent heating system by using the equation:

$$\frac{\text{Total Btu per hour}}{3415} = \text{kw rating of required electric heating} \qquad (1)$$

For comparison with steam radiation:

$$\frac{\text{3415 Btu (one kwhr)}}{240} = \text{14.2 sq ft of steam radiation}$$

While many empirical rules based on cubic contents, floor areas, etc., are used in steam heating practice, they should never be used to determine size of equipment for an electrical heating installation.

POWER PROBLEMS

The first point to determine is the cost of the power which is available for electric heating. Unlike fuels, there is no uniform cost for electric power because of the unequal cost of distribution to large and small users. The fact that electricity cannot be economically stored, but must be used as fast as it is generated, makes it impossible to operate power plants at uniform loads; hence, even the time of use may affect the cost of power.

Special low rates are sometimes available during certain prescribed hours of use, but wherever the use of power is unrestricted a demand charge based upon the rated connected load of each heating device must form part of the basic rate structure, so that unlike fuel heating, the cost of operating electric heating systems depends not only upon the actual energy used but also upon the demand charge for the available electrical service.

Homes are almost universally supplied with lighting current of 115 volts, which can only be used economically for small heaters. Usually

the service lines will not permit more than plug-in devices. The Underwriters permit approved heaters of 1320 watts or less to be plugged into approved baseboard receptacles. Where homes have 230 volt service for cooking and water heating, and rates are favorable, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 220, 440 or 550 volts. All polyphase heaters should be balanced between phases.

INSULATION

The value of building construction which incorporates built-in insulation to reduce the outward heat loss in winter and the inward heat gain in summer has been placed in the spotlight by the increasing adoption of complete air conditioning. With electric heating, adequate insulation is very important and will pay even better returns on the investment than for less expensive fuels.

PROBLEMS IN PRACTICE

1 ● Under what conditions are electric heaters most feasible?

a. When electric power rates are low and other fuels high in cost.

b. Where the total required heat is not great and cost of attention tends to offset the higher actual energy cost of electricity.

c. Where saving of space, elimination of a chimney and lower first cost of equipment are deciding factors.

d Where intermittent and local auxiliary use avoids the necessity of keeping up steam with large losses in long pipe lines.

e. Where accurate local automatic control reduces the total heat losses enough to compensate for the higher energy rate for electricity over other fuels

f. For isolated or unattended rooms and small buildings where other heat sources are not readily available.

g. For underground rooms and vaults where the return or disposal of condensation is difficult or where freezing may occur at times

h. Where corrosive conditions make pipe lines expensive to maintain.

i. Where the use of power for heating can be staggard to avoid periods of other use such as large motors, and thus prevent an increase of the basic demand charges for electrical service.

j. For fall and spring use, and as auxiliary to help out other heating systems at important points during extremely cold weather.

k Where dust, gases, odor, noise, or access for attendants must be excluded.

l As booster heaters in central fan systems to permit local temperature control in individual rooms

2 ● On what basis should electrical heating cost be compared with other fuels?

a. Initial investment with interest and depreciation.

b. Operating cost for energy.

c. Saving in repairs and attendance

d Economy due to local accurate temperature regulation

e. Safety, convenience, and cleanliness.

f. Saving in space.

3 ● At what rate for electric power is electric heating feasible?

a. The answer is complicated because operating cost of electric heaters depends upon at least three factors, namely, demand charge, energy charge, and the sliding scale according to the amount of power used for all purposes.

b. For off-peak use which avoids the demand charge an energy rate of 1 cent per kilowatt-hour is often attractive for heating systems of moderate size. Larger jobs usually require a rate of about ¾ cents per kilowatt-hour.

c. For unrestricted heating service for auxiliary purposes an average rate of 2 cents per kilowatt-hour may be satisfactory for small installations.

d. Wherever other factors make electric heating especially desirable the rates may be even higher than those mentioned above.

e. For domestic hot water supply a rate of 1 cent per kilowatt-hour is usually satisfactory.

4 ● What advantages have electric fan unit systems over plain convection heaters?

a. Better distribution of heat to the floor level.

b. Elimination of condensation on machinery, windows, and other cool surfaces

c. Wider choice as to location of the heater, and reduced cost of wiring connections.

d. The fan can be operated for air circulation only, during periods when heat is not required

5 ● In a fan type electric heating system, what important features are required that are not needed for a steam system?

a. A heating coil supplied with steam at constant pressure will remain at approximately constant temperature regardless of the amount of air passing over it, but the condensation rate will change The temperature of an electric coil supplied with a constant amount of energy will rise if the quantity of air is decreased, and fall if the quantity of air is increased. This happens because the input of electrical energy is constant while the input of steam energy varies with the condensation rate.

b. Because the temperature of an electrical coil will rise upon decreased air flow, the Underwriters require the installation of an approved thermal safety trip switch located in the heating chamber to cut off the electrical heating circuit automatically in case the air flow should be interrupted. This switch should remain off until manually reset.

c. Electrical heating elements vary greatly in their capacity for storing heat units after the power is shut off. In a fan system it is very important to use elements having the lowest possible heat storage capacity to avoid overheating motors and improperly operating the thermal safety trip switches when air flow ceases due to normal shut-downs

d Because the input of an electric heater is constant for a given rating, it is necessary to provide a selective electrical control which will automatically compensate for variations in the volume of air flow This cannot be done by mixing dampers alone as in a steam system.

6 ● What problems must be considered in connection with electric water heating?

a. For the best available power rate consult the Electric Power Company supplying service.

b. The maximum daily and hourly demand for hot water

c. The size of storage tank necessary to carry over the peak load periods when no reheating is done.

d. The kilowatt rating required to reheat enough water during off-peak periods.

e. Standby radiation losses of the storage tank and hot water piping.

f. Cost of providing electrical supply lines of ample capacity with fuses, meters and switches.

g. Tank materials and design to avoid expensive replacements due to corrosion.

Chapter 41

RADIANT HEATING

Physical and Physiological Factors, Control of Heat Losses, Rate of Heat Production, British Equivalent Temperature, Application Methods, Calculation Principles, Mean Radiant Temperature, Measurement of Radiant Heating

HEATING for health and comfort is generally understood to mean that heat must be supplied in order to control the rate of heat loss from the human body so that the physiological reactions are conducive to a feeling of comfort. In convection heating, it is generally the function of the heating medium to transfer the heat to the air and thence to the occupant of the room, while the primary object of radiant heating is to warm the surrounding surfaces without appreciably heating the air. The difference between convection heating and radiant heating is therefore partly physical and partly physiological. Reference to low temperature radiation is actually not heating at all, except in a secondary sense. Low temperature radiation is produced not to heat the individual in the room, but to reduce the net rate at which the body surface loses heat by radiation.

Comfort requires that heat be removed from the body at the same rate as it is generated by the oxidation of food stuffs in the body tissue. Furthermore, the heat should be dissipated in a manner conducive to the physiological requirements of the human body. Actually the feeling of heat and cold in an individual is not so much a measure of the rate at which body heat losses take place as compared with the heat generated within the body, as it is an indication that the sensation of the body is more susceptible to the manner in which the heat is abstracted from the body. This principle is the basis upon which radiant heating is founded.

CONTROL OF HEAT LOSSES

Heat is transferred from any warm dry body to cooler surroundings principally by convection and by radiation, the approximate total being the sum of the two. Where the body surface is moist as with the human body, there is additional loss of heat through evaporation from both the body surface and the respiratory tract.

The rate of heat loss by convection depends upon the difference between the temperature of the body and of the surrounding air, and on the rate of air motion over the body.

The loss by radiation of a given surface depends entirely upon the difference between the temperature of the body and the mean surface temperature of the surrounding walls and objects. This latter temperature is called the *mean radiant temperature* (MRT)

Because these two types of heat losses act in a supplementary manner toward each other, a required rate of heat loss can be secured by having a relatively low air temperature and a relatively high MRT, or *vice versa.* Thus, if the air is reduced from a given temperature to a lower temperature, the amount of heat lost from the body by convection is increased, and this increase can be compensated for by raising the MRT. Similarly, with a higher air temperature the same total heat loss will be maintained by a correspondingly lower MRT.

Within limits the sensation of feeling cold can be avoided in two ways; first, by raising the air temperature surrounding the body, and secondly, by allowing the thermal radiation from warm objects to impinge on the body with sufficient intensity to make up for a lower air temperature.

It is the object of a heating installation to avoid the necessity for human body adaptation and also to provide comfort for those individuals doing the least physical work. While some conditions may take care of the heat loss from the body without controlling the generation of heat within, other conditions stimulate the production of heat within us, which enables the body to respond to the environment and generate more heat to meet the conditions

Rate of Heat Production

The normal rate of heat production in a sedentary individual is about 400 Btu[1] per hour, or (since the entire surface area of an average adult is 19.5 sq ft) about 20.5 Btu per square foot per hour. When considering radiant heating, it is necessary to calculate the radiation and the convection loss separately. The human body is of complicated shape, and radiation only takes place freely from the exposed outer surface. There are considerable portions of the body which radiate most of their heat to other portions, such as: the legs, arms, lower part of head etc. It is necessary to determine the equivalent surface of the body from which heat is radiated and a similar value for convection. The total surface for convection may be assumed as an approximate value of 19.5 sq ft and 15.5 sq ft for radiation.

The heat generated in the average human body is approximately 400 Btu per hour of which 75 per cent or 300 Btu per hour is the approximate value of the heat given off by radiation and convection. While it is difficult to differentiate the exact proportion of these two values, it is found that if the body gives off about 190 Btu per hour by radiation or 12.26 Btu per hour per square foot of radiant body surface, conditions of greatest comfort will result. This leaves 110 Btu per hour to be released by convection, or 5.64 Btu per hour per square foot of convected body surface.

[1]A.S.H V E RESEARCH REPORT No 830—Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C Houghten, W W. Teague, W. E. Miller, and W P. Yant (A S H.V E TRANSACTIONS, Vol. 35, 1929, p. 245)

The loss by evaporation, which depends on the air temperature, air movement and humidity, together with the loss by respiration makes up the balance of 100 Btu per hour. All of these values are relative because the total will vary materially with change of position, occupation, age, race, etc.

The mean normal surface temperature of the human body, taken over the whole area, including not only the exposed skin surface but also surfaces of the clothes and the hair, has been very extensively used as 75 F, particularly in England where radiant heating has been practiced for nearly 30 years. However, results obtained by Aldrich[2] in rooms in which the air and wall surface temperatures were approximately 72 F gave mean values nearer 83 F than 75 F. In both England and America mean wall and air temperatures of 72 F seem to be unwarranted; so it is not unreasonable to assume that a body surface temperature lower than 83 F may eventually be accepted. Some values have already been suggested as being the most suitable for the American climate, but the accepted standard for United States practice must be ultimately derived from research and practical experience.

The mean surface temperature of an inert body which will maintain the optimum heat loss by radiation and convection in a uniform environment of a given temperature may be calculated from fundamental equations for radiation and natural convection by substituting comparable cylinders for the body. While it may be possible to produce effects on a cylinder or any other body of a particular size and shape to estimate similar effects on the human body, it should be remembered that the heat loss from the body varies greatly with movement. Every movement of the body not only alters its shape but also the velocity of the air passing over it and the surface exposed to radiation. This fact makes it difficult to compare the effect of any environment on a cylinder to that of a human body. Heilman[3] gives the following equations:

$$H_r = 0.1723\, e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \tag{1}$$

$$H_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times \left(T_s - T_a\right)^{1.266} \tag{2}$$

where

H_r = heat loss by radiation, Btu per square foot per hour
H_c = heat loss by convection, Btu per square foot per hour.
T_s = absolute temperature of the body surface, degrees Fahrenheit.
T_w = absolute temperature of the walls, degrees Fahrenheit
T_a = absolute temperature of the air, degrees Fahrenheit.
$T_m = \dfrac{T_s + T_a}{2}$
D = diameter of cylinder, inches.
e = the ratio of actual emission to black body emission

If it is assumed that a normal adult has an average height of 5 ft 8 in.

[2]A Study of Body Radiation, by L. B. Aldrich (Smithsonian Miscellaneous Collections, Vol 81, No 6, December 1928)

[3]Surface Heat Transmission, by R. H. Heilman (*A.S.M.E. Transactions, Fuels and Steam Power Section*, Vol. 51, No 22, September-December, 1929)

and an average body surface of 19.5 sq ft and 15.5 sq ft for convection and radiation respectively, an equivalent effect can be considered on two cylinders 5 ft 8 in. high by 13.15 in. diameter and 10.45 in. diameter respectively.

BRITISH EQUIVALENT TEMPERATURE

The British Equivalent Temperature (BET) is the temperature of an environment which is effective in controlling the rate of sensible heat loss from a sizable black body in still air when the body has a maintained surface temperature equal to that of the human body. The BET is, therefore, a function of both the air temperature and the mean radiant temperature. Its numerical value in a uniform environment (walls and air at the same temperature) is equal to the temperature of the walls and the air. In a non-uniform environment (walls and air at different temperatures) the BET for America is at present considered to be equivalent to that of a uniform environment in which an 83 F surface loses sensible heat at the same rate as it does in the non-uniform environment. As originally defined, the BET was based on a body surface temperature of 75 F, but 83 F has been accepted as giving results more nearly conforming with American practice[4]. Temperatures selected depend on the clothes worn by the individual, which explains why ladies in evening dress desire a higher body surface temperature than a man dressed in evening suit leaving only hands and head uncovered.

For accurate calculations it would be more logical to assume a body surface temperature applicable to the room being occupied, but for general purposes it is considered sufficient to take an average of 83 F for all rooms. The higher the BET the less the heat loss from the body, as the rate of loss in still air is approximately proportional to the difference between the BET and the mean body surface temperature.

If the BET were 83 F, there could be no sensible heat loss from a surface at that temperature; so the temperature of a normal body surface would have to rise to a point where the heat generated in the tissues could be dissipated.

APPLICATION METHODS

There are several methods of applying radiant heating, as follows:

1. *By warming the interior surfaces of the building.* Pipe coils are embedded in the concrete or plaster of the walls or ceilings, the heating medium being hot water circulating through the pipe coils. These coils are generally constructed of small pipe spaced about 6 in. apart (Fig. 1). This has the effect of warming the entire concrete or plaster surface in which the pipes are embedded. Since the temperature of the heating medium should not exceed about 130 F due to the possibility of cracking the plaster, the area of the panel must be sufficient to supply the requisite quantity of heat at this low temperature. When carefully designed, this method produces comfortable and economical results, but offers some slight obstacles when alterations or additions to the building are desirable. Normally the hot water circulation is maintained by means of a circulating pump and facilities have to be provided to eliminate all air at the top of the system. All of the pipes are welded together and tested after erection to a hydraulic pressure of 500 lb per square inch

[4]A S H V.E. RESEARCH REPORT No. 962—Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperatures, by A. C Willard, A P. Kratz, and M K Fahnestock (A S H V E. TRANSACTIONS, Vol 39, 1933, p 303).

2. *By placing hot water or steam pipes under the floor.* With this arrangement the whole floor surface of a room is raised to a temperature sufficient to give comfortable conditions. This method is used extensively for schools and hospitals where large quantities of outside air are desirable (Fig. 2). In some cases special floors are con-

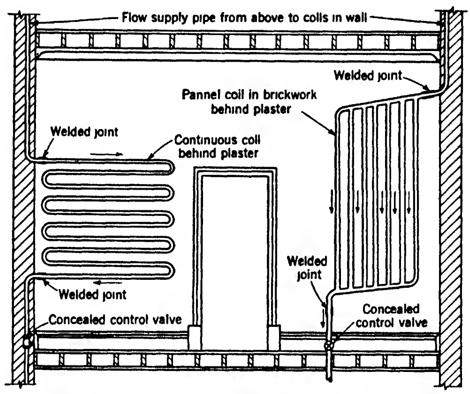

FIG. 1 PIPE COILS LOCATED IN INTERIOR WALL SURFACES

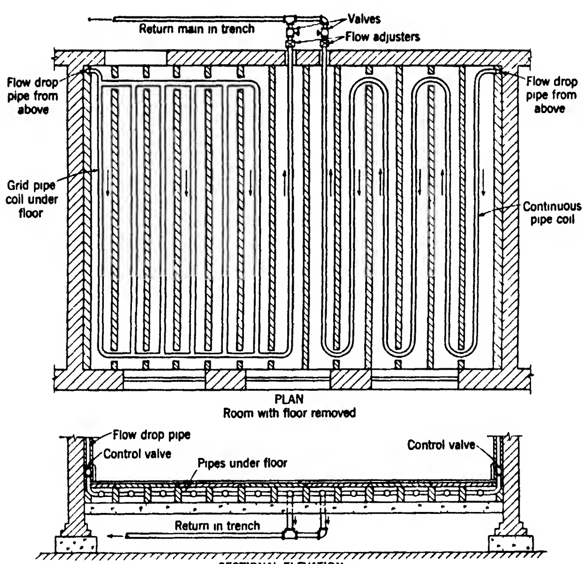

FIG. 2. ARRANGEMENT OF CONTINUOUS PIPE COIL IN FLOOR CONSTRUCTION

structed in sections so that a whole floor can be lifted to examine the pipes. The floor surface may be of concrete, wood blocks, marble or any other material unaffected by heat. Pipes under the floor may be larger than those embedded in the plaster walls and ceilings.

3. *By circulating warm air through shallow ducts under the floor.* In this design the entire floor surface of a room is heated as in method 2. This method while being more

expensive in construction, is effective and quite suitable for cathedrals and large public buildings (Fig. 3). To provide a uniform floor temperature, special consideration should be given to the design of the air ducts so that equal distribution is obtained.

4. *By attaching separate heated metal plates or panels to the interior surfaces.* These plates or panels are placed either in an insulated recess so that the surface of the panel is flush with the surface of the walls or ceilings, or they may be secured to the face of the wall. They may be covered with wood veneers and decorated to harmonize with other parts of the room, or they can be cast into panels to imitate oak or other wood designs. With flat plate panels it is a common practice to use a frame of plaster, wood, metal or composition around the panels to allow for expansion. These plates may be heated with either hot water or steam and connected similarly to an ordinary radiator system.

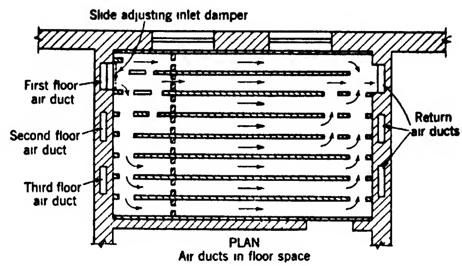

FIG. 3 DIAGRAM OF AIR DUCTS FOR FLOOR HEATING

5. *By electric heated metal plates or panels.* These plates or panels are either placed in insulated recesses of walls or ceilings or fastened to the face as desirable They should not have a surface temperature above 160 to 200 F. Some electric panels have a much higher surface temperature but a lower temperature gives a more comfortable condition and is more efficient.

6. *By electrically heated tapestry mounted on screens and on the wall* For this purpose the screen is woven with an electric continuous conductor being the warp and wool or silk being the weft. Such screens are useful to plug in at any position for emergency local heating without taking care of a large room or office

Note. If all of the heating panel is installed at one end of a large room there may be a marked difference between the BET on the two sides of the body It is usually desirable therefore that the heat be distributed at different parts in the room so that no uncomfortable effects will be felt from unequal heating.

CALCULATION PRINCIPLES

The calculations for radiant heating are entirely different from those for convective heating. The purpose of the latter is to determine the rate of heat loss from the room by conduction, convection, and radiation when maintained in the desired condition; radiant heating involves the regulation of the rate of heat loss per square foot from the human body.

The first step in the calculations for radiant heating is to ascertain the necessary mean radiant temperature (MRT); next, the size, temperature, and disposition of the heating surfaces required in the room to produce this MRT are estimated; and after this the determination of the convective heat is made.

Mean Radiant Temperature

If the whole of the interior surface of a room were at the same temperature, this temperature would represent the MRT. Such a condition

TABLE 1. TOTAL BLACK BODY RADIATION TO SURROUNDINGS AT ABSOLUTE ZERO[a]

Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor *e*				Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor *e*			
	e 1.00	*e* 0.95	*e* 0.90	*e* 0.80		*e* 1.00	*e* 0.95	*e* 0.90	*e* 0.80
30	99.3	94.3	89.4	79.4	71	136.5	129.6	122.9	109.3
35	103.5	98.3	93.2	82.8	72	137.4	130.5	123.6	109.9
40	107.6	102.4	96.8	86.1	73	138.4	131.5	124.5	110.6
45	112.1	106.5	100.9	89.7	74	139.6	132.6	125.6	111.7
46	112.9	107.3	101.6	90.4	75	141.0	133.9	126.9	112.8
47	113.9	108.2	102.5	91.1	80	146.6	139.4	132.0	117.4
48	114.8	109.1	103.4	91.9	85	152.3	144.6	137.1	121.9
49	115.6	109.9	104.1	92.4	90	157.9	149.9	142.1	126.4
50	116.5	110.6	104.9	93.2	100	169.6	161.1	152.6	135.7
51	117.5	111.6	105.8	94.0	110	181.6	172.5	163.5	145.4
52	118.4	112.5	106.5	94.7	120	194.8	185.0	175.4	155.9
53	119.4	113.4	107.4	95.5	130	210.1	199.6	189.1	168.1
54	120.2	114.2	108.2	96.2	140	223.2	212.1	201.0	178.5
55	121.1	115.1	109.0	96.9	150	237.1	225.2	213.5	189.7
56	122.1	116.0	109.9	97.7	160	251.1	238.8	226.0	201.0
57	123.1	117.0	110.9	98.5	170	270.5	257.0	243.5	216.4
58	124.0	117.8	111.6	99.2	180	288.0	273.8	259.1	230.4
59	124.9	118.6	112.4	99.9	190	306.5	291.0	275.8	245.1
60	125.8	119.5	113.4	100.7	200	325.2	309.0	292.8	260.3
61	126.6	120.3	114.0	101.4	210	348.0	330.6	313.1	278.4
62	127.7	121.4	114.9	102.2	220	371.5	353.0	334.4	297.1
63	128.6	122.2	115.8	102.9	250	437.8	415.9	394.0	350.2
64	129.6	123.1	116.7	103.7	300	575.0	546.1	517.5	460.0
65	130.5	124.0	117.5	104.4	350	740.0	703.0	666.0	592.0
66	131.6	125.0	118.4	105.4	400	942.1	895.0	847.5	753.5
67	132.5	125.9	119.3	106.0	450	1176.0	1117.0	1059.0	941.0
68	133.5	126.8	120.1	106.8	500	1464.0	1390.0	1318.0	1171.0
69	134.5	127.8	121.1	107.6	550	1791.0	1701.0	1613.0	1434.0
70	135.5	128.8	121.9	108.4	600	2405.0	2284.0	2165.0	1925.0

[a]These factors are calculated from the formula

$$Q = e\left(\frac{0.1723 \times T^4}{100{,}000{,}000}\right)$$

where

Q = total black body radiation, Btu per square foot per hour.
e = emissivity
T = absolute temperature, degrees Fahrenheit.

seldom exists, however, since the actual surface temperature in any heated space having surfaces exposed to the outer air varies greatly for different sides of the enclosure. It is therefore necessary to ascertain by calculation the mean of these interior surface temperatures.

The mean temperature in this sense is not the arithmetic average of the actual thermometric temperatures of the surfaces, but the temperature corresponding to the average rate of heat emission per square foot of surface. The temperature corresponding to this mean emission can be taken from Table 1. Conversely, the emission at different temperatures and also the emissivity factors can be obtained from this table. For instance, 1 sq ft of surface at 50 F will emit 104.9 Btu per square foot per hour to surroundings at absolute zero if the emissivity of the surface is 0.9.

If the area in square feet of each part of the space is multiplied by the emission value corresponding to its actual temperature, and these products are added together, the gross amount of radiant heat discharged into the room by the wall surface per hour is obtained. This quantity, divided by the total interior surface, gives the average amount of heat coming into the room from the surface of the walls per square foot of surface per hour.

Interpolating in Table 1, the total radiation from a surface at 83 F for an emissivity of 0.95 is 142 Btu per square foot per hour. The difference between 142 Btu and the average amount of heat coming into the room is the amount which will be lost per square foot per hour by radiation from a body at 83 F. If a rate at which it is desired that heat be lost from the body by radiation and convection be assumed, the mean radiant emission from the walls required to give the desired result can be determined from Table 1, as can also the required air temperature for the corresponding convective effect.

The determination of the amount of radiant heating surface needed in a room requires knowledge of the climate, the type of structure, the type of heating, and the surface temperature of the walls. This problem can be solved only on an empirical basis. After some experience, however, it is possible to estimate these variables with a considerable degree of accuracy for any climate or construction.

Assume that a mean radiant temperature of 65 F is desired. Table 1 shows that with all the walls at this temperature, and with an emissivity of 0.95, the gross heat emission is 124 Btu per square foot per hour. The total emission of radiation into the room from that surface would therefore be $A \times 124$, where A is the total inside area of the room. This is the *desired* emission.

If the whole area be divided into a number of different parts which are each at a uniform temperature—a_1, a_2, a_3,—and each is multiplied by the value of the heat emission corresponding to that temperature, and if all these products are added together, their sum will represent the total *actual* emission of radiation into the room at these temperatures without the aid of any hot surface.

The difference between the desired emission and the actual emission represents the additional heat which must be supplied by the hot surface. The temperature of the proposed hot surface must then be selected, and its emission per square foot at that temperature determined from Table 1. This emission is divided into the additional amount of heat needed, adjusted for the fact that the heating units will shield the walls behind them, and the quotient obtained will be the area of the required heating surface.

It is evident that this method of calculation is approximate, and depends for its accuracy on a correct estimate of the ultimate surface temperatures attained by the actual wall surfaces.

It is necessary also to calculate how much heat will be given off by the same surfaces by convection, and thereby to determine whether this amount of convected heat will warm entering ventilating air to the temperature maintained. If it will not, additional convection surfaces must be introduced to make up the deficiency.

TABLE 2. SURFACE AREAS, TEMPERATURES AND EMISSIONS FOR A ROOM OF 5760 CU FT

	AREA SQ FT	ASSUMED SURFACE TEMPERATURE (DEG FAHR)	HEAT EMISSION (BTU PER SQ FT PER HOUR)	TOTAL HEAT EMISSION FROM AREA (BTU PER HOUR)
External Wall	297	50	110.6	32,850
Glass	279	45	106.5	29,710
Inner Wall	480	55	115.1	55,250
Ceiling	480	55	115.1	55,250
Floor	480	55	115.1	55,250
Total	2016			228,310

Example 1. The surface areas, temperatures, and emissions for a room having a volume of 5760 cu ft are given in Table 2. The figures for temperatures are fairly representative of American practice with well-built walls, and are based on an emissivity of 0.95 which approximates that of most paints and building materials.

The mean radiant temperature of the room is 228,310/2016 = 113.2 Btu per square foot per hour which, as seen from Table 1, corresponds to an MRT of 53 F for an average emissivity of 0.95.

For an average individual having a body surface area of 15 5 sq ft under conditions of comfort with a body surface temperature of 83 F, the heat given off by radiation when calculated by means of Equation 1 is 217 Btu per hour, or 14 Btu per square foot per hour. This corresponds to an environmental emission of 142 − 14 = 128 Btu per square foot per hour, and, according to Table 1, to an MRT of 69.2 F.

If this body be placed in the room described, it will lose heat at the rate of 15.5 × (142 − 113 2) = 446 Btu per hour. This loss is 229 Btu per hour more than the 217 Btu per hour calculated, or 14.77 Btu per square foot per hour, more than the rate of heat loss for comfort.

In order to determine the amount of radiating surface necessary to maintain the MRT at 69.2 F, assume the surface temperature of the hot plates to be installed to be 160 F, which is approximately the temperature they would have if heated by hot water.

The 2016 sq ft total area of the surfaces of the room multiplied by 128 which is the emission in Btu per square foot per hour necessary to maintain a body surface temperature of 83 F, gives a total desired emission of 258,048 Btu per hour. It is necessary to supply enough radiant heating surface to increase the total actual mean radiant heat emission by the room from 228,310 as shown in Table 2, to the 258,048 Btu desired. The additional heat needed is the difference between these figures, or 29,738 Btu. Since, from Table 1, the emission per square foot at 160 F is 238 8 Btu, the required radiant heating surface needed is 29,738/238.8 = 124 sq ft. The effect of this surface suitably placed would be to raise immediately the mean radiant temperature to the required degree and to maintain it at that value as long as the surfaces remained at the values assumed.

The calculation may be simplified by preparing tables showing, at the usual temperatures, the area of hot surface required to bring each square foot of actual wall surface at various temperatures up to a general standard of from 60 F to 70 F. It would then be necessary only to multiply the respective areas by the appropriate factors, and to add the results, to obtain the required total.

MEASUREMENT OF RADIANT HEATING

Convection heating, having as its object the raising of the air temperature to a specified degree, must be measured by thermometric methods which indicate essentially the air temperature, and not the rate of heat loss from the human body. Radiant heating, having as its object the control of the rate of heat loss from the human body, can be measured

only by methods which basically are calorimetric, that is, which measure directly the rate of heat loss from an object maintained at the temperature of the body, irrespective of air temperature.

The apparatus for this purpose consists essentially of a hollow sphere, or cylinder, containing a fluid which can be maintained accurately at the accepted mean surface temperature of the human body, with an accurate means of measuring the rate of heat supply required to maintain the temperature at that exact point. The latter measurement can be made with sufficient accuracy by electrical methods. Although a definite BET is desirable, the mean radiant and air temperatures may both vary, provided the heat loss by radiation and convection from a surface at 83 F is maintained at the correct proportion or within reasonable limits.

This instrument, the *eupatheoscope*, can readily be adapted as a thermostat by electrical control to shut off or turn on heat when the critical temperature of 83 F or any other predetermined temperature on the surface of the vessel is increased or decreased. A modification of the instrument is called the *eupatheostat*.

Another instrument for maintaining comfort conditions is at present available only in a model adapted to British practice as it is designed for a temperature of 75 F. It consists of a blackened copper sphere of approximately 6 in. diameter in which is housed a cylindrical sump containing a volatile liquid. In operation, a small electric heating coil drawing about 5 watts creates in the sphere a vapor pressure which is constant as long as the heat losses from the sphere are standard. If the temperature of the air or the MRT becomes too high for comfort, a greater pressure is created, owing to a smaller loss of heat from the sphere. This increase of pressure acts on a diaphragm and shuts off the supply of heat to the room.

For testing work, the *globe thermometer* is a very useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 to 9 in. in diameter, usually made of thin copper and painted black. The temperature thus recorded is termed the *radiation-convection temperature*.

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PROBLEMS IN PRACTICE

1 ● Where did radiant heating derive its name?

The term radiant heating was introduced about 28 years ago to designate flat heating surfaces made to give off practically all their heat by radiant ether waves instead of relying on convected warm air.

2 ● What is actually meant by radiant heating and what are its underlying principles?

The term radiant heating now applies to methods of heating where, instead of heating the air in a room to a predetermined temperature, flat heating surfaces are placed in a room so that the average effective temperature of walls, ceiling, glass and floor surfaces exposed to the body is just sufficient to prevent the body losing too much heat by radiation It takes into consideration that the body generates more heat than it requires, so that it does not require any heat from without. The surplus heat, however, must be given off according to the physiological requirement of the body.

3 ● What kind of heating surfaces are in general use?

The heating units may have flat iron surfaces heated with steam or hot water and placed in side walls or under windows, or they may be supported on the ceiling and suitably decorated and connected as ordinary steam or hot water radiators. Hot water pipes may be embedded in the floor, walls or ceiling, and when in the floors they may be covered with concrete and wood blocks or other suitable material, the finish of the surface being more important than the composition of the material. When in the ceiling or walls, they can be covered with plaster to harmonize with the rest of the room Electrical radiant heaters are made by embedding resistance elements in porcelain, or electric conductors may be woven into thick paper and fastened to the walls and ceilings, electric wires may be woven with tapestry to form portable screens for local heating.

4 ● What surface temperatures are generally used?

Where hot water pipes are embedded in plaster, the surface temperature varies from 90 to 130 F. Where flat iron plates are used these may vary from 140 to 220 F. With electric resistances embedded in porcelain the surface temperature may vary from 200 to 500 F. High surface temperatures are not recommended.

5 ● What kind of heat rays are commonly generated for radiant heating?

All heat rays are generally assumed to be the same as light rays, they travel at the speed of light, but they are invisible and longer The rays used in heating are 0 00005 to 0 0001 in. long, compared with invisible red rays of about 0 000027 in.

6 ● When and why does the human body feel cold?

The body feels cold not only when it loses heat at a greater rate than it can generate it but also when heat is abstracted from the body disproportionately. Since the human body generates more heat than is necessary, it is only necessary to provide conditions that will regulate the correct ratio of losses, the provision of suitable radiant heating surfaces is one way to establish these conditions

7 ● Why is the heat loss from the body by radiation important?

The heat loss by radiation is proportional to the fourth power of the temperature difference between the surface of the body and the average surface temperature of the surrounding walls, windows, etc., whereas for convection losses, it is only proportional to the 1 25 power.

8 ● What is the approximate relation for heat losses?

Heat losses from the body when in a sedentary position are approximately as follows: radiation 49 per cent, convection 23 per cent, evaporation 15 per cent, respiration 11 per cent, and miscellaneous 2 per cent. Actually, it depends upon age, environment and other conditions.

9 ● What generally is the air temperature necessary to give equal comfort effect for sedentary conditions?

With radiant heating, 64 to 66 F. With convection heating, 70 to 72 F.

10 ● Why is there a saving in fuel consumption with radiant heating?

A saving is effected because the differential between inside and outside temperature is much less for radiant heating. Less ventilating air is necessary and this can be supplied at a much lower temperature.

11 ● Describe how to calculate the required amount of radiant heating surface.

a. Obtain the mean heat emission in Btu per square foot per hour for room surfaces X, using values in Table 1, and surface temperatures as shown in second column of Table 2.

b. Deduct X from 142 (142 being the emission per square foot given off by the human body at 83 F surface temperature) $= Y$ in Btu per square foot per hour.

c. From (142-X) deduct 11.1 (11.1 being the average radiation which the human body should lose per square foot for comfort conditions) $= (142\text{-}X\text{-}11.1) = Z$.

d. Multiply total interior surface of room by Z and divide by the emission per square foot from radiant heater, giving the surface S of radiant heater in square feet.

12 ● Give a simple formula to calculate radiant heating surface required, and explain.

$$S = \frac{(142 - X - 11.1)\,A}{B}$$

where

S = surface of radiant heater, square feet.
142 = Heat emission, Btu per square foot per hour which the human body would give off at 83 F, with surroundings at absolute zero.
X = mean heat emission, Btu per square foot per hour from surfaces of room.
11.1 = heat emission, Btu per square foot per hour from human body.
A = total surface, square feet of walls, ceilings, windows, etc., in room
B = heat emission per square foot from radiant heater surface.

13 ● What natural evidence have we that air temperature alone is no criterion of comfort and that radiant heat affects the body more quickly?

When standing in the sunshine on a cool spring day, a person feels perfectly comfortable, but when a cloud passes over the sun, he instantly feels much cooler as the shadow reaches him. A shielded thermometer recording the temperature of the air shows no reduction in air temperature in so short a period, so that the person actually feels a sensation of cold which an ordinary thermometer cannot register. This shows that light and heat rays are shut off simultaneously and travel at the same speed, it also proves that radiant rays affect the comfort of the body quicker than air temperature does.

Chapter 42

DISTRICT HEATING

Steam Distribution Piping, Selection of Pipe Sizes, Provision for Expansion, Capacity of Returns with Various Grades, Conduits for Piping, Pipe Tunnels, Inside Piping, Steam Requirements, Fluid Meters and Metering, Rates, Utilization, Automatic Temperature Control.

THOSE phases of district heating which frequently fall within the province of the heating engineer are outlined here with data and information for solving incidental problems in connection with institutions and factories. Some data are included to cover the piping peculiar to heating systems which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

STEAM DISTRIBUTION PIPING

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter the local conditions so fully control the layout that little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per hour per square foot of equivalent heating surface is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Miscellaneous steam requirements such as laundry, cooking, or process should be individually calculated.

The steam requirements for water heating should be taken into account,

but in most types of buildings this load will be relatively small compared with the heating load and will seldom occur at the time of the heating peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend upon (1) boiler pressure, (2) whether exhaust or live steam, (3) pressure requirements of apparatus to be served. If steam has been passed through electrical generating units, the pressure will be considerably lower than if live steam, direct from the boilers, is used.

The advantages of low pressure distribution (2 to 30 lb per square inch) are (1) smaller heat loss from the pipes, (2) less trouble with traps and valves, (3) simpler problems in pressure reduction at the buildings, and (4) general reduction in maintenance costs. With distribution pressures not exceeding 40 lb per square inch there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than this a second reducing value or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators.

The advantages of high pressure distribution are (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than building heating.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 lb per square inch, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 lb per square inch is usually demanded although 30 lb per square inch is sufficient if the mangle is equipped with a large number of rolls and if a slow rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 lb per square inch.

PIPE SIZES

The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of the Unwin pressure drop formula. This is one of several formulas which may be used. Unwin's formula, which gives pressure drops slightly larger than actual test results, is as follows:

$$P = \frac{0.0001306\, W^2 L \left(1 + \frac{3.6}{D}\right)}{dD^5} \qquad (1)$$

where

P = pressure drop, pounds per square inch
W = weight of steam flowing, pounds per minute
L = length of pipe, feet.
D = inside diameter of pipe, inches.
d = average density of steam, pounds per cubic foot.

This formula is similar to the Babcock formula given in Chapter 16.

Information on provision for expansion will be found in Chapter 18.

Where steam and return piping are installed in the same conduit, the return piping usually follows the same grade as the steam piping. In general, the condensation is pumped back under pressure. Where the condensation returns by gravity, Table 1 gives the sizes of the return piping. It is evident that at points where the grade is great, smaller pipes can be installed.

CONDUITS FOR PIPING

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural or cast steel set in concrete. In general, cast steel is preferable to structural steel.

TABLE 1. CAPACITY OF RETURNS FOR UNDERGROUND DISTRIBUTION SYSTEMS IN POUNDS OF CONDENSATE PER HOUR

Size[a] of Pipe In.	Pitch of Pipe per 100 Ft						
	6″	1′	2′	3′	5′	10′	20′
1	448	998	1890	2240	3490	5490	7490
1¼	1740	2490	3990	4880	6480	9480	13500
1½	2700	4190	5740	7480	9480	14500	20900
2	4980	7380	10700	13900	16900	24900	36900
3	13900	22500	30900	37400	50400	74800	105000
4	30900	44800	64800	79700	105000	154000	229000
5	54800	79800	120000	144800	195000	294000	418000
6	90000	138000	187000	237000	312000	449000	....
8	190000	277000	404000	508000	660000	938000	....
10	344000	498000	724000	900000	1190000	....	....
12	555000	798000	1148000	1499000	1990000	....	....

[a]Size of pipe should be increased if it carries any steam

In laying out underground conduits the following points should be borne in mind:

1. The depth of the buried conduit should be kept at a minimum. Excavation costs are a large factor in the total cost.

2 An expansion joint, offset, or bend should be placed between each two anchors.

3. If the distance between buildings is 150 ft or less and the steam line contains high-pressure steam, the line may be anchored in the basement of one building and allowed to expand into the basement of the second building. If the steam line contains low-pressure steam (up to 4-lb pressure), this method may be used if buildings are 250 ft or less apart.

4. If the distance between buildings is between 150 ft and 300 ft and the steam line contains high-pressure steam, the lines should be anchored midway between the buildings and allowed to expand into the basements of both buildings. If the steam line contains low-pressure steam this method may be used if buildings are between 250 ft and 600 ft apart. No manhole is required at the anchor, and a blind pit is all that is necessary.

5. For longer lines, manholes must be located according to experience, physical conditions and the expansion value of the type of expansion joint or bend that is used. The

minimum number of manholes will be required when an expansion bend, or an anchor with double expansion joint, is placed in each manhole and the pipes are anchored midway between manholes.

6. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied The hydrostatic test pressure should be one-and one-half times the maximum allowable pressure and it should be held for a period of at least two hours without evidence of leakage. In any case the pressure should be no less than 100 lb per square inch.

There are many types of conduits, some of which are manufactured products and some of which are built in the field. The styles and construction of conduits commonly used may be classified as follows. Some of the more common forms are illustrated in Fig. 1.

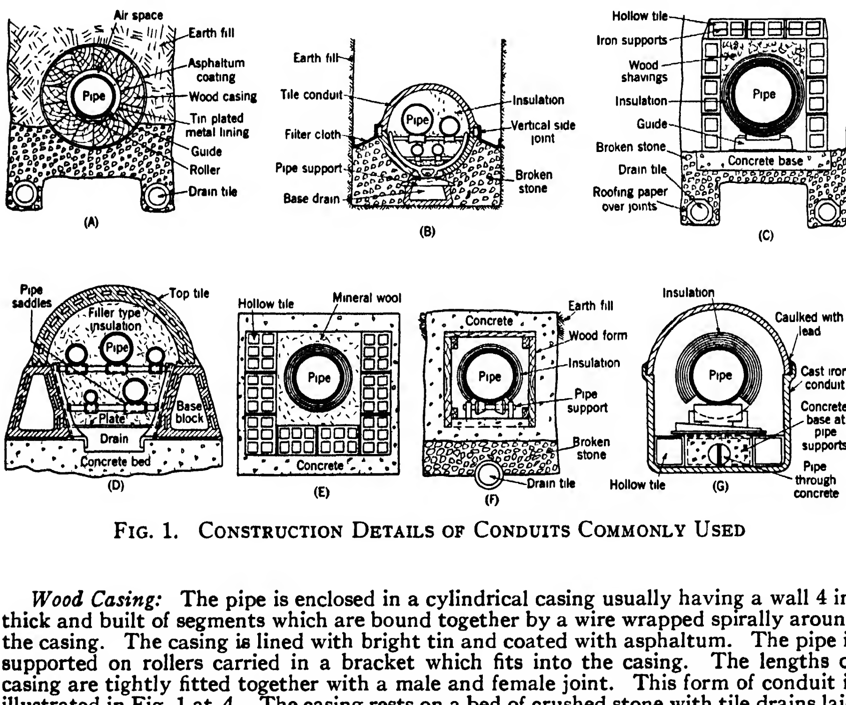


FIG. 1. CONSTRUCTION DETAILS OF CONDUITS COMMONLY USED

Wood Casing: The pipe is enclosed in a cylindrical casing usually having a wall 4 in thick and built of segments which are bound together by a wire wrapped spirally around the casing. The casing is lined with bright tin and coated with asphaltum. The pipe is supported on rollers carried in a bracket which fits into the casing. The lengths of casing are tightly fitted together with a male and female joint. This form of conduit is illustrated in Fig. 1 at *A*. The casing rests on a bed of crushed stone with tile drains laid below. The tile drains are of 4-in. field tile or vitrified sewer tile, laid with open joints.

Filler Type: The pipes are supported on expansion rollers properly supported from the conduit or independent masonry base. The pipes are protected by a split-tile conduit, and the entire space between the pipes and the tile is filled with an insulating filler. Thus the pipes are nested and the insulation between them and the tile effectively prevents circulation of air. The conduit is placed on a bed of gravel or crushed rock from 4 to 6 in. thick, which is extended upward so as to come about 2 in. above the parting lines of the tile. A tile underdrain is placed beneath the conduit throughout the entire length and is connected to sewers or to some other point of free discharge. At *B* and *D* in Fig. 1 are shown two forms of tile conduit of the filler type.

Circular Tile or Cast-Iron Conduit: The pipes are carried on expansion rollers supported on a frame which rests entirely on the side shoulders of the base drain foundation. The pipes are protected by a sectional tile conduit, scored for splitting, or a cast-iron conduit, both being of the bell and spigot type. The conduit has a longitudinal side joint

for cementing, after the upper half of conduit is in place, so shaped that the cement is keyed in place while locking the top and bottom half of the conduit together with a water-tight vertical side joint. The cast-iron conduit has special side locking clamps in addition to the vertical side joint. The entire space between the conduit and the pipes is filled with a water-proofed asbestos insulation. The conduit is supported on the base drain foundation, each section resting on two sections of the base drain, thus interlocking. The base drain is so shaped that it provides a cradle for the conduit, resting solidly on the trench bottom and providing adequate drainage area immediately under the conduit The underdrain is connected to sewers or some other point of free discharge. For tile conduit the base drain is vitrified salt glazed tile and for cast-iron conduit it is either extra heavy tile or cast-iron. A free internal drainage area is also provided to carry away any water that may collect on the inside of the conduit from a leaky pipe or joint in the conduit. Broken stone is filled in around the base drain and up to the vertical side joint. The broken stone is covered with an asphalted filter cloth to prevent sand from sifting through the broken stone and clogging the drainage.area of the base drain. The tile conduit is made in 2-ft lengths and the cast-iron conduit in 4-ft lengths, cast in separate top and bottom halves. Special reinforcing ribs give the cast-iron conduit ample strength with minimum weight.

Insulated Tile Type: The insulating material, diatomaceous earth, is molded to the inside of the sectional tile conduit. The space between the pipes and the insulating conduit lining may also be filled with insulation. The pipes are carried on expansion rollers supported on a frame which rests on the side shoulders of the base drain foundation. This type of conduit has the same mechanical features as those described under the heading Circular Tile or Cast-Iron Conduit

Sectional Insulation Type (Tile or Cast-Iron): Each pipe is insulated in the usual way with any desired type of sectional pipe insulation over which is placed a standard waterproof jacket with cemented joints. The pipes are enclosed in a sectional tile or cast-iron conduit as described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Concrete Trench): A type of construction frequently used in city streets, where service connections are required at frequent intervals, the pipes are insulated as described in the preceding paragraph, and are enclosed in a box or trench made either entirely of concrete, or with concrete bottom and specially constructed tile sides and tops. The pipes are supported on roller frames secured in the concrete. At *C* and *E*, Fig. 1, are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The use of loose material in addition to the sectional insulation is, of course, optional and is only justifiable where high pressure steam is used. The conduit shown at *F* is of a similar type and has the advantage of being made entirely of concrete and other common materials.

Sectional Insulation Type (Bituminized Fibre Conduit): Each pipe is individually insulated and encased in a bituminized fibre conduit. The insulating material is 85 per cent carbonate of magnesia sectional pipe covering, applied in the usual manner as on overhead pipes, except that bands are omitted. After every fifth section of magnesia covering there is applied a short, hollow section of very hard asbestos material in the bottom portion of which rests a grooved-iron plate carrying ball-bearings upon which the pipe rides when expanding or contracting. This short expansion section is of the same outside diameter as the adjacent 85 per cent magnesia covering. Over the pipe covering and expansion device there are placed two layers of bituminized fibre conduit with all joints staggered, and the surface of each conduit is finished with liquid cement. Conduits are placed on a bed of crushed rock or gravel, approximately 6 in. deep, and this is extended upward to about the center line of the conduit when trench is backfilled. Underdrains leading to points of free discharge are placed in the gravel or crushed rock beds.

Special Water-Tight Designs: It is occasionally necessary to install pipes in a very wet ground, which calls for special construction. The ordinary tile or concrete conduit is not absolutely water tight even when laid with the utmost care. The conduit shown at *G*, Fig. 1, is of cast-iron with lead-calked joints and is water-tight if properly laid. It is obviously expensive and is justified only in exceptional cases. A reasonably satisfactory construction in wet ground is the concrete or tile conduit with a water-proof jacket enclosing the pipe and its insulation, and with the interior of the conduit carefully drained to a manhole or sump having an automatic pump. It is useless to install external drain tile when the conduit is actually submerged.

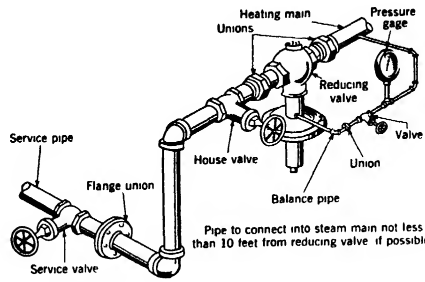


FIG. 2. CONNECTIONS FOR REDUCING VALVES OF SIZE LESS THAN 4 IN.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

On account of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes not installed where provision for the heating lines is the only consideration, but only where they are required to accommodate miscellaneous other services or provide underground passage between buildings.

OVERHEAD DISTRIBUTION

In some industrial and institutional applications, the distribution piping may be installed, entirely or in part, above ground. This method of construction has the advantage of requiring no excavation and being easily maintained.

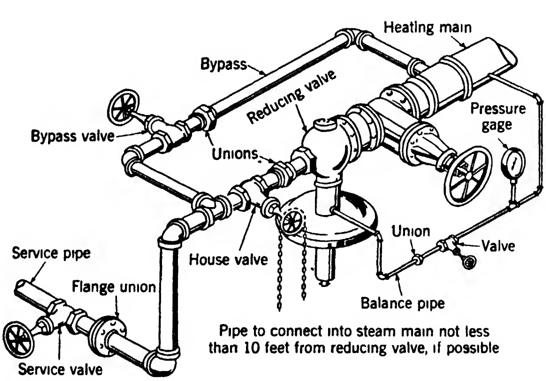


FIG. 3. CONNECTIONS FOR REDUCING VALVES OF SIZE 4 IN. AND LARGER, AND FOR EXPANDED VALVES

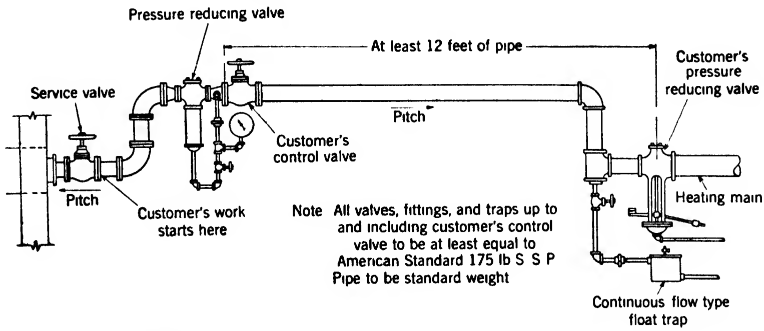


FIG 4. STEAM SUPPLY CONNECTION WHEN USING TWO REDUCING VALVES

INSIDE PIPING

Figs. 2 and 3 show typical service connections used for low pressure steam service. As shown in Fig. 2, no by-pass is used around the reducing valve on sizes less than 4 in. Fig. 3 illustrates the use of a by-pass around reducing valves 4 in. and larger. This latter construction permits the operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam until repairs are made.

Fig. 4 shows a typical installation used for high pressure steam service. The first reducing valve, effects the initial pressure reduction. The second reducing valve reduces the steam pressure to that required.

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests but principally to insure satisfactory and economical service to the consumer. There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

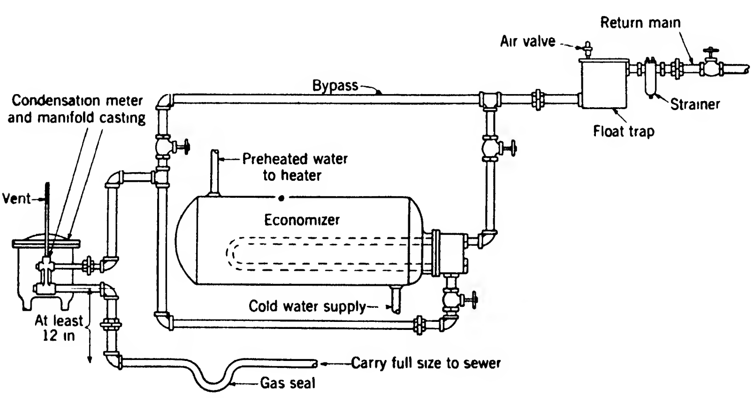


FIG 5. RETURN PIPING FOR CONDENSATION METER

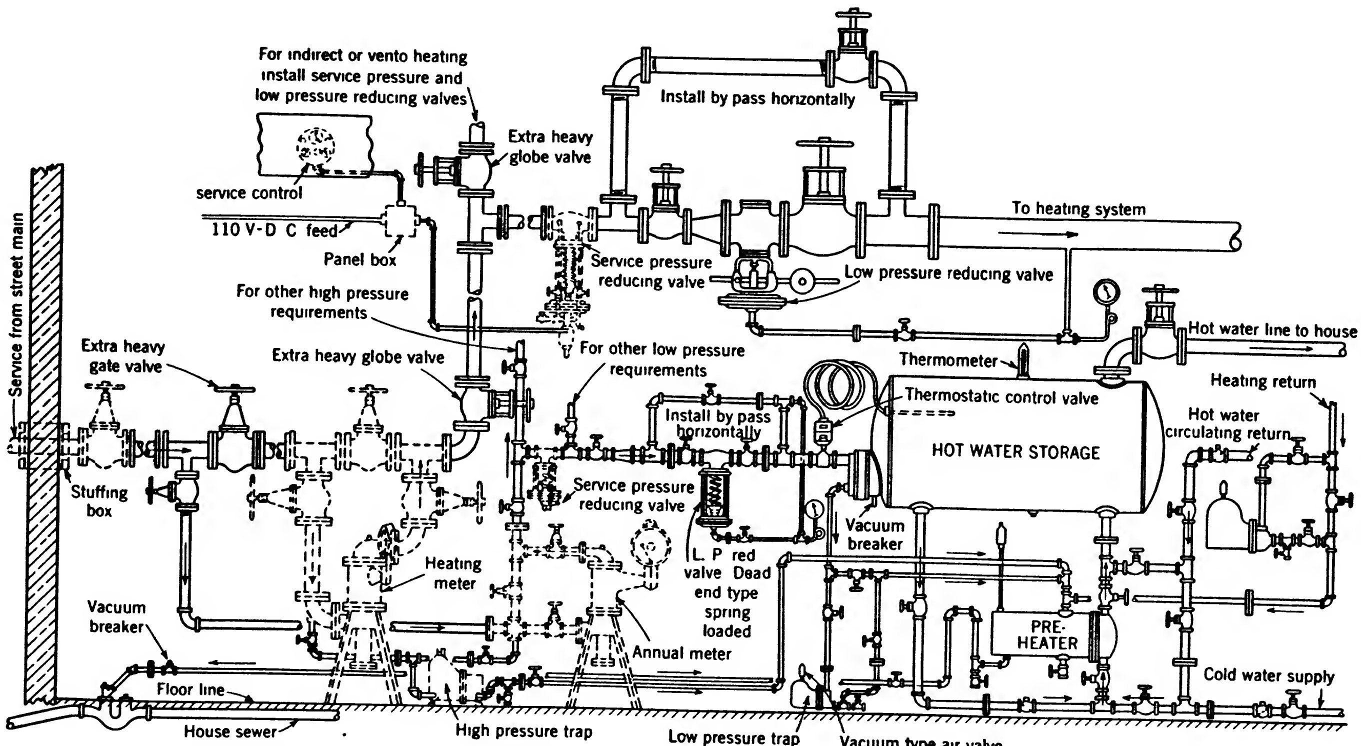


FIG. 6. TYPICAL SERVICE INSTALLATION ALSO SHOWING WATER HEATER AND ECONOMIZER CONNECTIONS

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very cold weather without endangering plumbing It is necessary, however, to have an ample amount of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones each with a separately controlled heat supply is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind

2 Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling coil, or as is more frequently done, by a water heating economizer (see Fig 5) which preheats the hot water supply to the building. Fig. 6 shows a typical steam service installation for high pressure steam, complete for steam flow metering, water heating, preheating, automatic heating control, and for using steam for other purposes.

The condensation from the heating system, after leaving the trap, passes through the economizer. The supply to the hot water heater passes through the economizer, absorbing heat from the condensate. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the economizer and the water heater proper, not at the economizer inlet, because the recirculated hot water is itself at a high temperature. The number of square feet of heating surface in the economizer should be approximately equal to one per cent of the equivalent square feet of heating surface in the building.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensate if storage capacity is provided for the preheated water. Frequently a type of economizer is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature.

This may be done in several ways, as by the use of temperature controls of various types or by orifice systems Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds pressure, thus producing some control over the heat output of the heating system by varying the temperature of the steam in the radiators. Several proprietary systems are on the market which accomplish this automatically, either with outdoor or indoor controls or a combination of both One form of control which appears to be well suited for controlling district steam service to a building is the weather compensating control. It regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply, and at the same time insures delivery of steam to all the radiators. This type of control can be equipped with time clocks and thermostats to provide a warming-up period in the morning.

Another form of regulation, known as the time-limit control, is sometimes employed for regulating the steam supply from the central station main to the building. Such a control provides an intermittent supply of steam to the radiators either throughout the 24 hours of the day or during the daytime hours only. The setting of a switch may provide no service, continuous service, or periodic service For the latter, by means of several intermittent settings, steam will be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. These settings afford from 15 to 80 per cent of the maximum heating effect required on days of zero temperature. A night switch with a variety of settings may be adjusted so as to maintain throughout the night the intermittent supply called for by the day switch setting, or may be set to interrupt the opera-

tion of the day switch and entirely cut off the supply of steam to the radiation at night during certain hours which are selected by the operating engineer.

The maximum in economical operation and satisfactory heating can only be obtained by the use of some automatic temperature control system.

FLUID METERS

The perfection of fluid meters has contributed more to the advancement of district heating than any other one thing. These meters may be classified as follows:

1. *Positive Meters:* The fluid passes in successive isolated quantities—either weights or volumes. These quantities are separated from the stream and isolated by alternately filling and emptying containers of known capacity.

2. *Differential Meters:* In the differential meter, the quantity of flow is not determined by simple counting, as with the positive meter, but is determined from the action of the steam on the primary element.

Additional sub-divisions of these two general classifications can be made as follows:

- Fluid Meters
  - Positive - Quantity
    - Weighing
      - Weighers
      - Tilting trap
    - Volumetric
      - Rotary
      - Bellows
  - Differential
    - Quantity - Current - Turbine
    - Rate of flow
      - Head (Kinetic)
        - Venturi
        - Flow nozzle
        - Orifice
        - Pitot tube
      - Area (Geometric)
        - Orifice and plug
        - Cylinder and piston
      - Head area (Weir)
        - V-notch
        - Special notch

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations:

1. Its use in a new or an old installation.
2. Method to be used in charging for the service.
3. Location of the meter.
4. Large or small quantity to be measured.
5. Temporary or permanent installation.
6. Cleanliness of the fluid to be measured.
7. Temperature of the fluid to be measured.
8. Accuracy expected.
9. Nature of flow: turbulent, pulsating, or steady.
10. Cost.
    (*a*) Purchase price.
    (*b*) Installation cost.
    (*c*) Calibration cost.
    (*d*) Maintenance cost.
11. Servicing facilities of the manufacturer.
12. Pressure at which fluid is to be metered.
13. Type of record desired as to indicating, recording or totalizing
14. Stocking of repair parts.

15. Use of open jets where steam is to be metered.
16. Metering to be done by one meter or by a combination of meters.
17. Use as a check meter.
18. Its facilities for determining or recording information other than flow.

Condensation Meters

The majority of the meters used by district heating companies in the sale of steam to their customers are condensation meters.

The condensation meter is a popular type for use on small and medium sized installations, where all of the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution has made it standard equipment with many heating companies.

Two types of condensation meters are in general use: the *tilting bucket* meter and the *revolving drum* or *rotor* meter of which there are several makes on the market. Condensation meters should not be operated under

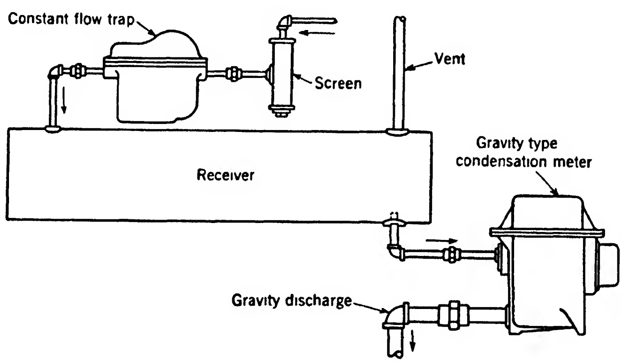


FIG. 7. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

pressure; they are made for either gravity or vacuum installation. Continuous flow traps are necessary ahead of the meter if a vented receiver is not used. Where bucket traps are used, a vented receiver before the meter is essential. If desirable a receiver may be used with a continuous flow trap, but this is not necessary.

Fig. 7 illustrates a gravity installation using a vented receiver ahead of the meter, while Fig. 8 shows a vacuum installation without a master trap.

Flow Meters

Steam flow meters are available in many types and combinations. The *orifice and plug* meter is one in which the steam flow varies directly as the area of the orifice. The vertical lift of the plug, which is proportional to the flow, is transmitted by means of a lever to an indicator and to a pencil arm which records the flow on a strip chart. The total flow over a given period is obtained by measuring the area by using a planimeter on the chart and applying the meter constant.

Fig. 9 shows a typical orifice-type meter connection and indicates typical requirements in the installation of this type of meter.

Flow meters using an orifice, Venturi tube, flow nozzle, or Pitot tube as the primary device are made by a number of manufacturers and can be obtained in either the mechanically or electrically operated type. The electric flow meter makes it possible to locate the instruments at some distance from the primary element.

Flow meters employing the orifice, Venturi tube, flow nozzle or Pitot tube should be so selected as to keep the lower operating range of the load above 20 per cent of the capacity of the meter. This is desirable for accuracy as the differential pressure at light loads is too small to properly actuate the meter. A few general points to be considered in installing a meter of this type are:

1. It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available.

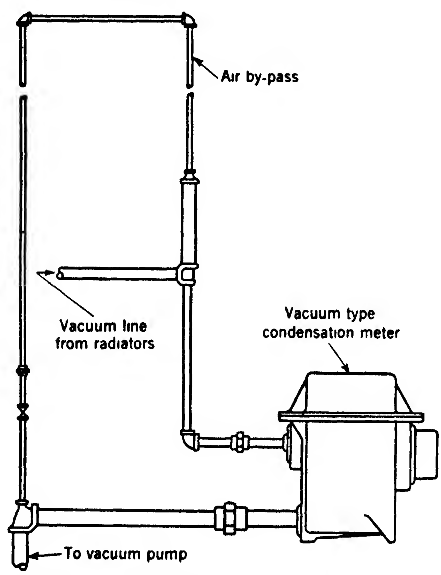


FIG. 8. VACUUM CONDENSATION METER INSTALLATION WITHOUT MASTER TRAP

2. Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company.
3. The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is above.
4. Meter piping should be kept free from leaks.
5. Sludge should not be permitted to collect in the meter body.
6. The meter body and meter piping should be kept above freezing temperatures
7. It is best not to connect a meter body to more than one service.
8. Special instructions are furnished for metering a turbulent or pulsating flow.

STEAM REQUIREMENTS

Steam requirements for heating various types of buildings are given in Chapter 12.

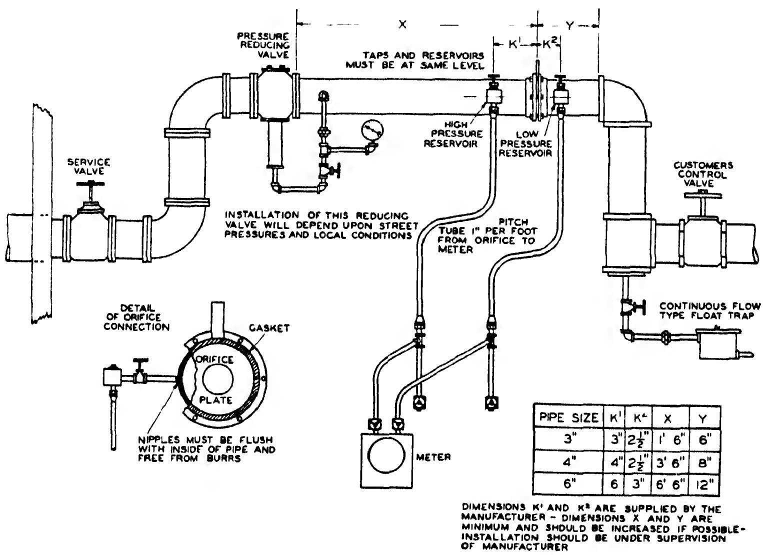
Steam requirements for water heating can be satisfactorily estimated

by using a consumption of 0.0025 lb per day per cubic foot of heated space for office buildings, and 0.0065 lb per day per cubic foot for apartment houses.

Additional data on steam requirements of various types of buildings in a number of cities may be found in the Handbook of the *National District Heating Association.*

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are



PIPE SIZE	K^1	K^2	X	Y
3"	3"	$2\frac{1}{2}$"	1' 6"	6"
4"	4"	$2\frac{1}{2}$"	3' 6"	8"
6"	6	3"	6' 6"	12"

Fig. 9. Orifice Meter Steam Supply Connection

other requirements to be met; the rate for each class of service should be based upon the cost to the utility company of the service supplied and upon the value of the service to the consumer, and it must be between these two limits. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not conform with this principle. Lastly, the rate schedule must be reasonably easy for the intelligent layman to comprehend.

Depreciation should be based on a careful estimate of the life of various elements of the property. Appropriations to reserves should be made, with generosity in good years and with discretion in less favorable years.

Glossary of Terms

Load Factor. The ratio, in per cent, of the average hourly load to the

maximum hourly load. This is usually based on a one year period but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface or required radiator surface and the demand of the particular installation. It varies from 0.25 to 0.3 lb per hour per square foot of surface.

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

A. Flat Rates.
  1. Radiator surface charge. *Obsolescent.*

B. Meter Rates.
  1. Straight-line.
  2. Step. *Obsolescent.*
  3. Block.
    (*a*) Class rates.

C. Demand Rates.
  1. Flat demand.
  2. Wright.
  3. Hopkinson.
  4. Doherty (or Three charge)

Straight-Line Meter Rate. The price charged per unit is constant, and the consumer pays in direct proportion to his consumption without regard to the difference in costs of supplying the individual customers.

Block Meter Rate. The pounds of steam consumed by a customer are divided into blocks of thousands of pounds each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules for it has the advantage of proportioning the bill according to the consumption and the cost of service It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand) and those having a low load factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefitted at the expense of the others.

Demand Rates. These refer to any method of charge based on a measured maximum load during a specified period of time.

The *flat demand rate* is usually expressed in dollars per *M* lb of demand per month or per annum. It is based on the size of a customer's installation, and is seldom used except where a flow meter is not practicable

The *Wright demand rate* is similar in calculation to the block rate except that it is expressed in terms of hours' use of the maximum demand. It is seldom used but forms the basis for other forms of rates.

The *Hopkinson demand rate* is divided into two elements:

(*a*) A charge based upon the demand, either estimated or measured;
(*b*) A charge based upon the amount of steam consumed.

This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates

The *Doherty rate* is divided into three elements:

(*a*) A charge based upon demand.
(*b*) A charge based upon steam consumed.
(*c*) A customer charge.

In the *Hopkinson rate,* the last two elements are combined into one element.

Demand rates are comparatively new and are not yet widely used; though they are equitable and competitive they are difficult for the average layman to understand.

They are of benefit to utility companies and to consumers because the investment and operating costs can be divided to suit the particular circumstances into demand, customer, and consumption groups through the use of some modification of the Hopkinson rate. Demand rates are an advantage to the customer in that the use of such a rate reduces the rate per thousand pounds to the long-hour user.

Fuel Price Surcharge. It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per M lb of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the variations without necessitating frequent changing of the whole rate structure.

UTILIZATION

Considerable savings can be made by the proper and intelligent operation of heating systems. It should be borne in mind that a heating system is designed to heat a building to 70 F inside when the outside temperature is at its lowest point for that particular locality. There is a tendency to overheat the building at any time the outside temperature is above the design temperature unless some method of regulation is used, either automatic or manual.

The general rules for economical operation are as follows:

1. Weatherstrip all windows, and calk all window frames.

2. Provide revolving or vestibule doors on all entrances. Separate shipping and receiving rooms by partitions so that the ever-open large doors will not ventilate the entire building.

3. Keep the radiation near the outside walls, under the windows, if possible.

4. Eliminate all unnecessary ventilation. Ventilating equipment is sized to meet extreme requirements Do not supply ventilation to a theater or auditorium adequate for an audience of 2000 when there are only 200 present.

5. Determine the hours that heating is required during the day and see that the steam is shut off for the maximum time at night, on Sundays, and holidays.

6 Shut steam off entirely in unoccupied sections of the building, taking care to avoid freezing the water in the plumbing system.

7. Shut off steam during the day whenever possible. During the year steam can be shut off about 55 per cent of the total daytime, the saving is proportional. An automatic control will do it, but it can be done by hand with amazingly good results.

8. Determine the temperature required for the occupancy of the building. Do not heat a storage garage or a furniture warehouse to the temperature required in a hospital ward.

9. Provide some good means of temperature control.

10. In a hot water heating system keep the temperature of the water down to correspond with existing outdoor temperatures

11. In a vacuum system maintain a high vacuum. If this is not possible, locate and eliminate all leaks.

12. Install separate lines for those parts of the building that require long-hour or all-night heating. It is much cheaper than heating the entire building all night.

13. See that the entire system responds rapidly when steam is turned on. Locate and eliminate the cause of any sluggish circulation. Balance the radiation, provide adequate air elimination, and correct any trapped run-outs to provide quick system drainage.

14. Keep the system in good repair. Worn, damaged, or defective valves and traps will not function properly.

15. Insulate all steam pipes not used as heating surface.

16. Do not obstruct radiators or prevent the free circulation of air around them; to do so seriously reduces the heating capacity of a radiator.

17. Extract the heat in the condensate for hot water or some other useful purpose.

18. Provide thermometers and recording pressure gages so that the engineer can operate the system with full knowledge of what he is accomplishing.

19. Make all valves and controls convenient and accessible, either direct or through remote control. It is only human nature to delay and avoid doing that which is inconvenient.

20. Keep a daily record consistently, based on weather requirements, and watch it every day.

21. Control the heat supplied to water tanks located on or above the roof. Such tanks require heat to prevent freezing. No heat is required when the temperature in the tank is above 32 F.

22. Investigate every complaint of *no heat* by tenants; find the cause and correct it. Do not overheat an entire building to correct a local condition in one room.

AUTOMATIC TEMPERATURE CONTROL

As stated in Chapter 37, automatic control properly applied to heating, ventilating and air conditioning systems makes possible the maintenance of desired conditions with maximum operating economy.

In addition to the large possibilities for economy, the use of adequate temperature control provides more healthful, comfortable, and efficient working conditions in the buildings because through its use the building is uniformly heated with correct temperatures and drafts from open windows and over-heating are eliminated.

There are many types of temperature control available, each adaptable to a particular type of building, but all require uniform distribution of steam and proper venting.

Before the installation of any type of modern temperature control equipment, it is necessary to see that the heating system is put in good operating condition. In general, the heating system in a building is not given the attention that other mechanical equipment is given because it will continue to function, after a fashion, even though changes in piping, location of radiation, settlement of piping, and the normal wear and tear or other changes have taken place. Through all this depreciation of the system, it becomes more and more costly to operate and parts of the building have to be greatly overheated in order to prevent underheating in a small section of the building. Vents, traps, vacuum pumps, and valves should be given a careful inspection and replaced or repaired if required. The piping should be of adequate size and graded properly. The return piping should have a careful inspection, and any pockets or lifts removed and properly vented. These inspections and repairs are not costly and prevent a much greater outlay in future years. In most cities district heating companies will be willing to make a survey of heating systems and offer recommendations as to operation and changes in piping layout.

The selection of control equipment depends upon the type and size of building and the degree of saving possible.

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PROBLEMS IN PRACTICE

1 ● a. What are the advantages and disadvantages of a low pressure distribution system?

b. High pressure?

a. The advantages of a low pressure distribution system include: (1) smaller heat loss from the pipes, (2) less trouble with traps and valves, (3) simpler problems with pressure reducing equipment at the buildings, and (4) no danger to building heating equipment from high pressure through failure of the reducing valves.

The disadvantages of a low pressure system are: (1) larger pipe sizes, and (2) decreased field of usefulness owing to small pressure range.

b. The advantages of a high pressure system are. (1) smaller pipe sizes, and (2) greater adaptability of the steam to various uses other than building heating.

The disadvantages of a high pressure system are: (1) large heat loss from the pipes, (2) the high pressure traps and valves required often give more trouble than low pressure traps and valves do, (3) extra heavy fittings are required, and (4) usually two reducing valves or some form of emergency relief is necessary to protect the building piping system.

2 ● What points should be borne in mind when laying out an underground steam conduit?

The conduit should be reasonably water-proof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or the conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit.

3 ● What is considered the proper pressure for a hydrostatic test before completing the conduit?

In the case of any underground piping which is to be buried or otherwise made inaccessible, the assembled lines shall first be tested hydrostatically at a pressure of one and one-half times the maximum allowable service pressure and held for a period of at least two hours without evidence of leakage. In any case the hydrostatic pressure should not be less than 100 lb per square inch

4 ● What are the common methods for salvaging heat in condensate?

The most common methods are: (1) the use of a water heating economizer for preheating the hot water supply to the building, and (2) the use of a cooling radiator.

5 ● Is the steam consumption less in a building that shuts off its steam at night than in one that does not? Why?

It has been thoroughly demonstrated that the steam consumption is less in a building where the steam is shut off at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

6 ● What are the common methods used to graduate the heat supply according to variations in outside temperature?

a. Thermostats controlled by building temperatures and operating a motor valve.

b. A weather compensating thermostat will regulate the steam supply according to outside temperatures. These controls can be arranged to give short intervals of intermittent steam supply. Those without time clocks can be equipped with suitable timing arrangements.

c. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds to produce some control over the heat output. There are systems on the market which will do this automatically.

d. The use of orifice systems insures proper distribution and graduation of heat supply.

e. The time-limit control which may be set to provide no service, continuous service, or periodic service, is also used. For periodic service, steam may be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period This type of service is provided by several intermittent settings A night switch will maintain the intermittent day setting, or interrupt the day operation and cut off the supply of steam at night during any desired hours.

7 ● What is the common method of determining the size of mains in a distribution system?

On the basis of pressure drop: The initial pressure and the minimum permissible terminal pressure are specified, and the pipe sizes are so chosen that the maximum estimated amount of steam may be transmitted without exceeding this pressure difference. The steam's velocity is disregarded and it may reach a magnitude in excess of 35,000 fpm which is not considered high.

8 ● Determine the size of pipe from the following data using Unwin's formula: length of pipe, 600 ft; steam to be carried, 90,000 lb per hour, dry saturated; initial pressure, 100 lb per square inch, gage; and final pressure, 40 lb per square inch, gage.

The pressure drop, $P = 100 - 40 = 60$ lb per square inch.

The weight of steam per minute, $W = \dfrac{90{,}000}{60} = 1500.$

The length of pipe in feet, $L = 600$

The average density of steam d in pounds per cubic foot, taken from Keenan's Tables:

$$\begin{aligned} \text{At 100-lb gage, } d &= 0.2578 \\ \text{At 40-lb gage, } d &= 0.1285 \\ \text{Average, } d &= 0.1932 \end{aligned}$$

The diameter of the pipe in inches $= D$.

Substituting the values in the Formula (1):

$$60 = \frac{0.0001306 \times 1500^2 \times 600 \left(1 + \frac{3.6}{D}\right)}{0.1932 \times D^5}$$

$$D = 7.35 \text{ in.}$$

Therefore, an 8-in. pipe should be used.

Chapter 43

WATER SUPPLY PIPING AND WATER HEATING

Maximum Possible Flow, Maximum Probable Flow, Average Probable Flow, Factor of Usage, Kind of Pipe Used, Sizing of Risers, Sizing of Mains, Sizing of Systems, Hot Water Supply, Hot Water Heating, Hot Water Storage, Swimming Pool Heating Requirements

DOMESTIC water supply systems present the engineer with a design problem that requires combining the somewhat empirical rules and formulæ in use with the more or less exact hydraulic principles involved. Unlike heating and ventilating layouts, there are practically no definite data for estimating the quantity of water likely to be consumed or the probable rate of water flow at any particular moment.

Metered results in one building often show two or three times the metered amount in another building of the same size and with the same type of tenants. In hotels, one riser will often have an almost constant flow that may never be reached by another at peak load. In office buildings, the women's toilets show a far greater daily consumption than those of the men, yet at no time will they approach the hourly consumption of the men's toilet during the first hour of the day. This condition has led to a multiplicity of rules of practice which vary as much as the data used. All must of necessity be based on an assumed rate of consumption and on an assumed probability of simultaneous use, and while the formulæ employed may have been derived on sound technical basis the assumptions are often in error.

To arrive at a safe standard, the approximate rate of flow of each fixture to be supplied must be known and the probable number of fixtures in use at any one time must be assumed. Obviously, the maximum number of fixtures assumed to be in use must be taken at the peak of demand and the lines must be made adequate to supply such a peak regardless of the riser or branch on which the demand may occur. This means that all water piping under the usual conditions will be over-sized.

In tall buildings it is customary to divide the water supply systems, both hot and cold, into sections of 10 to 20 stories. Such *zoning*[1] or

[1]It is impractical to attempt to size piping so as to produce the proper pressure on fixtures at different levels by employing friction, owing to the fact that this friction will be built up to the amount desired only in times of maximum demand and at all other times the friction will be only a fraction of the maximum friction so that the fixtures by this method are subjected to a varying pressure on the water supply line. A much more practical method is to throttle the flow at the fixture, or to use flow regulators, so that the quantity of water delivered will approximate the fixture demands and so that this is accomplished without splashing or noise.

sectionalizing is for the purpose of avoiding excessive pressures on the fixtures in the lower stories of each system. This limits the consideration of water pipe sizes to horizontal mains and to risers not exceeding 20 stories in height or about 200 ft.

For the purpose of this chapter the following terms will be used and should be clearly distinguished from one another:

Maximum Possible Flow: The flow which would occur if the outlets on all fixtures were opened simultaneously. This condition is seldom, if ever, obtained in actual practice except in cases of gang showers controlled from one common valve, and similar conditions.

Maximum Probable Flow: The maximum flow which any pipe is likely to carry under the peak conditions. This is the most important amount to be considered in pipe sizing.

Average Probable Flow: The flow likely to be required through the line under normal conditions.

It is evident that any pipe adequate to take care of the *maximum*

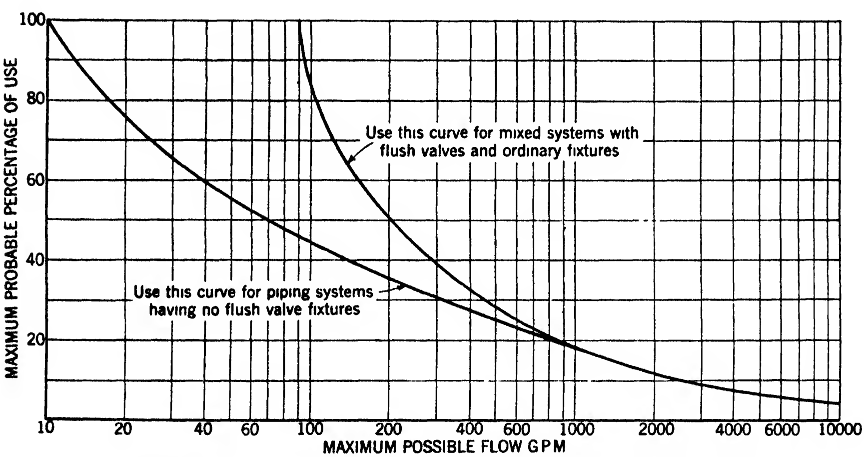


FIG. 1. CHART SHOWING RELATION BETWEEN MAXIMUM POSSIBLE FLOW AND MAXIMUM PROBABLE PERCENTAGE OF USE

probable flow will also be more than able to take care of the *average probable flow*, and hence the latter has no bearing on the pipe size.

MAXIMUM PROBABLE FLOW

There are two factors to be considered in calculating the maximum probable flow, namely, (1) the quantity of water that will flow from the outlets when they are open, and (2) the number of outlets likely to be open *at the same time*. Table 1 shows the maximum approximate rate of flow from each fixture when it is in use, and will serve as a guide in estimating maximum probable flow demands although there is considerable variation in different fixtures and valves. Probably the flow under normal water pressures, or with the pressure properly throttled, will not differ greatly from the values stated. With the aid of this table it is possible to calculate the maximum possible flow with all outlets open in both the hot and cold water lines.

To obtain the maximum probable flow it is necessary to multiply the maximum possible flow by a factor of usage, and this factor varies with the installation and the number of fixtures in the installation. It is evident that with two fixtures it is quite possible that both will at some time be in operation simultaneously. With 200 fixtures, it is unlikely the entire 200 would ever operate at the same time. Consequently, the factor of usage reduces as the number of fixtures becomes greater, all other things being equal.

TABLE 1. APPROXIMATE FLOW FROM FIXTURES UNDER NORMAL WATER PRESSURES

FIXTURES	COLD WATER (GALLONS PER MINUTE)	HOT WATER (GALLONS PER MINUTE)
Water-closets, flush valve	45[a]	0
Water-closets, flush tank	10	0
Urinals, flush valve	30[a]	0
Urinals, flush tank	10	0
Urinals, automatic tank	1	0
Urinals, perforated pipe per foot	10	0
Lavatories	3	3
Showers, 4 in. heads, ½ in. inlets	3	3
Showers, 6 in. heads or larger	6	6
Needle bath	30	30
Shampoo spray	1	1
Liver spray	2	2
Manicure table	1½	1½
Baths, tub	5	5
Kitchen sink	4	4
Pantry sink, ordinary	2	2
Pantry sink, large bibb	6	6
Slop sinks	6	6
Wash trays	3	3
Laundry tray	6	6
Garden hose bibb	10	0

[a]Actual tests on water-closet flush valves indicate 40 gpm as the maximum rate of flow with 30 lb pressure at the valve; this would increase to 60 gpm (about 50 per cent) at 90 lb pressure. The 45 gpm has been taken as an average flow; possibly, with very low pressures just sufficient to operate the flush valve, 30 gpm could be allowed with safety Urinal flush valves would vary proportionately in the same manner

In practice all the elements will vary according to conditions; in the case of flush valve closets the duration of flush with the kind and condition of supply apparatus, the interval between flushes with the number of people using the system and their habits; and the length of the rush period with the type of installation and its location. The effect of each of these time elements on the results should be considered in connection with any data on which it is based before passing judgment on the selection of the factor of usage. The longer the duration of the flush the greater is the probability of overlapping flow. In selecting the factor of usage shown in Fig. 1 for systems having flush valves, 10 seconds was chosen as the maximum duration of flush, a value that represents an approximate average as water closets are installed.

While the curve has been calculated for systems composed of water closets alone, it is possible to calculate probabilities for mixed systems of water closets and other smaller fixtures. It has been found however that for two systems both having the same maximum possible flow, one composed entirely of water closets and the other a mixed system of water

closets and smaller fixtures, the probability of a given rate of flow is greater for the system composed of water closets than for the mixed

TABLE 2. SCHEDULE OF SIZES FOR DOWN-FEED RISER (SEE FIG. 2)

Portion of Riser	Allowable Drop per Lb per 100 Ft	Maximum Probable Flow, Gallons per Minute																
		5	10	15	20	25	30	40	50	60	70	80	90	100	125	150	200	250
T	35	3/4	1	1 1/4	1 1/4	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2	2 1/2	3	3	3	3 1/2	3 1/2
S	20	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	2	2	2	2	2	2 1/2	3
R	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
Q	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
P	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
O	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
N	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
M	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
L	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
K	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
J	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
I	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
H	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
G	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
F	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
E	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
D	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
C	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
B	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
A	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2

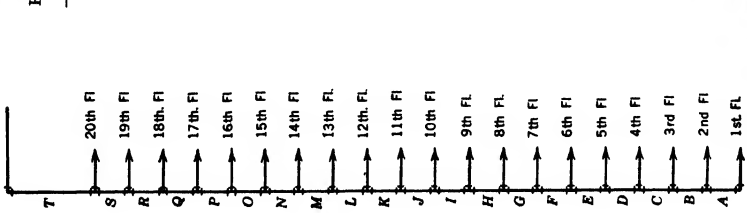


FIG. 2. TYPICAL RISER FOR 20-STORY BUILDING

system. The use of this chart then would produce results which would be on the safe side for mixed systems.

For systems composed entirely of fixtures other than flush valve fixtures the curve has been extended for smaller maximum possible flow values.

This chart applies to a normal building and not to installations where the inmates may all be required, for instance to bathe on certain days of the week and at certain hours of those days; or in schools for example where all the showers in the gymnasium may be used simultaneously after instruction periods. In such special cases a new factor of usage must be developed based on the maximum probable usage under the conditions involved.

Example 1. Assume that in a normal building, such as a residential hotel or an apartment house, there are 50 flush valve water-closets, 50 lavatories, 50 sinks and 50 baths, and that it is desired to determine the maximum probable flow in a line supplying all of these fixtures with both hot and cold water.

Cold Water

50 W. C. x 45 gpm .... 2250 gpm
50 Lavs x 3 gpm .... 150 gpm
50 Sinks x 4 gpm .... 200 gpm
50 Baths x 5 gpm .... 250 gpm

Maximum possible flow .... 2850 gpm

Fig 1 shows a factor of usage of 9 per cent.

Maximum probable flow of cold water is 2850 × 0 09 .... 257 gpm

Hot Water

50 Lavs x 3 gpm .... 150 gpm
50 Sinks x 4 gpm .... 200 gpm
50 Baths x 5 gpm .... 250 gpm

Maximum possible flow .... 600 gpm

Fig 1 shows a factor of usage of 23 per cent

Maximum probable flow of hot water is 600 × 0 23 .... 138 gpm

Total for main supplying cold and hot water (2850 + 600) × 0 08 .... 276 gpm

It should be noted that this is a *rate of flow* or an *instantaneous demand*

KIND OF PIPE USED

Before entering into the actual sizing of pipe, it is necessary to consider the kind of pipe to be used, and to make suitable allowance for corrosion and fouling during the lifetime of the system. For example, if brass, copper or alloy pipe is contemplated, it is probable that the quantities indicated in Example 1 are ample; if galvanized pipe is to be used, then it is quite likely that after a period of say 15 years the area may be decreased as much as 25 per cent and the quantitities of water assumed should be increased by 35 per cent to allow for this reduction of area; if the water contains lime it is possible that 50 per cent of the area may be lost and in such cases the flow should be doubled and no branch pipe connected to fixtures should be less than ¾ in. In all of the following calculations, the assumption is made that the water is fairly good and that a corrosion resistant type of pipe is to be used.

SIZING A DOWN-FEED RISER

Down-feed systems are commonly used for tall buildings. In sizing a riser arranged for down-feed, the gravity head permits a pressure drop that is almost prohibitive in an up-feed riser. There is a gain in riser head of 0.43 × 100 or 43 lb per 100 ft of run and hence it is quite permissible to size such a riser on the basis of a pressure drop of 30 lb per 100 ft of run, as the difference between the 43 lb generated and the 30 lb drop under maximum probable demand is ample to take care of the friction caused by the fittings. This method applied to the typical riser shown in Fig. 2 gives the schedule of sizes indicated in Table 2 for any flow from 5 to 250 gal.

SIZING AN UP-FEED RISER

When the riser is an up-feed, the opposite condition occurs; that is, there is a drop in pressure as the top of the riser is approached, due to the natural reduction in the gravity pressure, and to this must be added the pipe friction plus that introduced by the pipe fittings, all of which produce an excessive drop when compared to the conditions existing with a down-feed riser.

To size an up-feed riser the minimum pressure of the street main, or other source of supply, should be ascertained and from this should be subtracted the pressure to be maintained at the highest fixture, namely, 15 lb per square inch, plus the height in feet above the source of water pressure, multiplied by 0.43 to change from feet of head to pounds of pressure. The total length of run from the source of pressure to the farthest and highest fixture should be ascertained, and this should be changed to equivalent length of run to allow for the loss occasioned by the pipe fittings. Table 3 gives the additional lengths necessary to allow for the various fittings and valves. The drop allowable in pressure per 100 ft of run may then be obtained by multiplying the surplus pressure (over that required for the gravity head and to supply 15 lb at the fixture) by 100 and by dividing this by the equivalent length of run to the farthest or highest fixture.

TABLE 3. APPROXIMATE ALLOWANCES FOR FITTINGS AND VALVES IN FEET OF STRAIGHT PIPE

Size of Pipe (Inches)	Type of Fitting or Valve					
	90-Deg Elbow	45-Deg Elbow	Return Bend	Gate Valve	Globe Valve	Angle Valve
½	4	3	8	2	48	8
¾	5	3	10	3	60	10
1	5	3	10	3	60	10
1¼	6	4	12	3	72	12
1½	7	5	14	4	84	14
2	7	5	14	4	84	14
2½	10	7	20	5	120	20
3	12	8	24	6	144	24
4	18	13	36	9	216	36
5	25	18	50	13	300	50
6	30	21	60	15	360	60

Where street water pressures are available the pressure drop through the meter and service pipe must be taken into consideration. Table 4 shows the pressure loss through meters. It also gives the minimum sizes of recommended service and maximum meter deliveries.

Example 2. Assume a street pressure of 60 lb, the height of the highest fixture 50 ft, and the length of the longest run 200 ft. Without knowing the additional length of pipe to be added for the fittings it will be assumed that this is about 100 ft. The surplus pressure which will be available for pressure drop will then be 60 lb − (15 lb + 50 ft × 0.43 lb) = 60 lb − (15 lb + 21.5 lb) = 23.5 lb.

To change this into drop per 100 ft: $\frac{23.5 \text{ lb} \times 100}{200 \text{ ft} + 100 \text{ ft}} = 7.8$ lb per 100 ft.

The pipe may then be sized from the maximum probable flow by selecting a size that does not give a drop in excess of 7.8 lb per 100 ft.

It will be seen from Example 2 that it is impossible to size up-feed risers without determining the drop allowable in both the horizontal feed mains and the toilet room branches. Having once ascertained this allowable drop, it is simply a matter of applying it throughout the system.

TABLE 4. PRESSURE LOSS THROUGH WATER DISC METERS[a]
A. W. W. A. Standards

RATE OF FLOW GPM	APPROX PRESSURE LOSS THROUGH METERS, LB PER SQ IN PIPE SIZE (IN)							
	⅝	¾	1	1½	2	3	4	6
5	1.5	0.5	0 2					
10	6.0	2.0	1 0	0.2				
15	14.0	5.0	2.0	0 6	0.2			
20	25 0	9.0	3 5	1.0	0.4			
25		13.5	5 5	1.5	0.6			
30		19.5	8.0	2.0	0.9			
35			11.0	3.0	1.0			
40			14.0	4.0	1.5			
45			18 0	5.0	2.0			
50			22.0	6.0	2.5	0.7		
75				14 0	5.5	1 5		
100				25 0	10.0	2.8	1.0	
125					15.0	4.0	1.5	
150					22.0	6 0	2 2	
175						8 0	3 0	
200						10.4	4.0	1.0
250							16.0	1.5
300							23.0	2.2
350								3.0
400								4.0
500								6 5
600								9.0
800								16 0
1000								25.0

MINIMUM SIZE OF SERVICE RECOMMENDED

RATE OF FLOW GPM	APPROX MINIMUM PIPE SIZE OF SERVICE, MAIN TO METER (IN) MAXIMUM LENGTH (FT)				
	30	75	100	150	200
1–20	¾	¾	1	1	1
20–30	¾	1	1	1½	1½
30–50	1	1½	1½	1½	1½
50–100	1½	1½	2	2	2
100–150	1½	2	2	2½	2½

SAFE MAXIMUM DELIVERY OF METERS

METER SIZE IN	CAPACITY, GPM BASED ON 25 LB LOSS THROUGH METER
⅝	20
¾	34
1	53
1½	100
2	160
3	315
6	500
8	1000

[a]Pressure loss through compound and current meters are less than shown in table. For exact information consult manufacturers

HORIZONTAL SUPPLY MAINS

The horizontal mains supplying the risers at the top of a down-feed system must be liberally sized unless the house tank is set at a much higher elevation than usual. To provide a gravity head on the highest fixtures of 15 lb per square inch it is necessary for the water line in the house tank to be nearly 40 ft higher, and with the line loss considered this becomes about 45 ft. Such heights are not often practical and as a result the pressure on the highest fixtures either is reduced to 7 lb (which is sufficient to operate a flush valve), or flush tank water-closets are substituted, or a separate cold and hot water supply is installed with a small pneumatic tank to give the increase in pressure necessary. The chief objection to the use of a pneumatic tank is that a separate hot water heater is required and this heater must be located either sufficiently below the highest fixtures to obtain a gravity circulation, or it must be provided with a circulating pump in order to force the hot water to the top floor level.

The most common solution is to place the house tank as high as the structural and architectural conditions will permit and then to use liberally-sized lines between the house tank and the upper fixtures, say for the two top stories, below which the riser sizes may be reduced to those indicated in Fig. 2 and Table 2. Where the house tank is only one story above the top fixtures, flush tank water-closets must be used and the drop in the entire run from the house tank down to the farthest fixture should not exceed 1 lb; the less, the better. This means that if the total equivalent run to the farthest top fixtures supplied is 300 ft, the drop per 100 ft should not exceed $\frac{1 \text{ lb} \times 100}{300}$ or 0.33 lb per 100 ft. The friction curves shown in Fig. 3 may be used for quickly determining the proper size of pipe to give any desired drop in pounds per 100 ft of equivalent run.

OVERHEAD DISTRIBUTION MAIN

Example 3. Suppose an installation has a house tank in which the water line is 20 ft above the level of the top fixtures to be supplied and that the length of run to the farthest fixtures on this level is 400 ft with the pipe fittings adding another 200 ft, making an equivalent length of 600 ft. What would be the size of main coming out of the tank where a maximum flow rate of 400 gpm may be expected, of the horizontal main where a maximum flow rate of 200 gpm may be expected, and of the riser down to the fixture level where the maximum flow rate is approximately 100 gpm?

Here the level of the water in the house tank is 20 ft above the faucet of the highest fixture and the gravity pressure will be 0.43 lb $\times$ 20 ft = 8.6 lb and, if a total pressure drop of 1 lb is assumed, the pressure on the farthest fixture under times of peak load will be 8.6 lb $-$ 1 lb = 7.6 lb while the drop per 100 ft of equivalent run will have to be $\frac{1 \text{ lb} \times 100}{600} = 0.1667$ lb.

Referring to Fig. 3 it will be noted that where the flow through the main is 400 gpm, an 8-in. pipe would be required; that where the flow is reduced to 200 gpm, a 6-in. pipe would be sufficient; and that where the flow is 100 gpm in the riser branch and riser, a 5-in. size would be correct. Of course these are somewhat excessive flows and the head from the tank is small so that large sizes are to be expected. It would be necessary to carry a 5-in. riser down to the branch to the top floor, then reduce to 4 in. for the branch to the floor below the top, and below this the sizes in Table 2 could be followed. In such a case, flush tank closets should doubtless be substituted.

Had the tank been set 10 ft higher, the head available to be used up in friction, but

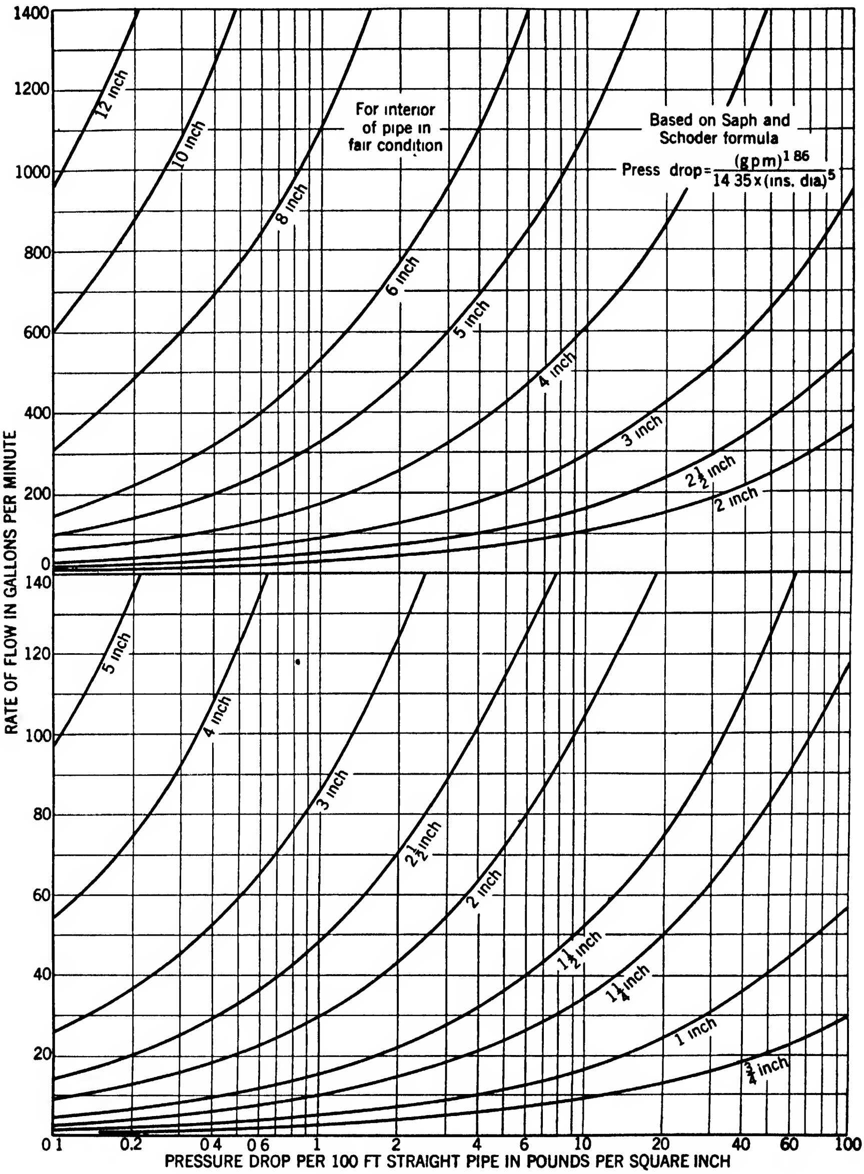
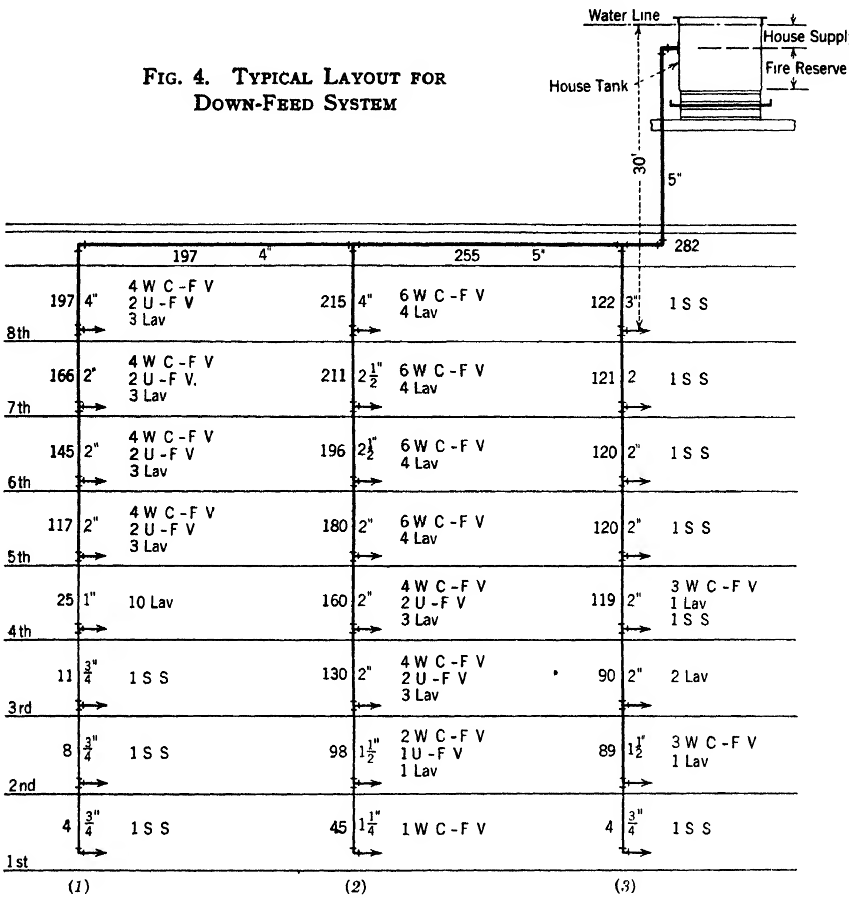


FIG. 3. CHART GIVING PRESSURE DROP FOR VARIOUS RATES OF FLOW OF WATER

still giving the same pressure at the top fixtures, would have been 0.43 lb × 10 ft or 4.3 lb greater and this, with the 1 lb drop used previously, would give a total allowable drop of 1 lb + 4.3 lb = 5.3 lb which, divided by the 600 ft equivalent run gives a drop per 100 ft of $\frac{5.3 \times 100}{600} = \mathbf{0.9\ lb}$

FIG. 4. TYPICAL LAYOUT FOR DOWN-FEED SYSTEM



and, with this drop, the sizes according to the chart (Fig. 3) are 6 in., 5 in., and 4 in., respectively, while if the run is reduced to 200 ft instead of 600 ft, the allowable drop will be $\frac{5.3 \text{ lb} \times 100}{200} = 2.7$ lb per 100 ft. This gives 5 in., 4 in., and 3 in., respectively, for the flows of 400, 200, and 100 gpm.

From Example 3 it is evident that, while the down-feed system possesses certain economies in size for the riser portion, it is quite likely to involve large distribution main sizes, especially when the tank is not elevated to a considerable degree.

SIZING A PIPING SYSTEM

Example 4. Fig. 4 shows a typical layout with three risers extending eight stories and with the fixtures noted on each floor. First this will be solved for a down-feed arrangement assuming that the level of the water in the house tank is 30 ft above the fixtures on the top floor, that the length of run from the tank to the farthest fixture is 200 ft, equivalent length of fittings 100 ft, and the pressure required at the fixture is 7 lb.

TABLE 5. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 1. Fig 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 S S	4	4	4	100	4	30	¾
2nd	1 S S	4	4	8	100	8	30	¾
3rd	1 S S	4	4	12	92	11	30	¾
4th	10 Lav	3	30	42	58	25	30	1
5th	4 W. C. 2 U 3 Lav	45 30 3	180 60 9 249	291	40	117	30	2
6th	4 W C. 2 U 3 Lav.	45 30 3	180 60 9 249	540	27	145	30	2
7th	4 W C 2 U 3 Lav	45 30 3	180 60 9 249	789	21	166	30	2
8th	4 W C 2 U 3 Lav	45 30 3	180 60 9 249	1038	19	197	2	4

The 30-ft head is equal to a static pressure of 0.43 × 30 or 12.9 lb per square inch and to maintain a pressure of 7 lb at the highest fixtures the drop allowable in pressure is 12.9 − 7.0 lb or 5.9 lb. As the total equivalent run is 300 ft, this is a drop per 100 ft of 1.97 lb, or practically 2 lb. Therefore, all risers and mains from the top floor back to the tank must be sized on the basis of a drop of 2 lb per 100 ft. Tables 5, 6, 7 and 8 show the schedule for Risers Nos. 1, 2 and 3 with the maximum possible flow taken from Table 1, the percentage of use at the peak taken from Fig. 1, and the maximum probable flow at the peak worked out for each portion of the riser, the riser sizes being taken from Table 2 as far as possible and from Fig. 3 where the amounts exceed the values given in this table; a drop of 30 lb per 100 ft is used except on the riser from the top floor back to the tank where 2 lb per 100 ft is the allowable limit.

The reduction in pipe size which would occur if flush tank water-closets were used on the top floor and only 3 lb pressure used on the fixtures is given in Tables 9 and 10. This illustrates why flush tank closets so frequently are substituted on the uppermost floor when a house tank is the source of water pressure.

If it is now assumed that Riser No. 1 is to be fed from the bottom and the minimum street pressure is 75 lb with the top fixture of the riser 80 ft above the main, the problem would be solved by determining the maximum rate of flow in each portion of the riser as shown in Table 11 and then finding the allowable drop which can be used per 100 ft. The 80 ft of riser height will use up 0.43 lb × 80 = 34.4 lb and the pressure at the top of the required 15 lb will make the total reduction 49.4 lb, leaving a balance of 25.6 lb which may be used up in friction. If the distance from the street main to the bottom of the riser, which will be assumed to be the farthest one on the horizontal line, is 100 ft, and if the fittings are sufficient to add another 100 ft, as well as the 80 ft of vertical distance up the riser, the total equivalent run will be 280 ft, which will be taken as an even 300 ft.

TABLE 6. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(*Riser No. 2. Fig. 4*)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In
1st	1 W C.	45	45	45	100	45	30	1¼
2nd	2 W. C. 1 U 1 Lav.	45 30 3	90 30 3					
			123	168	58	98	30	1½
3rd	4 W. C 2 U 3 Lav.	45 30 3	180 60 9					
			249	417	31	130	30	2
4th	4 W. C. 2 U 3 Lav.	45 30 3	180 60 9					
			249	666	24	160	30	2
5th	6 W C 4 Lav	45 3	270 12					
			282	948	19	180	30	2
6th	6 W C 4 Lav.	45 3	270 12					
			282	1230	16	196	30	2½
7th	6 W. C 4 Lav	45 3	270 12					
			282	1512	14	211	30	2½
8th	6 W C. 4 Lav.	45 3	270 12					
			282	1794	12	215	2	4

Then the allowable drop per 100 ft will be $\frac{25.6 \text{ lb} \times 100}{300} = 8.5$ lb and the sizes shown in Fig. 5 are based on this amount of drop. Of course the other risers will have the same maximum flows at the bottom as they formerly had at the top, namely 215 and 122 gal, respectively, for Risers Nos. 2 and 3. Combining these maximum flows in the same manner as pursued in the down-feed system it is seen that the maximum flow between Riser No. 2 and Riser No. 3 is 255 gpm, and between Riser No. 3 and the street main, 282 gpm which at a drop of 8.5 lb gives the main sizes indicated. It will be noted that in determining the maximum flow in an up-feed riser it is necessary to begin at the top floor and work down instead of beginning at the bottom floor and working up as was done in the down-feed sizing.

SIZING UP-FEED AND DOWN-FEED HOT WATER SYSTEMS

Hot water supply systems, when of the circulating type, have a few differences to be considered although the same general principles of sizing apply to these lines as to the cold water lines. Owing to the fact that there are no flush valves on the hot water piping and also because many plumbing fixtures have no hot water connections, the sizes of the hot water piping in general will be considerably less than the cold water piping in the same building. On the other hand it is almost invariably

TABLE 7. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(*Riser No. 3. Fig. 4*)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN
1st	1 S S	4	4	4	100	4	30	¾
2nd	3 W. C. 1 Lav.	45 3	135 3 138	142	63	89	30	1½
3rd	2 Lav	3	6	148	61	90	30	1½
4th	3 W C. 1 Lav 1 S S.	45 3 4	135 3 4 142	290	41	119	30	2
5th	1 S S.	4	4	294	41	120	30	2
6th	1 S S	4	4	298	40	120	30	2
7th	1 S S	4	4	302	40	121	30	2
8th	1 S S	4	4	306	40	122	2	3

required that a gravity circulation be kept up in such hot water lines and this often has a considerable influence on the size. There are three methods of arranging circulation lines, as follows:

1. By using the plain up-feed with a return carried back from the top of the riser and paralleling it.
2. By carrying a supply riser up in one location thus supplying fixtures on up-feed, then crossing over at the top and coming down past another collection of fixtures and supplying these by a down-feed
3. By carrying all of the water to the top of the building and dropping risers wherever needed, feeding all hot water on a down-feed system.

TABLE 8. SIZE OF DISTRIBUTION MAIN FOR DOWN-FEED SYSTEMS (SEE FIG. 4)

RISER NO	MAXIMUM GPM RISER	MAXIMUM GPM MAIN	PROBABLE USE (PER CENT)	PROBABLE GPM	ALLOWABLE DROP LB PER 100 FT	SIZE OF MAIN IN
1	1038	1038	18	187	2	4
2	1794	2832	9	255	2	4
3	306	3138	9	282	2	5

In the first instance the up-feed riser may be sized for the same pressure drop as used for the cold water riser and, from the top of the riser *just below the top fixture connection*, a return circulation line may be carried back to the main return line in the basement and connected through a check valve, set on a 45-deg angle, and a gate valve; these return circulation lines should never be less than ¾ in., and on the farther half of the risers, not less than 1 in. to favor circulation in the far end. Typical top and bottom connections for such risers are shown in Fig. 6.

TABLE 9. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISERS WITH FLUSH TANK WATER-CLOSETS AND URINALS ON TOP FLOOR ONLY (SEE FIG. 4)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In.
	Riser No. 1							
7th and below				789	21	166	30	2
8th	4 W. C. 2 U 3 Lav	10 10 3	40 20 9 69	858	20	172	3.3	4
	Riser No. 2							
7th and below				1512	14	211	30	2½
8th	6 W C. 4 Lav.	10 3	60 12 72	1594	14	223	3.3	4
	Riser No. 3							
7th and below				302	40	121	30	2
8th	1 S S	4	4	306	40	122	3.3	3

For the second arrangement of hot water risers (Fig. 7*b*), circulation lines are run back from the last fixture supplied to the main return circulation line in the same manner as just described, using ¾ in. for the near risers and 1 in. for the far risers. The sizing is much more difficult, as it is necessary to start at the bottom floor of the return riser and work back to the top of this riser and then carry the maximum flow across on to the top of the corresponding supply riser and work down on this riser from the top floor to the bottom. Naturally this gives a much greater flow in the supply riser and aids circulation by reducing pipe friction. The allowable loss per 100 ft in such lines must be made about half that used for the cold water risers which do not have the combined up- and down-travel which the hot water must make.

In the third and most common arrangement (Fig. 7*c*) all of the water is carried from the tank or heater directly to the top of the building and is there distributed to the risers which are down-feed and may be sized in the

TABLE 10. SUMMARY OF RISER SIZES TO GIVEN MAIN SIZES WITH FLUSH TANK WATER-CLOSETS AND URINALS ON TOP FLOOR ONLY. (SEE FIG. 4)

Riser No.	Maximum Gpm Riser	Maximum Gpm Main	Probable Use (per cent)	Probable Gpm	Allowable Drop Lb per 100 Ft	Size of Main In.
1	858	858	20	172	3.3	4
2	1594	2452	10	245	3.3	4
3	306	2758	9	248	3.3	4

regular down-feed manner if the total equivalent run either from the street main or house tank is taken into consideration. The return circulation lines from the bottom of each riser should be arranged in the manner already outlined and any riser not going to the basement to supply fixtures must have these returns carried down to the basement from the termination of the supply riser at whatever level it may end.

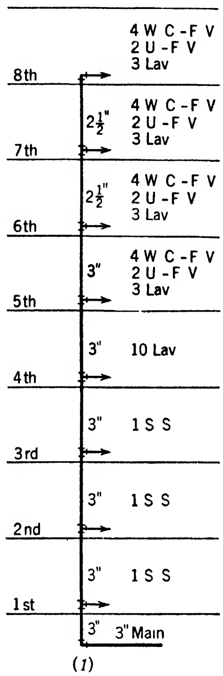


FIG. 5. UP-FEED SYSTEM

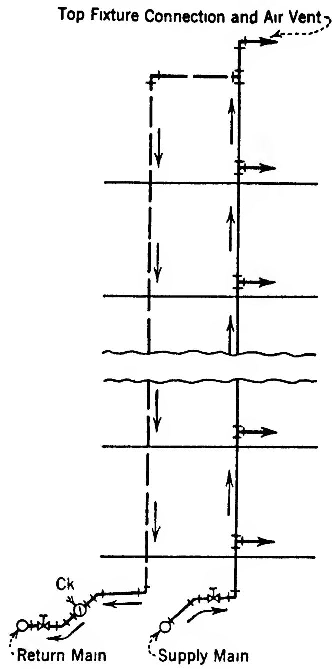


FIG. 6. SUPPLY AND RETURN MAIN CONNECTIONS FOR HOT WATER SUPPLY SYSTEM

All risers, both hot and cold, should be valved at the main with an extra check valve on the hot water return circulation so that the risers may be cut off and repaired when necessary without disturbing the service in the remainder of the system.

HOT WATER SUPPLY

Having designed the service hot water piping, the next step is to furnish some means of heating the water and in this respect it is necessary to pass from the maximum probable flow to the *maximum probable hourly demand*, which is quite different. If an instantaneous heater were used, it would require adequate capacity to provide for the heating of the water as fast as it is drawn and a heater of this type should be sized on the basis of the maximum probable flow with the accompanying heavy drafts on the heating device and with intervals of no draft at all. To balance these inequalities of flow the storage-type heater is often utilized so that the

TABLE 11. TYPICAL CALCULATION OF PIPE SIZES ON UP-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS (SEE FIG. 5)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN
8th	4 W. C 2 U 3 Lav	45 30 3	180 60 9					
			249	249	44	109	8.5	2½
7th	4 W. C. 2 U 3 Lav	45 30 3	180 60 9					
			249	498	28	139	8.5	2½
6th	4 W. C 2 U 3 Lav	45 30 3	180 60 9					
			249	747	22	164	8 5	3
5th	4 W. C 2 U 3 Lav.	45 30 3	180 60 9					
			249	996	18	179	8.5	3
4th	10 Lav.	3	30	1026	18	185	8 5	3
3rd	1 S S	4	4	1030	18	186	8 5	3
2nd	1 S. S.	4	4	1034	18	187	8 5	3
1st	1 S S	4	4	1038	18	188	8 5	3

TABLE 12. SUGGESTED STORAGE TANK SIZES FOR HOMES AND APARTMENTS

ALL YEAR SERVICE BASED ON BOILER WATER AT 180 F				SERVICE DURING HEATING SEASON BASED ON BOILER WATER AT 215 F			
Tank Capacity Gal	Piping Connections		Number of Baths or Families	Tank Capacity Gal	Piping Connections		Number of Baths or Families
	Boiler, In	Tank, In.			Boiler, In	Tank, In	
30	1	¾	1	30	1	¾	1
35	1¼	¾	1	40	1	¾	1
40	1¼	¾	1–2	52	1	¾	1
50	1¼	¾	1–2	66	1¼	1	1–2
60	1¼	1	1–2	82	1¼	1	2–3
72	1½	1	2–3	100	1¼	1	3
80	2	1	2–3	120	1¼	1	4
100	2	1¼	3–4	144	1½	1	5
125	2	1¼	4–5	160	2	1¼	6
150	2	1¼	5–6	200	2	1¼	6–7
200	2	1½	6–7	250	2	1½	7–9
250	2½	1½	7–9	300	2	1½	9–11
300	2½	1½	9–11	400	2½	2	11–15
400	3	2	11–15	500	2½	2	15–18
500	3	2	15–18	600	3	2½	18–21

water demand can be heated during periods of light demand and stored up for use during the periods of heavy demand. The total water consumption per person usually varies between 100 and 150 gal per day when laundry and culinary operations for the occupants are carried out on the same premises. The maximum hourly demand under these conditions will be found to be about one-tenth of the average daily consumption.

If one-third of the total water used is hot water and 125 gal per day is assumed as a fair average of consumption per person, it is apparent that each person uses about 40 gal of hot water per day. If one-tenth of this represents the peak hourly load, then 4 gph must be allowed per person for the heaviest demand. If the average occupancy of apartments is 3 persons, the peak hour demand per apartment will be about 12 gph. It is customary to allow 10 gph of heating capacity per apartment. Water in excess of this heating capacity drawn out during the peak hours is

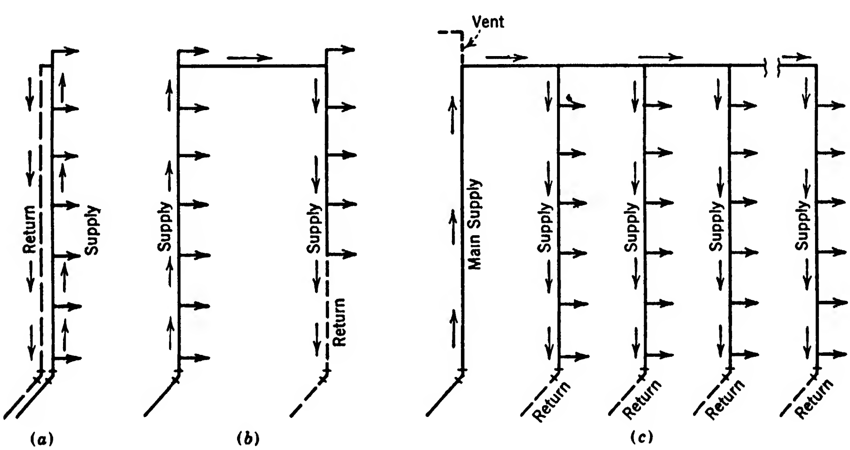


FIG. 7. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

provided for by storage in the hot water tank where this water is heated during hours when the demand is below the average. Table 12 gives suggested storage tank sizes for homes and apartments based on the number of families or baths.

HOT WATER HEATERS

Various types of heaters are available for supplying the hot water for domestic service in buildings. In any hot water supply system the water should be heated to a temperature between 150 and 180 F. Where the hot water requirements include supplies for kitchens, laundries or process work, the higher temperatures are used. In buildings where steam is available throughout the year, the hot water supply is usually taken from this source. In smaller domestic installations the fuel-burning device is generally automatically arranged so that hot water is supplied the entire year and not merely when the boiler is used for heating purposes.

Water is heated by various methods using heat exchangers arranged so that the boiler heating medium gives up its heat to the water in the hot water circulating system. These heat exchangers may be classified as follows:

1. Submerged steam heating coil in storage tank.
2. Submerged water heating coil in storage tank. (Fig. 8).
3. Indirect water heater, mounted on side of boiler below water line. (Fig. 9).
4. Submerged indirect water heater, placed in boiler below water line. (Fig. 10).

The efficiency of these heaters may be estimated as nearly 100 per cent as the heat loss from surface radiation of the heater and tank shell when covered with insulating material is generally reduced to a minimum. The capacities of these heaters are usually available from manufacturers

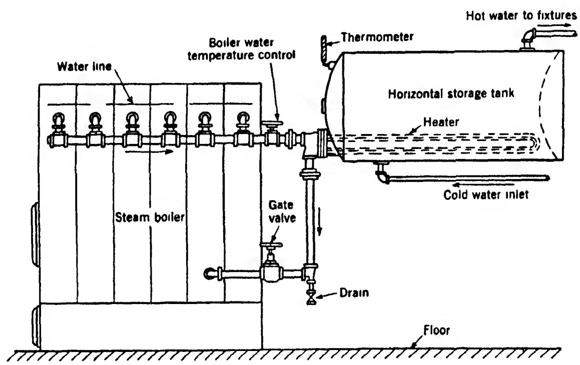


FIG. 8. HOT WATER HEATING COIL SUBMERGED IN STORAGE TANK

rating tables. The area of the inside surface of a heating coil may be determined from the following equation:

$$A = \frac{Q \times 8.33\,(t_o - t_i)}{K_o \times t_m} \tag{1}$$

where

A = surface area of coil, square feet.
Q = quantity of water heated, gallons per hour.
t_o = hot water outlet temperature, degrees Fahrenheit.
t_i = cold water inlet temperature, degrees Fahrenheit.
K_o = coefficient of heat transmission, Btu per hour per square foot surface.
For copper or brass coils K_o = 240 (steam) and 100 (hot water).
For iron coils K_o = 160 (steam) and 67 (hot water).
t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water temperature. t_m is approximately = $\left[t_s - \frac{(t_o + t_i)}{2}\right]$

Equation 1 may also be used for determining revised heating coil ratings under different temperature conditions as stated in the manufacturers ratings. When selecting a water heater, the conditions of operation should be carefully considered, as well as the location of the

storage tank and the piping arrangement between the boiler, heater and tank. It is generally good practice to allow a margin of safety when selecting an indirect heater of the proper size to provide for loss of efficiency due to the accumulation of scaling on the coils and piping. Heat exchangers classified according to (3) and (4) may be used with or without a storage tank, but when tanks are omitted, the indirect water heaters should be increased in size so as to heat the water instantaneously as it is needed.

The storage tank should be installed as high as possible. Horizontal tanks are preferable for all medium size installations and absolutely essential on larger installations. Where possible the storage tank should be installed with the bottom of the tank at or above the boiler water line. Horizontal storage tanks smaller than 18 or 20 in. diameter are not

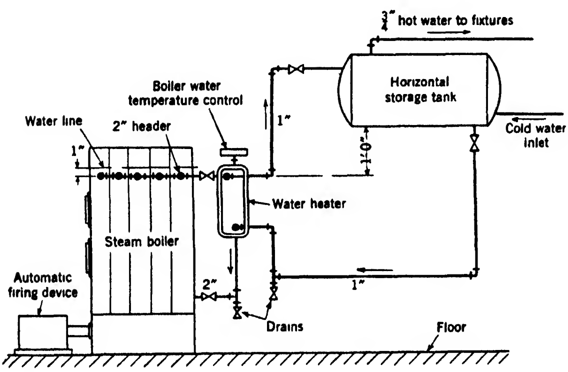


FIG. 9. INDIRECT WATER HEATER MOUNTED ON SIDE OF BOILER

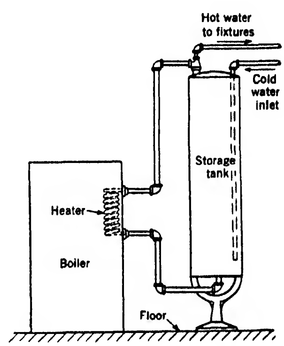


FIG. 10. INDIRECT WATER HEATER PLACED IN BOILER

recommended because of the difficulty of preventing the hot and cold water from mixing, and especially is this an important consideration when large quantities of water are withdrawn.

Pipe sizes between the water heater and boiler should be full size of the heater tappings (Table 12). When a heater is connected to a horizontal sectional boiler, it is recommended that connections be made to all sections and joined together a few inches below the water line as shown in Fig. 8, so that steaming is prevented in those sections which are not connected to the header.

When a steam coil is used for heating the water, an automatic thermostatic valve may be installed in the steam supply to the coil. The operation of this automatic valve is controlled by a thermostat located in the storage tank which permits the proper amount of steam to enter the coil so as to maintain an even water temperature.

An indirect water heater may be used on either a steam or hot water system, and generally this type of heater is provided with a temperature control device located in the boiler water circulating connection to the water heater. The setting on this thermostatic valve may be as low as

140 F or as high as 180 F and may be readily adjusted to meet particular requirements. With this type of control it is impossible to overheat the hot water supply which is an important safety consideration in some installations. This type of system may also be conveniently used during the non-heating season with the operation of the fuel burning device controlled by the water heater thermostat. (See Chapter 37). During the heating season the water heater temperature control functions as a low limit control.

When an indirect water heater is applied to a gravity hot water system, it is necessary to provide a valve in the supply to the heating system to prevent the flow of hot water from the boiler when heat is not required in the house. This valve may be controlled from a room thermostat and the automatic fuel-burning device controlled from the water heater thermostat. To prevent circulation in a forced hot water heating system flow control valves may be installed in the flow and return lines which act merely as check valves when the circulating pump is not operating. In this arrangement the pump is controlled by the room thermostat and the automatic fuel-burning device is controlled from the water heater thermostat.

STORAGE CAPACITY AND BOILER ALLOWANCES

The amount of storage provided in the hot water tank or heater is somewhat a matter of choice but is usually made ample to carry over the peak shortage which is likely to occur and is based on the assumption that only 75 per cent of the storage capacity will be available, as it has been found that if more than this amount is withdrawn from storage, the tank is so cooled down as to make the balance useless. The general rule may be cited that the less the heating capacity the greater must be the storage, and the greater the storage the less may be the heating capacity down to a point where the heating capacity will fail to be sufficient to heat up the tank storage during the periods of small load.

Example 5. A heater to supply 500 persons will have an average daily use of about 500×40 gal $= 20{,}000$ gal and this is an average of $\frac{20{,}000 \text{ gal}}{24} = 833$ gph but the peak hour will require $\frac{1}{10}$ of 20,000 = 2000 gal and the shortage during the peak hour, if the heating capacity is made to suit the average hourly use of 833 gal, will be $2000 - 833 = 1167$ gal so that the storage capacity, based on 75 per cent being available from this capacity without cooling the tank excessively, will be $\frac{1167}{0.75} = 1556$ gal.

Should it be desired to reduce the size of storage tanks and to use a greater heating capacity, it is only necessary to increase the heating capacity to say 1200 gph which then gives $2000 - 1200 = 800$ gal as the shortage during the peak hour, and the necessary storage will be $\frac{800 \text{ gal}}{0.75} = 1067$ gal; or the heating capacity can be increased to 1500 gal, leaving a shortage of $2000 - 1500 = 500$ gal.

Good design requires that the heating capacity be made as small as possible without introducing undesirable amounts of storage, as the heating capacity directly determines the load on the source of heat.

As indicated in Example 5, the heating load is proportional to the heating capacity and the boiler capacity must be increased for higher heating capacities and may be reduced for smaller heating capacities with greater storage. It may be assumed that a boiler capacity of about 4 sq ft of equivalent steam heating surface[3] (radiation) must be provided for every gallon of water heated 100 F or from 50 F to 150 F, which is the temperature rise most commonly assumed and required. On this basis it will be seen that the various conditions cited in Example 5 will require additional boiler capacity as follows:

Heating Capacity (Gph)	Additional Boiler Capacity (Sq Ft EDR)
833	3332
1200	4800
1500	6000

From this it is apparent that it is less costly to provide ample storage and to reduce boiler capacity than to diminish the storage and supply a greatly increased boiler capacity to compensate.

The boiler allowance value of 4 sq ft of equivalent steam radiation for each gallon of water heated through a temperature range of 100 F is based on an hourly heating rate. When reduced heating capacities are desired for economic reasons of boiler design and selection, engineers frequently recommend that the heating rate be extended over a period of two hours in which case the boiler allowance value would be reduced to 2 sq ft of equivalent steam radiation. Similarly any other heating rate may be established and a corresponding value of boiler allowance determined.

Reliable information based upon the installations of several heaters in existing heating systems indicates varying arbitrary values of boiler allowances to be used. When these values are selected for usage, a careful analysis of the varying factors involved in determining these values should be considered so that the proper heating allowances may be provided.

ESTIMATING HOT WATER DEMAND BY FIXTURES

In buildings where the occupancy is doubtful and only the number of plumbing fixtures can serve as a basis for determining the probable hot water demand, the problem is not so simple owing to the fact that a fixture gives no information as to how heavy a service may be demanded from the fixture and this amount of service is really the governing factor in making an estimate of the probable hot water demand. Table 13 may prove of some value in this respect as it gives the maximum assumed quantity of hot water per hour which will be demanded of any fixture and then gives a percentage of this amount which may be assumed as probable in different types of buildings. Table 14 gives approximate hot water requirements in various types of buildings.

Example 6. Let it be assumed that an apartment house with 20 apartments has 20 baths, 20 lavatories, 20 kitchen sinks and 20 laundry trays; what is the probable maximum hourly demand for hot water?

---

[3]Actual requirement for 100-deg temperature difference $= \frac{100 \times 8.33}{240} = 3.48$ sq ft per gallon of water heated

20 Baths at 40 gal and 33 per cent	270 gal
20 Lavs. at 20 gal and 25 per cent	100 gal
20 Sinks at 30 gal and 33 per cent	200 gal
20 Trays at 50 gal and 60 per cent	600 gal
Total	1170 gal
Probable peak use at one time	35 per cent
Probable actual peak demand	409 gph

If three persons are assumed to an apartment the total daily use of hot water should approximate 20 × 3 × 40 gal = 2400 gal and if the peak hour is 10 per cent of this amount, the peak hour by this method shows a probable demand of one-tenth of 2400 gal, which indicates that the values in Table 13 are safe.

SWIMMING POOL HEATING REQUIREMENTS

Swimming pools present a problem of hot water heating demand which is frequently overestimated. Few outdoor swimming pools require water heating, and in some cases they require the addition of cold water to regulate the temperature. The recirculation system of a swimming pool consists of the pumps, hair and lint catchers, and filters together with all necessary pipe connections to the inlets and outlets of the pool. The water heater, the sterilizing equipment and suction cleaner are usually installed or connected to the recirculation system and may be considered as integral parts of the system.

The recirculation system and all its component parts should be designed to provide the required volume of circulation so that the water turnover ratio is at least two times per day and where heavy loads are anticipated the turnover ratio should be increased to three times or more. Many states have regulations prescribing the circulation turnover.

The water heaters for swimming pools are usually instantaneous steam

TABLE 13 ORDINARY MAXIMUM HOURLY DEMAND FOR HOT WATER FOR VARIOUS FIXTURES IN GALLONS AND PROBABLE PERCENTAGE OF USAGE

TYPE OF BUILDING	LAVATORIES		BATHS	SHOWERS	SLOP SINKS	KITCHEN SINKS	PANTRY SINKS	FOOT BATHS	WASH TRAYS	AV MAX USE[a]
	Private	Public								
MAXIMUM PROBABLE USAGE GPH	20	20	40	300	30	30	20	20	50	
	Probable Usage in Per Cent of Maximum Ordinary Use									
Apt. house	25	50	33	67	67	33	50	25	60	35
Club	25	75	50	67	67	67	100	25	80	60
Gym.	25	100	100	100	......	....	......	100	......	80
Hospital	25	75	50	33	67	67	100	25	80	45
Hotel	25	100	50	33	100	67	100	25	80	70
Industrial	25	150	100	100	67	67	......	100	......	90
Laundries	25	100	......	......	33	....	......	......	100	100
Office building	25	75	......	......	50	....	......	......	......	20
Baths	25	150	150	100	50	....	......	......	......	100
Residences	25	......	50	33	50	33	50	50	60	50
Schools	25	75	......	100	67	33	100	50	......	25
Y. M. C. A.	25	100	100	100	67	67	100	100	80	75

[a]Percentage of fixtures likely to be demanding maximum probable usage at any one time

coil heaters. These heaters should be sized so that they will have sufficient capacity to heat the water delivered by the circulating pump 15 F per hour.

The water temperature in a pool is usually maintained at about 72 F. A few states have regulations prohibiting higher water temperatures than 70 F. The room temperature should be approximately 5 F higher, but not more than 8 F higher nor less than 2 F lower, than the water temperature.

Example 6. Assume a swimming pool 75 ft long, 30 ft wide with an average depth of 6 ft. If the water is to be heated from a temperature of 50 to 65 F, what capacity heater and steam consumption is required with a turnover ratio of two times per day?

Pool volume: $75 \times 30 \times 6 \times 7.5 = 100{,}000$ gal.

With a turnover ratio of twice in 24 hr, the heating capacity is: $\dfrac{100{,}000 \times 2}{24} = 8333$ gal per hour.

The steam consumption would be: $\dfrac{8333 \times 8.33\ (65 - 50)}{970} = 1080$ lb steam per hour.

Regulation of swimming pool temperatures is essential for successful operation and economy. It is therefore recommended that the steam supply to the heater be provided with a by-pass which may be used for pool filling and initial heating and that a smaller by-pass be installed with an automatic control valve having the capacity to heat the circulation water approximately 5 F per hour.

TABLE 14. HOT WATER CONSUMPTION IN VARIOUS TYPES OF BUILDINGS FOR DIFFERENT PURPOSES

TYPE OF BUILDING	CONDITIONS	GALLONS
Hotels	Room with basin only	10 (per day)
	Room with bath	
	(Transient)	40 (per day)
	(Men)	40 (per day)
	(Mixed)	60 (per day)
	(Women)	80 (per day)
	Two-room suite and bath	80 (per day)
	Three-room suite and bath	100 (per day)
Public Buildings	Public bath or lavatory	150 (per day per fixture)
	Public shower	200 (per day per fixture)
	Public lavatory with attendant	200 (per day per fixture)
Industrial Buildings	Per office employee	2 (per day)
	Per factory employee	5 (per day)
	Cleaning floors	3 (per 1000 sq ft per day)
Restaurants	$0.50 Meals	0.5 (per customer with hand washing)
		1 0 (per customer with machine washing)
	$1.00 Meals	1.0 (per customer with hand washing)
		2.0 (per customer with machine washing)
	$1 50 Meals	1.5 (per customer with hand washing)
		4.0 (per customer with machine washing)

PROBLEMS IN PRACTICE

1 ● The heating capacity of an indirect water heater is 100 gal per hour, using steam at 215 F and raising the water from a temperature of 50 to 150 F. Determine the heating capacity of the same water heater using water at a temperature of 180 F for the heating medium.

Using Equation 1, and because the surface area of the water heater is the same for each condition, the two conditions may be equated as follows:

$$\frac{100 \times 8.33\,(150 - 50)}{240\left[215 - \frac{(150 + 50)}{2}\right]} = \frac{Q \times 8.33\,(150 - 50)}{100\left[180 - \frac{(150 + 50)}{2}\right]}$$

Q = 28.98 gal per hour, capacity of heater using water at a temperature of 180 F.

2 ● Why is it impractical to size water supply piping so pipe friction will produce an equal pressure on each fixture?

Because the friction would be built up only in periods of maximum flow and at all other times it would be only a fraction of that required.

3 ● What is the purpose of zoning water supply systems in tall buildings?

To avoid excessive pressures in the lower stories.

4 ● Define the maximum possible flow, the maximum probable flow, and the average probable flow.

The maximum possible flow is the flow which would occur if all of the outlets on the system were opened at one and the same time. The maximum probable flow is the flow which will occur with probable peak conditions. The average probable flow is the flow likely to occur under a normal condition of use.

5 ● What is the factor of usage?

This is the percentage of the maximum possible flow which is likely to occur at peak load.

6 ● How many feet higher than the uppermost fixtures must the water line in a house tank be to provide about 15 lb per square inch pressure at the fixture outlet?

Allowing for pipe losses, about 45 ft.

7 ● What methods of hot water circulation commonly are employed with hot water supply systems?

a. Up-feed risers with returns having no connections paralleling the risers.

b. Up-feed risers with returns in other locations, and with connections taken off both supply and return.

c. One main up-feed riser, without connections, supplying all down-feed risers for all fixtures.

8 ● Which method of hot water supply generally is the most satisfactory?

The single main up-feed riser supplying drop risers for all fixtures.

9 ● How much of the water stored in a hot water storage tank really is available for use?

About 75 per cent, because when only 25 per cent of the original water remains in the tank it has been so cooled down by the entering water that it is too cold for satisfactory use.

10 ● In cases of intermittent demand, does a large hot water storage tank increase or decrease the steam load for water heating?

It decreases the steam load in cases of intermittent demand but causes no change in the steam load if the demand is constant.

Chapter 44

TEST METHODS AND INSTRUMENTS

Pressure Measurement, Temperature Measurement, Air Movement, Humidity Measurement, Carbon Dioxide Determination, Dust Determination, Flue Gas Analysis, Measurement of Smoke Density, Heat Transmission, Eupatheoscope

SEVERAL types of measuring apparatus are available for accurately determining the thermal capacity and air movement of gaseous vapors and homogeneous materials. This chapter gives a brief description of the principal instruments used in connection with the proper control and testing of heating and air conditioning installations.

TEST METHODS

The SOCIETY has adopted standard test methods or codes for testing and rating most heating, ventilating and air conditioning equipment. A list of the titles of these test codes may be referred to on pages 854 and 855. Many of the test instruments required are specified and described in these codes.

PRESSURE MEASUREMENT

Atmospheric pressure is usually measured by a *mercurial barometer* which, in its simplest form, consists of a glass tube about 3 ft long, closed at the upper end, filled with mercury and inverted in a shallow bath of mercury. The pressure of the atmosphere on the exposed top of the mercury in the cistern supports a column of mercury in the tube to a height of about 30 in. Readings are taken of the height of the column between the levels of mercury in the tube and in the cistern. Atmospheric pressure is the same as the pressure exerted by this supported column of mercury, and, in pounds per square inch, is equal to its height in inches times 0.491, which is the weight in pounds of 1 cu in. of mercury at 32 F. At latitude 45 deg and sea level, and at a temperature of 32 F, the atmosphere will support a column of mercury 29.921 in. in height. The pressure of 14.7 lb per square inch, derived by multiplying 29.921 by 0.491, is called *standard* or *normal barometric pressure*. Since the height of the barometer depends on the density of the mercury as well as on the pressure of the atmosphere, and since the density is dependent on the temperature, mercurial barometer readings should always be corrected for temperature.

The following equation may be used to make corrections for temperature

$$h = h_1 [1 - 0.000101 (t_1 - t)] \quad (1$$

where

h = height of mercury column corrected to temperature t, inches.
h_1 = actual height of mercury column, inches.
t_1 = actual temperature of mercury column, degrees Fahrenheit.
t = temperature to which column is to be corrected, degrees Fahrenheit.

Atmospheric pressure may also be measured by means of an *aneroid* barometer. In this instrument atmospheric pressure is made to move an indicating pointer either by bending the thin corrugated top of a partially exhausted metallic box, or by distorting a bent, thin-walled metal tube. The aneroid barometer contains no liquids, is portable but is less accurate than the mercurial barometer.

Pressures above or below atmospheric are usually measured by means of gages which indicate the difference between the pressure being measured and atmospheric pressure at the same time and place. A gage which indicates pressures higher than atmospheric is known as a *pressure gage*, and a gage which indicates pressures lower than atmospheric is known as a *vacuum gage*. The most common type of these gages contains a flexible hollow metal tube of oval cross section, known as a *Bourdon tube*. When subjected to unequal inside and outside pressures, this tube tends to straighten out, and a pointer motivated by this straightening indicates the pressure difference on a suitable graduated scale.

High vacuum readings such as are encountered in condenser and steam jet refrigeration practice are commonly obtained by the use of mercury column vacuum gages. When the readings obtained with the mercurial barometer and those with the mercury vacuum gage have both been corrected to 32 F, the difference in the two readings will give the absolute vacuum in inches of mercury. Equation 1 may be used to make corrections for temperature.

In the measurement of small pressure differences, the *U* tube in one of its many forms is convenient, inexpensive and it may be built for any desired degree of accuracy. *U* tube manometers may be fabricated from glass and rubber tubing or any of the numerous commercial forms may be used.

A gage which indicates pressures slightly above or below atmospheric is known as a *draft gage*. It is essentially a *U* tube containing either water, kerosene, alcohol, or mercury, with one leg exposed to the air and the other connected to a point where the pressure is to be determined. When the pressure being read is equal to atmospheric, the level of the liquid in the legs will be the same, indicating a zero gage pressure. When a pressure is applied to one leg, one side will fall and the other will rise an equal amount. The difference in height between the two liquid levels indicates the pressure expressed in inches of liquid used in the gage.

Various forms of high sensitivity draft gages[1] frequently called micromanometers[2] are available for the measurement of small pressure differen-

[1]Fluid Velocity and Pressure, by J. R. Pannell (Edward Arnold and Co., London, 1924).
[2]Illinois Micromanometer, University of Illinois (*Engineering Experiment Station Bulletin* No 120, p 91)

tials and may be sensitive to pressures as small as 0.001 in. of water. These gages are often useful where measurements are to be made on pressure differentials less than 0.1 in. of water, although their total range may extend as high as 5 to 10 in. of water.

TEMPERATURE MEASUREMENT

In engineering work, thermometers are largely employed to measure the intensity of heat. Those most commonly used are *liquid-in-glass* thermometers. Mercury and alcohol are the liquids most frequently used.

Mercurial thermometers depend on the uniform expansion of mercury to indicate changes in temperature. An amount of mercury held in a sealed tube with a bulb at one end will rise to one definite level when immersed in melting ice, and to another definite level when immersed in boiling water. These two points are marked, and the space between them is divided into a number of equal portions, each of which is called a degree. In the Fahrenheit scale, there are 180 deg thus obtained, while the centigrade scale has 100 and the Reâumur has 80. Like divisions are marked off on the column above and below these two determined points in order that a greater range of temperature may be read.

Mercurial thermometers may be used in a temperature range from -40 to $+932$ F.

Alcohol thermometers are similar in construction to mercurial thermometers but are useful in a lower temperature range (-94 to $+248$ F).

Industrial thermometers in a large number of designs are available, but for test purposes etched stem thermometers are most frequently used. The etched stem thermometer has greater sensitivity and less lag than most industrial thermometers.

For precision temperature measurements, it is necessary to correct the thermometer reading for emergence of the stem if any part of the mercury column is exposed to a temperature other than that being measured (unless the thermometer has been calibrated under like conditions). The emergent stem correction may be calculated by the following equation:

$$K = 0.00009\, D\, (t_1 - t_2). \tag{2}$$

where

K = correction to be added, degrees Fahrenheit.

D = length of emergent stem, degrees Fahrenheit on thermometer stem.

t_1 = temperature indicated on the thermometer, degrees Fahrenheit.

t_2 = temperature of exposed mercury stem, degrees Fahrenheit.

Thermocouples[3] may be used to measure any range of temperatures up to 2900 F. When two dissimilar metals are joined at two points and a temperature difference exists between these junctions, an electromotive force will be developed. Its magnitude depends on the character of the metals and the difference in temperature between the junctions. A potentiometer or sensitive galvanometer of high resistance connected to the thermocouple will give a deflection which is a function of the temperature difference between the hot and cold junctions. Thermocouples con-

[3]A S H V E RESEARCH REPORT No 943—Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, by A. P Kratz and E L. Broderick (A S H V E TRANSACTIONS, Vol. 39, 1933, p 55)

nected in series are called *thermopiles.* Thermocouples for the measurement of high temperatures are calibrated with the aid of the known melting points of pure metals.

Resistance thermometers are suitable for temperature measurements up to 1800 F. These thermometers depend for their operation on the change of resistance with temperature of a platinum, nickel, or copper wire coil, and they are calibrated in the same way as thermocouples.

Pyrometers of various types may be used for temperatures above 500 F. The *mercurial pyrometer* is a thermometer with an inert gas, such as nitrogen or carbon dioxide, above the mercury column to prevent the mercury from boiling. The *radiation pyrometer* consists of a thermopile upon which the radiation from a hot source is focused by a concave mirror or lens. A sensitive galvanometer or potentiometer with a calibrated temperature scale indicates the thermo-electromotive force created by the heat on the thermopile. The *optical pyrometer* measures radiant energy by comparing the intensity of a narrow spectral band, usually red light emitted by the object, with that emitted by a standard light source (electric lamp). *Thermo-electric pyrometers* operate on the same principle as thermocouples. When measuring high temperatures, it is customary to hold the cold junction at room temperature and this may cause some error if the room temperature is above or below the calibration point. For extremely precise temperature measurements, the cold junction is usually immersed in melting ice to fix the cold junction temperature. Various forms of hand-operated and automatic cold junction temperature compensators are also available.

In the measuring of room temperature care must be exercised to prevent the results from being affected by the body heat of the observer, by air currents from doors, windows and other openings, or by radiant heat from some local source such as a radiator or wall. All glass thermometers should be mercury thermometers with engraved stems. The total graduations of the thermometers should be from 20 to 120 F, in one degree graduations. No ten degrees should occupy a space of less than one-half inch. The accuracy throughout the whole scale must be within one-half degree. The operator should take hold of the top and no part of the body, including the hand, should be nearer than 10 in. to the bulb. The thermometer should not be closer than 5 ft to any door, window, or other opening; should not be closer than 12 in. to any wall; and should be between 3 and 5 ft from the floor. A sling instrument should be used for extreme accuracy. Thermocouples or resistance thermometers may also be used for room temperature measurements, an advantage being that the operator can read temperatures from outside the room if desired, and thus eliminate the errors which might be caused by his presence close to the temperature measuring device.

For measuring duct temperatures a duct thermometer should be used, with the bulb extending into the duct at least 6 in. When the thermometer is to be permanently located in the duct, a pipe flange or nipple should be used to receive the threaded portion of the thermometer stem. When the thermometer is not to be permanently located, a cork or rubber stopper may be placed around the stem to prevent errors from air leakage. Readings should be taken at various locations in a duct so due consideration may be given to temperature stratification. Other forms of

temperature measuring devices may be used, but the active part must be at least 6 in. from the duct wall.

Recording instruments may be used for testing and for making continuous records of operation. Potentiometer and Wheatstone bridge recorders for thermocouples and resistance thermometers respectively may have accuracies of ± 1/3 per cent of their range, or, for example, to ± 1 F in a range of 0 to 300 F. This accuracy compares favorably with that of other forms of temperature measuring devices.

AIR MOVEMENT MEASUREMENT

The quantity, velocity and pressure of air moved by a fan or flowing through a duct or grille may be determined by various methods. The instruments in common use are the Pitot tube, anemometer, direct reading velocity meter, and Kata-thermometer, the latter being suitable for low air velocities and being commonly used for measurements at points where the air is not confined in a duct. Electrical anemometers are also available, operating on the principle of measurement of the variation of resistance of a hot wire cooled to various degrees by air velocities past the wire. The use of calibrated nozzles, orifice plates, and Venturi meters are recognized methods, which, however, have little application in connection with ventilation practice.

Pitot Tube

This usually consists of two tubes, one within the other, which when properly held in the air stream will register the total or impact pressure and the static pressure, respectively. If these tubes are connected to opposite sides of a draft gage, or other type of *U* tube, the recorded pressure will be the differential or velocity pressure. Volume measurements may thus be made in a duct of known area. Pitot tube measurements are preferably used for air velocities exceeding 20 fps. Volumetric determinations from Pitot tube readings should take into account the barometric pressure and the temperature and humidity of the air measured.

Air flow in ventilation practice is generally in the turbulent range. When stratification of velocity, vortex motion, or violent eddy currents of air in ducts exist, accurate velocity pressure measurements are difficult To insure accuracy a straight section of duct from 5 to 10 times its own diameter is desirable in order to straighten out the air currents. If it is necessary to take Pitot tube readings in shorter sections of straight duct, the results must be considered subject to some doubt and checked accordingly. For accurate work it is necessary to make a traverse of the duct, dividing its cross-section into a number of imaginary equal areas and taking a reading in the center of each, the average of the velocities corresponding to these pressures giving the true velocity in the duct.

A pitot tube of standard design and the traverse method of obtaining average velocity are completely described in the A.S.H.V.E. Standard Test Code for Centrifugal and Axial Fans.[4]

For precise work the shape, size and calibration of the Pitot tube are important considerations in the determination of the correct air flow.

[4]A.S H V.E TRANSACTIONS, Vol 29, 1923, p 407. Amended June, 1931 Also see Standard Test Code for Centrifugal and Axial Fans, Edition of 1938.

Extensive test results comparing the characteristics of several Pitot tube types are available in the published reports[5] of the government.

Anemometer

The vane-type anemometer is most frequently used for test work. It consists of a small, delicate, fan-like rotor connected to a revolution counter. The instrument is held in the air stream where the velocity is to be measured. It is calibrated to read directly in linear feet. The velocity in feet per minute is obtained by dividing the reading (linear feet) by the elapsed time, in minutes.

The vane anemometer is delicate, requires frequent calibration and is suited only to low velocities (less than 3000 fpm). The vanes of the instrument should never be touched.

The following procedure for obtaining anemometer readings is based on research conducted at Armour Institute of Technology in cooperation with the A.S.H.V.E. Research Laboratory[6].

Supply Grilles. The surface of the grille should be marked off into a number of equal areas approximately 6 in. square. A 4-in. anemometer should be used and should be held at the center of each section in contact with the grille (or as close as possible) for a period of time sufficient to insure an average reading. In the case of supply grilles, the instrument should always be held with the dial facing the operator. The average of the corrected readings should then be used in the following formula to obtain the flow in cubic feet per minute:

$$cfm = CV\frac{A + a}{2} \text{ or } \frac{CVA\,(1 + p)}{2} \tag{3}$$

where

V = average of corrected anemometer readings, feet per minute.
A = gross area of grille, square feet.
a = net free area of grille, square feet.
p = percentage of free area of grille expressed as a decimal.
C = a coefficient that varies with the velocity from grille and may vary slightly with type of grille. For average use, with supply grilles, C can be taken as 0.97 at velocities from 150 to 600 fpm, and as 1.00 at higher velocities.

Particular care should be exercised in the case of long, narrow grilles. The nature of the approach sometimes results in there being a narrow strip along the top or bottom of the grille through which no air will be flowing. This may be detected by holding the anemometer completely out of the air stream and then moving it slowly inward over the grille until the vanes just start to move. The distance which the vanes extend over the grille opening at this moment will indicate the width of the dead strip. Only the remaining portion of the grille should be considered in making the calculations for gross and free area.

Exhaust Grilles. The surface of the grille should be marked off and readings taken in the same manner as with supply grilles, except that the instrument should be held with the dial facing the grille, and in contact with it. The traverse should be taken at a uniform rate, allowing suf-

[5]Technical Notes No. 546, *National Advisory Committee for Aeronautics*, November, 1935.

[6]A.S.H.V.E. RESEARCH REPORT No. 966—Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies (A.S.H.V.E. TRANSACTIONS, Vol 36, 1930, p. 201, Vol 37, 1931, p. 619, and Vol. 39, 1933, p. 373).

ficient time in each space to minimize the percentage of error. In the case of exhaust grilles it is found that the formula:

$$cfm = KVA \tag{4}$$

in which

V = average indicated velocity obtained by the anemometer traverse.

A = gross area of grille, square feet.

K = coefficient determined by experiment. For average use, with exhaust grilles, K may be taken as 0.873 for all usual velocities[7].

This formula is of advantage, especially with ornamental grilles, in that the free area need not be measured.

The flow of air through registers and grilles is of considerable importance, being frequently the only convenient method of measuring the volume of supply air to a room. While duct measurements, if available, are more dependable, grille measurements provide a fairly accurate method, if care is taken in the technique of using the anemometer.

Direct Reading Velocity Meter

An instantaneous direct reading air velocity instrument available in a portable case is used for recording air movement on a calibrated scale. Air entering the meter actuates a vane movement to which is attached a pointer with control hair springs and a magnetic damping arrangement.

Velocity meters are available in either orifice, shutter or tube types. The orifice unit is used where the instrument can be placed in the air stream when obtaining a reading such as in rooms or large spaces or at unrestricted outlets of ducts. The use of the shutter type is similar to the orifice style except that it has means for changing the scale range. The shutter is adjusted so that the large ports are fully open for low velocity readings. For high range readings the shutter is turned until the large openings are closed and only a small port is open. The shutter is omitted in the tube type of meter and in place of this fitting a tube attachment is threaded to the case. A flexible rubber tube and specially designed metal jets are used for obtaining high range readings. Jets may be secured for unusual applications such as in obscure locations, surging air currents or leakage from ducts and similar requirements. Due to the connecting tube flexibility, the jet can be moved as required while the instrument is held stationary.

Where it is desired to obtain air velocity readings within a duct, special jet and additional meter fittings are used which indicate directly the true air velocity with no corrections being essential for static pressure conditions. Air enters the meter through one side of the jet and is discharged back into the duct through the other side of the jet.

Kata-Thermometer

The Kata-thermometer can be used to determine air velocities provided the walls and surrounding objects are at or near the room temperature. Especially at low velocities it constitutes a useful instrument for readily detecting drafts.

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[7]A.S.H.V E RESEARCH PAPER—The Flow of Air Through Exhaust Grilles, by A. M Greene, Jr. and M. H Dean (A S H V.E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1938, p. 619).

The instrument is essentially an alcohol thermometer with a bulb approximately ⅝ in. in diameter and ½ in. long with a stem 8 in. long reading from 100 F to 95 F, graduated to tenths of a degree. To take readings the bulb is heated in water until the alcohol expands and rises into a top reservoir. The time in seconds required for the liquid to fall from 100 F to 95 F is recorded with a stop watch and this time is a measure of the rate of cooling.

The dry Kata loses its heat by radiation and by convection so for constant velocities the time of cooling is a function of the dry-bulb temperature of the surrounding air. The wet Kata, which has a cloth covering fitted snugly around its bulb, loses heat by radiation, convection, and evaporation, and for constant velocities its rate of cooling is a function of the wet-bulb temperature of the air irrespective of the dry-bulb temperature or relative humidity. It does not follow, however, that the difference in rate of cooling of the dry and the wet Kata is caused by evaporation. A change in the wet-bulb temperature produces a change in the surface temperature of the wet Kata which in turn affects the heat lost by radiation and by convection.

Several precautions should be taken to obtain the best results with this instrument:

1. To obtain velocity readings use the dry Kata since the error in timing is reduced.
2. The instrument should be heated and allowed to cool two or three times before recording the final time of cooling. The first reading is not reliable.
3 All traces of moisture must be removed from the dry Kata before timing to eliminate error introduced by evaporation.
4. Use only the formula applying to a particular instrument. Each Kata receives an individual calibration.

HUMIDITY MEASUREMENT

The sling psychrometer is the recognized standard instrument for determining humidities. In order to obtain accurate readings considerable skill is required on the part of the operator. The wicking and water must be clean and the temperature of the water should be slightly above the wet-bulb temperature of the surrounding air. The psychrometer should be swung rapidly and several and frequent observations should be made to see that the wet-bulb temperature has become stationary before the final reading is noted. Care should be taken that the wet-bulb has reached a minimum temperature, but the wick must still be moist. Standard psychrometric tables should be used[8].

In making wet-bulb measurements below 32 F the same procedure is followed as above 32 F. The water is liquid at the start, but as the sling is operated it will freeze rapidly enough so that in quickly giving up the latent heat of fusion, the indicated wet-bulb temperature may drop below the actual wet-bulb temperature. After the liquid on the bulb has become thoroughly frozen the wet-bulb temperature will rise to normal. A very thin film of ice is more desirable than a thick film. Care must be taken to read the temperatures in the region below 32 F accurately because the spread between the wet- and dry-bulb is small.

[8]Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew-Point; *U. S. Department of Agriculture*, Weather Bureau, Washington, D C

In taking humidity readings in ducts it is usually impracticable to use a sling psychrometer. For this work the stationary hygrodeik arranged for bolting on to the side of the duct, with two bulbs extending into the duct, will be found very convenient. Owing to the velocity of the air passing over the bulbs within the duct an accurate reading will be secured, corresponding to that given by the sling psychrometer.

Various forms of humidity recorders are available, some merely recording wet- and dry-bulb temperatures, and others recording relative humidity directly. Any form of wet- and dry-bulb device must have sufficient air velocity over the thermometer bulbs to insure accurate readings; this velocity should be secured by a fan if the air is not itself in motion. A minimum velocity of 900 fpm is usually recommended but velocities from 300 to 1000 fpm have been found suitable under favorable conditions[9]. For extremely low humidities, or for humidity measurements above 212 F, a thermal conductivity method is available[10].

CARBON DIOXIDE DETERMINATION[11]

At ordinary concentrations carbon dioxide is not harmful. The amount of carbon dioxide in the air is a convenient index of the rate of air supply, and of the distribution of the air within rooms. Unequal carbon dioxide concentrations in parts of a room indicate improper air distribution.

The Petterson-Palmquist apparatus has been generally accepted as the standard device for the determination of carbon dioxide in air investigations. The principle involved is the measurement of a given volume of air, the absorption of the contained carbon dioxide in a caustic potash solution, and the remeasurement of the volume of air at the original pressure in a finely graduated capillary tube, the difference in volume representing the absorbed carbon dioxide. (See Report of Committee on Standard Methods for Examination of Air, *American Public Health Association*, Vol. 7, No. 1; *American Journal of Public Health*, Jan., 1917.)

A thermal conductivity method may also be used to measure carbon dioxide in air over a range of 0 to 1.5 per cent[12].

Where field conditions are such that this apparatus may not be conveniently used, as in street cars, air samples may be collected in clean bottles having mercury-sealed rubber stoppers, and these may be subjected to laboratory analysis.

DUST DETERMINATION

Many laboratory methods have been developed to measure the dust in the air. These involve the collection of dust on sticky plates, on filter paper, in water, on porous crucibles, or by electric precipitation, and the subsequent determination of the amount of dust by microscopic counting, weighing, or titration. While there is no standard method, the Hill

[9]Discussion, W. H Carrier and C O Mackey (*A S M E. Transactions*, Vol. 59, No 6, August, 1937, pp 528-30)

[10]Gas Analysis by Measurement of Thermal Conductivity, by H A. Daynes (*Cambridge Press*, 1933)

[11]A S H V.E RESEARCH REPORT No 959—Indices of Air Changes and Air Distribution, by F C Houghten and J L. Blackshaw (A S H V.E. TRANSACTIONS, Vol 39, 1933, p 261).

[12]Loc Cit Note 10

dust-counter, using a microscope, the impinger[13], using chemical changes in water, and the Lewis sampling tube[14], involving the analytical weighing of a porous crucible, are accepted. All test results should be accompanied by the name of the instrument used as great variation in counts with the different instruments will be obtained. The SOCIETY has developed a code[15] for the testing and rating of air cleaning devices used in general ventilation work.

FLUE GAS ANALYSIS

The analysis of flue gases by chemical means is made with the *Orsat apparatus*. A solution of KOH is used to absorb the CO_2. Free oxygen is absorbed by a mixture of pyrogallic acid and KOH. The solution for absorbing the CO is cuprous chloride. The apparatus consists of a burette surrounded by a water jacket, to receive and measure the volume

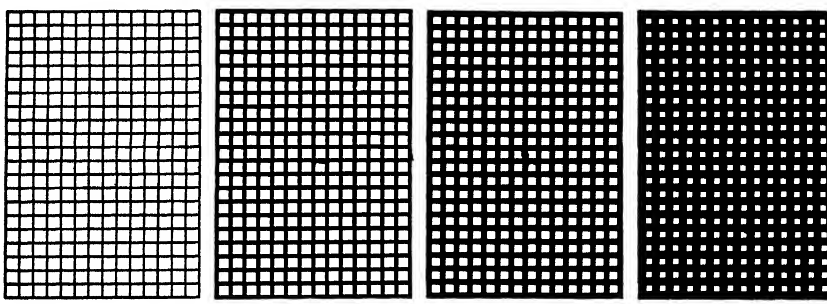

FIG. 1. RINGELMANN SMOKE CHART

of gas. The burette is connected by a manifold of glass to *pipettes* containing liquids for absorbing CO_2, O_2 and CO.

Various forms of automatic indicating and recording gas analysis devices are available, operating on either chemical or physical principles. Such devices are convenient for plant operation.

MEASUREMENT OF SMOKE DENSITY

Relative smoke density is usually measured by comparison with the Ringelmann Chart (Fig. 1). In making observations of the smoke issuing from a chimney, four cards ruled like those in Fig. 1, together with a card printed in solid black and another left entirely white, are placed in a horizontal row and hung at a point 50 ft from the observer and conveniently in line with the chimney. At this distance, the lines become invisible, and the cards appear to be of different shades of gray, ranging from very light gray to almost black. The observer glances from the smoke coming from the chimney to the cards, which are numbered from 0 to 5, determines which card most nearly corresponds with the color of

[13]Public Health Bulletin, No. 144, 1925, *U. S. Public Health Service*.

[14]Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by Samuel R Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277)

[15]A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A.S H.V.E. TRANSACTIONS, Vol. 39, 1933, p 225)

the smoke, and makes a record accordingly, noting the time. Observations are made continuously during one minute, and the estimated average density during that minute recorded. The average of all the records made during a boiler test is taken as the average figure for the smoke density during the test, and the entire record is plotted on cross-section paper in order to show how the smoke varied in density from time to time.

Smoke recorders are available which give a much more accurate indication of the amount of smoke being produced than does the Ringelmann Chart. They all depend upon projecting a beam of light through the smoke flue or through a separate compartment from which a sample of the flue gas is drawn continuously. The light of the beam which passes through without being absorbed by the smoke is measured to determine the smoke density. Most of these instruments make use of a photo-electric cell or a thermopile to measure the relative amount of light which has not been absorbed. Standard electrical instruments serve for indicating or recording.

MEASUREMENT OF RATE OF HEAT TRANSMISSION

The standard methods of testing built-up wall sections are by means of the *guarded hot-box*[16] and the *guarded hot-plate*[17]. The *Nicholls heat-flow meter* may be used for testing actual walls of buildings.

It would be obviously impossible to determine the air-to-air heat transmission coefficients of every type of wall construction in use with the heat-flow meter, the guarded hot-box or the guarded hot-plate on account of the great amount of time involved. Hence, the method of computing the coefficients from the fundamental constants must be resorted to in most cases. The guarded hot-plate is used to determine the fundamental constants. The heat-flow meter, guarded hot-box and guarded hot-plate tests can be used to good advantage in checking the accuracy of the computed values.

If the hot-box or hot-plate methods are used, tests are usually run under still air conditions, which means there is no wind movement over the surfaces of the wall during the test. In the hot-plate method of test the inside surface coefficient is eliminated by the plates being in direct contact with the wall. In practice, some wind movement over the exterior surface of the wall should always be allowed for; hence, still-air coefficients cannot be used over the outside of the building during the heating season. Moreover, still-air transmission coefficients cannot be corrected to provide for moving air conditions by applying a single constant factor. Computed coefficients of transmission for various types of construction are given in Chapter 5.

EUPATHEOSCOPE

The eupatheoscope affords a means of evaluating the combined effect of radiation and convection in a given environment in terms of a standard environment and in some terms related to human comfort. (See Chapter 41).

[16]Standard Code for Heat Transmission through Walls (A S H V E TRANSACTIONS, Vol 34, 1928, p 253), and Report of the Committee on Heat Transmission, *National Research Council*

[17]A S H V E. RESEARCH REPORT No 685—Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A S H V E TRANSACTIONS, Vol 30, 1924 p 65)

PROBLEMS IN PRACTICE

1 ● The hand on a pressure gage attached to a steam line indicates a pressure of 15 lb per square inch and the barometric pressure is 14.7 lb per square inch. What is the absolute pressure, in pounds per square inch, being exerted by the steam?

The absolute pressure exerted by the steam in the pipe is equal to the pressure indicated by the gage plus that exerted by the atmosphere.

Total pressure = 15 + 14.7 = 29.7 lb per square inch.

2 ● What is the corrected barometric pressure of the atmosphere at 32 F when a mercurial barometer reading of 29.51 in. Hg, is determined in a room having a temperature of 91 F?

Substitute in Equation 1. $h = 29.51\ [1 - 0.000101\ (91 - 32)]$.

$h = 29.33$ in. Hg.

3 ● Outline the procedure to be followed in taking room temperatures.

In taking room temperatures, a standard mercury thermometer should be used, with care taken that no part of the observer's body is nearer than 10 in. to the thermometer bulb. The thermometer should be held at least 5 ft away from any window, door or opening; it should be at least 12 in. away from any wall, and should be between 3 and 5 ft from the floor.

4 ● What advantages other than its sensitiveness, has the U tube draft gage or manometer for measurement of low pressures?

Inherent accuracy without calibration and low cost of the essential parts, which are glass tubing and an ordinary scale.

5 ● Are thermocouples as accurate as mercury thermometers?

Within the range which can be measured with both instruments (below 1000 F) either one may be made as sensitive as the service requires. The accuracy of a thermocouple temperature measurement depends chiefly on: (1) an accurate calibration of the wire, (2) the sensitiveness of the electrical instrument, (3) accurate cold-junction control, and (4) proper placement of the sensitive junction.

6 ● When an anemometer is used for measuring the air discharged from a grille or register, does it read the velocity through the gross face area or the velocity through the net free area?

Neither. If either of these velocities is required, it should be calculated by means of Equation 3.

7 ● Do common errors made in humidity determination produce a result that is too high or too low?

A higher relative humidity than the true value is likely to be found, either because there is insufficient velocity over the wet-bulb or because the reading is not taken at the right time.

8 ● What is the purpose of the carbon dioxide determination?

It is an index of the adequacy of fresh air supply and also an indicator of air distribution

Chapter 45

TERMINOLOGY

Glossary of Physical and Heating, Ventilating and Air Conditioning Terms Used in the Text, Standard Abbreviations, Conversion Equations, Drafting Symbols, A.S.H.V.E. Codes

Absolute Humidity: See *Humidity.*

Absolute Pressure: The sum, at any particular time, of the gage pressure and the atmospheric pressure.

Absolute Temperature: The temperature of a substance measured above *absolute zero.*

Absolute Zero: The temperature (−459.6 F) at which the molecular motion of a substance theoretically ceases. This is the temperature at which the substance theoretically contains no heat energy.

Acceleration: The rate of change of velocity. In the fps system this is expressed in units of one foot per second per second.

$$a = \frac{V}{t}$$

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body. In the fps system this is 32.174 ft per second per second.

Adiabatic: An adjective pertaining to or designating variations in volume or pressure not accompanied by gain or loss of heat. When a substance undergoes adiabatic expansion, since it does not receive heat from without, the work which it does is at the expense of its internal energy, and therefore its temperature falls; similarly, when it is adiabatically compressed its temperature rises.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, fumes and smokes. (Air cleaners include air washers and air filters).

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Air Washer: An enclosure in which air is forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of moving air.

Atmospheric Pressure: The pressure exerted by the atmosphere in all directions, as indicated by a barometer. *Standard atmospheric pressure* is considered to be 14.7 lb per square inch, which is equivalent to 29.92 in. of mercury.

Baffle: A plate or wall for deflecting gases or fluids.

Blast: This word was formerly used to denote forced air circulation, particularly in connection with central fan systems using steam or hot water as the heating medium. As applied in this sense, the word *blast* is now obsolete.

Boiler: A closed vessel in which steam is generated or in which water is heated.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929).

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of $970.2 \times 34.5 = 33{,}471.9$ Btu per hour.

British Thermal Unit: The *mean* British Thermal Unit is $\frac{1}{180}$ of the heat required to raise the temperature of 1 lb of water from 32 F to 212 F. It is substantially equal to the quantity of heat required to raise 1 lb of water from 63 F to 64 F. One Btu $= \frac{1}{3415}$ kwhr.

By-pass: A pipe or duct, usually controlled by valve or damper, for short-circuiting fluid flow.

Calorie: The *mean* calorie is $\frac{1}{100}$ of the heat required to raise the temperature of 1 gram of water from *Zero* C to 100 C. It is substantially equal to the quantity of heat required to raise one gram of water from 14.5 C to 15.5 C.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distribution ducts. (See Chapter 21).

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated, owing to its decrease in density.

Coefficient of Transmission: The amount of heat (Btu) transmitted *from air to air* in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F *between the air on the inside and that on the outside of the wall, floor, roof or ceiling.*

Column Radiator: A type of direct radiator. This radiator has not been listed by manufacturers since 1926.

Comfort Line: The effective temperature at which the largest percentage of adults feel comfortable.

Comfort Zone (*Average*): The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. *Comfort Zone* (*Extreme*): The range of effective temperatures over which one or more adults feel comfortable. (See Chapter 3).

Concealed Radiator: A heating device located within, adjacent to, or exterior to the room being heated but so covered or enclosed or concealed that the heat transfer surface of the device, which may be either a radiator or a convector, does not *see* the room. Such a device transfers its heat to the room largely by convection air currents.

Conductance: The amount of heat (Btu) transmitted from surface to surface in one hour through one square foot of a material or construction, whatever its thickness, when the temperature difference is 1 F between the two surfaces.

Conduction: The transmission of heat through and by means of matter unaccompanied by any obvious motion of the matter.

Conductivity: The amount of heat (Btu) transmitted in one hour through one square foot of a homogeneous material *1 in. thick* for a difference in temperature of 1 F between the two surfaces of the material.

Conductor (*heat*): A material capable of readily conducting heat. The opposite of an insulator or insulation.

Constant Relative Humidity Line: Any line on the psychrometric chart representing a series of conditions which may be evaluated by one percentage of relative humidity; there are also *constant* dry-bulb lines, wet-bulb lines, effective temperature lines, vapor pressure lines, and lines showing other physical properties of air mixed with water vapor.

Control: Any manual or automatic device for the regulation of a machine to keep it at normal operation. If automatic, it is considered that the device is motivated by variations in temperature, pressure, time, light, or other influences.

Convection: The transmission of heat by the circulation of a liquid or a gas such as air. Convection may be *natural* or *forced.*

Convector: A heat transfer surface designed to transfer its heat to surrounding air largely or wholly by convection currents. Such a surface may or may not be enclosed or concealed. When concealed and enclosed the resulting device is sometimes referred to as a concealed radiator. (See also definition of *Radiator*). (See also Chapter 14).

Decibel: The standard unit for noise or sound intensity. One decibel is equal to ten times the logarithm to the base *e* of the ratio of the sound intensities.

Degree-Day: A unit, based upon temperature difference and time, used in specifying the nominal heating load in winter. For any one day there exists as many degree-days as there are degrees Fahrenheit difference in temperature between the average outside air temperature, taken over a 24-hour period, and a temperature of 65 F.

Dehumidify: To remove water vapor from the atmosphere; to remove water vapor or moisture from any material.

Density: The weight of a unit volume, expressed in pounds per cubic foot. $d = \frac{W}{V}$.

Dew-Point Temperature: The temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct Radiator: Same as *Radiator.*

Direct-Return System (*Hot water*)*:* A hot water system in which the water, after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*)*:* A pipe which carries steam downward to the heating units and into which the condensation from the heating units drain.

Down-Feed System (*Steam*)*:* A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft Head (*Side Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening. (*Top Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Drip: A pipe, or a steam trap and a pipe, considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry Air: Air with which no water vapor is mixed. This term is used comparatively, since in nature there is always some water vapor included in air, and such water vapor, being a gas, is dry.

Dry-Bulb Temperature: The temperature of the air indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air.

Dry Return: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. (See *Wet Return*).

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

Dynamic Head or Pressure: The total or impact pressure. This is the sum of the radial pressure and the velocity pressure at the point of measurement.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is a composite index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Enthalpy: Total heat or thermal potential.

Entropy: A ratio, evaluated for practical purposes by dividing the heat content of a unit weight of a substance by its absolute temperature. Useful in examining changes during a heat cycle. Entropy is constant during a reversible adiabatic change of state.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and atmospheric pressure.

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined. It is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932).

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932).

Extended Heating Surface: See *Heating Surface.*

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure or by being soldered to the core or by both pressure and soldering. An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector.

Fan Furnace System: See *Warm Air Heating System.*

Force: The action on a body which tends to change its relative condition as to rest or motion. $F = \frac{WV}{gt}$.

Fumes: Particles of solid matter resulting from such chemical processes as combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also, a firepot.

Furnace Volume (*total*): The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (*i.e.*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929).

Gage Pressure: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg

connected to the pressure source and the other exposed to atmospheric pressure.

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down-draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as defined above. (A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers).

Gravity Warm Air Heating System: See *Warm Air Heating System.*

Heat: A form of energy generated by the transformation of some other form of energy, as by combustion, chemical action, or friction. According to the molecular theory, heat consists of the kinetic and potential energy of the molecules of a substance. The addition of heat energy to a body increases the temperature or the kinetic energy of motion of its molecules (*sensible heat*) or increases their potential energy of position but does not increase the temperature, as when melting or boiling occurs (*latent heat*).

Heating Medium: A substance such as water, steam, air, electricity or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. *Extended heating surface (or extended surface):* Heating surface having air on both sides and heated by conduction from the prime surface. *Prime Surface:* Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also *Boiler Heating Surface*).

Heat of the Liquid: The sensible heat of a mass of liquid above an arbitrary zero.

Horsepower: A unit to indicate the time rate of doing work equal to 550 ft-lb per second or 33,000 ft-lb per minute. (One horsepower = 745.8 watts. In practice this is considered 746 watts).

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried through pipes from the boiler to the heating units.

Humidify: To add water vapor to the atmosphere; to add water vapor or moisture to any material.

Humidistat: A regulatory device, actuated by changes in humidity, used for the control of humidity.

Humidity: The water vapor mixed with dry air in the atmosphere. *Absolute humidity* refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. *Specific humidity* refers to the weight of water vapor in pounds carried by one pound of dry air. *Relative humidity* is a ratio, usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting

from the water vapor present in that space. Relative humidity is either the ratio of the actual partial pressure of the water vapor in the air to the saturation pressure at the dry-bulb temperature, or the ratio of the actual density of the vapor to the density of saturated vapor at the dry-bulb temperature. The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space.

Hygrostat: Same as *Humidistat.*

Inch of Water: A measure of pressure which refers to the difference in the heights of the legs of a water-filled manometer.

Insulation (*heat*)*:* A material having a relatively high heat-resistance per unit of thickness.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Latent Heat: See *Heat.*

Laws of Thermodynamics: The *first law* states that the total energy of an isolated system remains constant and cannot be increased or diminished by any physical process whatever. The *second law* states that no change in a system of bodies that takes place of itself can increase the available energy of a system.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter, in pounds, to which the unit of force (one pound) will give an acceleration of one foot per second per second.

$$m = \frac{W}{g}.$$

Mb, Mbh[1]**:** Symbols which represent, respectively, 1000 Btu and 1000 Btu per hour.

Mechanical Equivalent of Heat: The mechanical energy necessary to produce 1 Btu of heat energy. $J = 777.5$ ft-lb.

Micron: A unit of length, the thousandth part of one millimeter or the millionth of a meter.

Mol: The unit of weight for gases. It is defined as m lb where m denotes the molecular weight of a gas. For any gas the volume of 1 *mol* at 32 F and standard atmospheric pressure is 358.65 cu ft and the weight of a cubic foot is 0.002788 m lb.

One-Pipe Supply Riser (*steam*)*:* A pipe which carries steam upward to a heating unit and which also carries the condensation from the heating unit in a direction opposite to the steam flow.

One-Pipe System (*hot water*)*:* A hot water system in which the water flows through more than one heating unit before it returns to the boiler; consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

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[1] *These symbols were approved by the* A.S.H.V.E., June, 1933.

One-Pipe System (*steam*): A steam heating system consisting of a main circuit in which the steam and condensate flow in the same pipe, usually in opposite directions. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

Overhead System: Any steam or hot water system in which the supply main is above the heating units. With a steam system the return must be below the heating units; with a water system, the return *may* be above the heating units.

Panel Radiator: A heating unit placed on or flush with a flat wall surface and intended to function essentially as a radiator.

Panel Warming: A method of heating involving the installation of the heating units (pipe coils) within the wall, floor or ceiling of the room, so that the heating process takes place mainly by radiation from the wall, floor or ceiling surfaces to the objects in the room.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Potentiometer: An instrument for measuring or comparing small electromotive forces.

Power: The rate of performing work, expressed in units of horsepower, one of which is equal to 550 ft-lb of work per second, or 33,000 ft-lb per minute.

Prime Surface: See *Heating Surface.*

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere. *Psychrometric:* Pertaining to psychrometry or the state of the atmosphere as to moisture. *Psychrometry:* The branch of physics that treats of the measurement of degree of moisture, especially the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiation: The transmission of heat through space by wave motion.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects "it can see" and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called *radiator* is also a *convector* but the single term *radiator* has been established by long usage.

Recessed Radiator: A heating unit set back into a wall recess but not enclosed.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Relative Humidity: See *Humidity:* see also discussion of relative humidity in Chapter 1.

Return Mains: The pipes which return the heating medium from the heating units to the source of heat supply.

Reversed-Return System (*hot water*): A hot water heating system in which the water from several heating units is returned along paths arranged so that all circuits composing the system or composing a major sub-division of the system are practically of equal length.

Roof Ventilator: A device placed on the roof of a building to facilitate egress of air.

Saturated Air: Air containing as much water vapor as it can hold without any condensing out; in saturated air, the partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature.

Sensible Heat: See *Heat.*

Smoke: Carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Specific Gravity: The ratio of the weight of a body to the weight of an equal volume of water at some standard temperature, usually 39.2 F.

Specific Heat: The quantity of heat, expressed in Btu, required to raise the temperature of 1 lb of a substance 1 F.

Specific Volume: The volume, expressed in cubic feet, of one pound of a substance. $v = \frac{1}{d} = \frac{V}{W}$.

Split System: A system in which the heating and ventilating are accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point

Square Foot of Heating Surface (*equivalent*): *Equivalent direct radiation* (EDR). By definition, that amount of heating surface which will give off 240 Btu per hour. The *equivalent* square feet of heating surface may have no direct relation to the actual surface area.

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Standard Air: As defined by A.S.H.V.E. codes, *standard air* is air weighing 0.07488 lb per cubic foot, which is air at 68 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.92 in. of mercury. (Most engineering tables and formulae involving the weight of air are based on air weighing 0.07492 lb per cubic foot, which is dry air at 70 F dry-bulb with a barometric pressure of 29.921 in. of mercury. The error involved in disregarding the difference between the above two weights is very slight and in most instances may be neglected).

Static Pressure: The compressive pressure existing in a fluid. It is a measure of the potential energy of the fluid.

Steam: Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor Steam in contact with the water from which it has been generated may be *dry saturated* steam or *wet saturated* steam. The latter contains more or less actual water in the form of mist. If steam is heated, and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated.*

Steam Heating System: A heating system in which heat is trans-

ferred from the boiler or other source of steam to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate and preventing the passage of steam, or for allowing the passage of air as well as condensate.

Superheated Steam: See *Steam.*

Supply Mains (*steam*)*:* The pipes through which the steam flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface Conductance: The amount of heat (Btu) transmitted by radiation, conduction, and convection *from a surface to the air or liquid surrounding it,* or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg between the surface and the surrounding air or liquid.

Therm: Symbol used in the gas industry representing 100,000 Btu.

Thermal Resistance: The reciprocal of *conductance.*

Thermal Resistivity: The reciprocal of *conductivity.*

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the source of heat supply.

Ton of Refrigeration: The extraction of 12,000 Btu per hour.

Ton Day of Refrigeration: The heat removed by a ton of refrigeration operating for one day; 288,000 Btu.

Total Heat: A thermodynamic quantity, variously called heat content, thermal potential, enthalpy. It is the heat required per unit mass (Btu per pound) to raise a given substance to a given point from an arbitrary datum point. It is the sum of the heat of the liquid, the latent heat, and any miscellaneous heat which may be present.

Total Pressure: The sum of the static or radial pressure and the velocity pressure at the point of measurement.

Tube (or Tubular) Radiator: A cast-iron heating unit used as a radiator and having small vertical tubes.

Two-Pipe System (*steam or water*)*:* A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit.

Underfeed Distribution System (*hot water*)*:* A hot water heating system in which the main flow pipe is below the heating units.

Underfeed Stoker: A stoker which feeds the coal underneath the fuel bed.

Unit: As applied to heating, ventilating and air conditioning equipment this word means a factory-built and assembled equipment with apparatus for accomplishing some specified function or combination of functions.

It is loosely applied to a great variety of equipment. Usually the function is included in the name, and hence come terms like Unit Heater, Unit Ventilator, Humidifying Unit, and Air Conditioning Unit.

Units are said to be *direct* or *room*, when intended for location, or located in, the treated space; *indirect* or *remote*, when outside or adjacent to the treated space. They are *ceiling* units when suspended from above, and *floor* when supported from below. Other descriptive words include *free delivery* when the unit is not intended to be attached to ducts or similar resistance-producing devices, and *pressure* when for use with such ducts. Complete description requires the use of several of these qualifying words or phrases. (See Chapter 23).

Up-Feed System (*steam*)*:* A steam heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vapor: Any substance in the gaseous state.

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensation to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return. *Direct Vent Vapor System:* A vapor heating system with air valves which do not permit re-entry of air.

Vapor Pressure: The equilibrium pressure exerted by a vapor in contact with its liquid.

Velocity: The time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of one foot per second. $V = \frac{s}{t}$.

Velocity Pressure: The pressure corresponding to the velocity of flow. It is a measure of the kinetic energy of the fluid.

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning*).

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts. If the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing, it is termed a *gravity* system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a *fan furnace* system or a *central fan furnace* system. A fan furnace system may include air washers and filters.

Wet-Bulb Temperature: The lowest temperature which a water wetted body will attain when exposed to an air current. This is the temperature of adiabatic saturation.

Wet Return: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. (See *Dry Return*).

ABBREVIATIONS [2]

Absolute	abs
Acceleration, due to gravity	*g*
Acceleration, linear	*a*
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Area	*A*
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiler pressure	bp
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
Calorie	cal
Centigram	cg
Centimeter	cm
Centimeter-gram-second (system)	cgs
Change in specific volume during vaporization	v_{fg}
Cubic	cu
Cubic foot	cu ft
Cubic feet per minute	cfm
Cubic feet per second	cfs
Decibel	db
Degree[3]	deg or °
Degree centigrade	C
Degree Fahrenheit	F
Degree Kelvin	K
Degree Réaumur	R
Density, Weight per unit volume, Specific weight	*d* or ρ (rho)

$$d = \frac{1}{v}$$

Diameter	*D* or diam
Direct-current (as adjective)	d-c
Distance, linear	*s*
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, Vapor in contact with liquid	*Subscript* g
Entropy (The capital should be used for any weight, and the small letter for unit weight)	*S* or *s*
Feet per minute	fpm
Feet per second	fps
Foot	ft
Foot-pound	ft-lb
Foot-pound-second (system)	fps
Force, total load	*F*
Freezing point	fp
Gallon	gal
Gallons per minute	gpm
Gallons per second	gps
Gram	g
Gram-calorie	g-cal

[2]From compilations of abbreviations approved by the *American Standards Association*, Z, 10 a, c, f, and i. As a general rule the period is omitted in all abbreviations except where the omission results in the formation of an English word.

[3]It is recommended that the abbreviation for the temperature scale, F, C, K, be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted; as 68 F.

Head .... H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight) .... H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid .... h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor .... h_g
Heat of vaporization at constant pressure .... L or h_{fg}
Horsepower .... hp
Horsepower-hour .... hp-hr
Inch .... in.
Inch-pound .... in.-lb
Indicated horsepower .... ihp
Indicated horsepower-hour .... ihp-hr
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight) .... U or u
Kilogram .... kg
Kilowatt .... kw
Kilowatthour .... kwhr
Length of path of heat flow, thickness L
Load, total .... W
Mass .... m
Mechanical efficiency .... e_m
Mechanical equivalent of heat .... J
Melting point .... mp
Meter .... m
Micron .... μ (mu)
Miles per hour .... mph
Minute .... min
Molecular weight .... mol. wt
Mol. .... mol
Ounce .... oz
Power, Horsepower, Work per unit time .... P
Pressure, Absolute pressure, Gage pressure, Force per unit area .... p
Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred .... Q
Quality of steam, Pounds of dry steam per pound of mixture x
Revolutions per minute .... rpm
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor .... *Subscript* f
Specific gravity .... sp gr
Specific heat .... sp ht or c
Specific heat at constant pressure .... c_p
Specific heat at constant volume .... c_v
Specific volume, Volume per unit weight, Volume per unit mass v
Square foot .... sq ft
Square inch .... sq in.
Temperature (ordinary) F or C (*Theta* is used preferably only when t is used for Time in the same discussion). .... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature). .... T or Θ (*capital theta*)
Thermal conductance[4] (heat transferred per unit time per degree) C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, Unit conductance (heat transferred per unit time per unit area per degree) .... C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

[4]Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties* Examples are—conductivity and resistivity Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities* Examples are—conductance and transmittance. Terms ending *ion* designate rate of heat transfer Examples are—conduction and transmission

Thermal conductivity (heat transferred per unit time per unit area, and per degree per unit length) .... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer (heat transferred per unit time per unit area per degree) .... f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film).

Over-all coefficient of heat transfer, Thermal transmittance per unit area (heat transferred per unit time per unit area per degree over-all) .... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

Thermal resistance (degrees per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity .... $1/k$
Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure .... *Subscript* fg
Velocity V
Volume (total) V
Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight .... q
Watt .... w
Watthour .... whr
Weight of a major item, Total weight .... W
Weight rate, Weight per unit of power, Weight per unit of time .... w
Work (total) .... W

CONVERSION EQUATIONS

Fahrenheit degrees = 9/5 (centigrade degrees) + 32.

Centigrade degrees = 5/9 (Fahrenheit degrees − 32).

Absolute temperature, expressed in Fahrenheit degrees = Fahrenheit degrees + 459.6. In heating and ventilating work, 460 is usually used.

Absolute temperature, expressed in centigrade degrees = centigrade degrees + 273.1.

Power, Heat, and Work

1 ton refrigeration	= 12,000 Btu per hour; 200 Btu per minute
Latent heat of ice	= 143.33 Btu per pound
1 Btu	= 777.5 ft-lb; 0.293 whr; 252.02 mean calories
1 watthour	= 2,655.2 ft-lb; 3.415 Btu; 3600 joules; 860.648 mean calories
1 kilowatthour	= 3,415 Btu; 3.52 lb water evaporated from and at 212 F; 34.15 lb water raised 100 F
1 mean calorie	= 0.003968 Btu; 3.085 ft-lb; 0.0011619 whr
1 kilowatt (1000 watts)	= 1.3405 hp; 56.92 Btu per minute; 44,252.7 ft-lb per minute
1 horsepower	= 0.746 kw; 42.44 Btu per minute; 33,000 ft-lb per minute; 550 ft-lb per second
1 boiler horsepower	= 33,471.9 Btu per hour; 9.80 kwhr

Weight and Volume

1 gal (U. S.)	= 231 cu in.; 0.13368 cu ft
1 British or Imperial gallon	= 277.274 cu in.
1 cu ft	= 7.4805 gal; 1728 cu in.
1 cu ft water at 60 F	= 62.37 lb
1 cu ft water at 212 F	= 59.76 lb
1 gal water at 60 F	= 8.34 lb
1 gal water at 212 F	= 7.99 lb
1 lb (avdp)	= 16 oz; 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb
1 long ton	= 2240 lb

Pressure

1 lb per square inch	= 144 lb per square foot; 2 0416 in. mercury at 62 F; 2.309 ft water at 62 F; 27.71 in. water at 62 F
1 oz per square inch	= 0.1276 in. mercury at 62 F; 1 732 in. water at 62 F

1 atmosphere	= 14.7 lb per square inch 2116.3 lb per square foot 33.974 ft water at 62 F 30 in. mercury at 62 F 29.921 in. mercury at 32 F
1 in. water at 62 F	= 0.03609 lb per square inch 0 5774 oz per square inch 5.196 lb per square foot
1 ft water at 62 F	= 0.433 lb per square inch 62.355 lb per square foot
1 in. mercury at 62 F	= 0.491 lb per square inch 7.86 oz per square inch 1.131 ft water at 62 F 13.57 in. water at 62 F

Metric Units

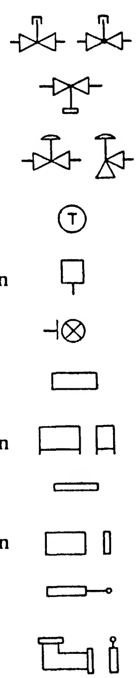
1 cm	= 0.3937 in.
1 in.	= 2.54 cm
1 m	= 3.281 ft
1 ft	= 0.3048 m
1 sq cm	= 0.155 sq in.
1 sq in.	= 6.45 sq cm
1 sq m	= 10 765 sq ft
1 sq ft	= 0.0929 sq m
1 cu cm	= 0.061 cu in.
1 cu in.	= 16.39 cu cm
1 cu m	= 35.32 cu ft
1 cu ft	= 0.0283 cu m
1 liter	= 1000 cu cm = 0.264 gal
1 kg	= 2 2046 lb
1 lb	= 0 4536 kg
1 metric ton	= 2205 lb (avdp)
1 gram	= 980.59 dynes = 0.002205 lb
1 kilometer per hour	= 0.6214 mph
1 gram per square centimeter	= 0 0290 in. mercury, at 0 deg C 0.394 in. water, at 15 C
1 kg per square centimeter (metric atmosphere)	= 14.22 lb per square inch
1 gram per cubic centimeter	= 0.03614 lb per cubic inch 62.43 lb per cubic foot
1 dyne	= 0.00007233 poundals
1 joule	= 10,000,000 ergs 0.73767 ft-lb
1 metric horsepower	= 75 kg-m per second 0.986 hp (U. S.)
1 kilogram-calorie (large calorie)	= 1000 gram-calories (small calorie) 3.97 Btu
1 kilogram-calorie per kilogram	= 1.8 Btu per pound
1 gram-calorie per square centimeter	= 3.687 Btu per square foot
1 gram-calorie per square centimeter per centimeter	= 1.451 Btu per square foot per inch
1 gram-calorie per second per square centimeter for a temperature graduation of 1 deg C per centimeter	= 2903 Btu per hour per square foot for a temperature graduation of 1 deg F per inch of thickness.

SYMBOLS FOR HEATING, VENTILATING AND AIR CONDITIONING DRAWINGS[5]

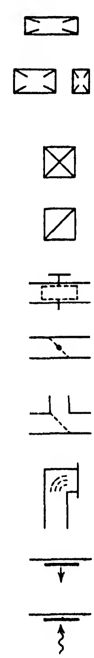
1. The objects of this standard set of symbols are to insure the correct interpretation of drawings and to conserve drafting room time by establishing simple and unmistakable symbols for the component parts of the heating and ventilating systems. In preparing the list of symbols an effort has been made to follow existing practice in so far as possible but the list cannot be expected to match exactly the existing practice of every drafting room

1. General Piping
2. Steam Piping
3. Condensate Piping
4. Cold Water Piping
5. Hot Water Piping
6. Air Piping
7. Vacuum Piping
8. Gas Piping
9. Refrigerant Piping
10. Oil Piping

11. Lock and Shield Valve
12. Reducing Valve
13. Diaphragm Valve
14. Thermostat
15. Radiator Trap Elevation
16. Radiator Trap Plan
17. Tube Radiator Plan
18. Tube Radiator Elevation
19. Wall Radiator Plan
20. Wall Radiator Elevation
21. Pipe Coil Plan
22. Pipe Coil Elevation



23. Indirect Radiator Plan
24. Indirect Radiator Elevation
25. Supply Duct, Section
26. Exhaust Duct, Section
27. Butterfly Damper Plan (or Elevation)
28. Butterfly Damper Elevation (or Plan)
29. Deflecting Damper Rectangular Pipe
30. Vanes
31. Air Supply Outlet
32. Exhaust Inlet



[5]From A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, edition of 1929, and American Standard Drawings and Drafting Room Practice Graphical Symbols (*American Standards Association*, Z14.2—1935).

33. Joint

34. Elbow—90 deg

35. Elbow—45 deg

36. Elbow—Turned Up

37. Elbow—Turned Down

38. Elbow—Long Radius

39. Side Outlet Elbow—Outlet Down

40. Side Outlet Elbow—Outlet Up

41. Base Elbow

42. Double Branch Elbow

43. Single Sweep Tee

44. Double Sweep Tee

45. Reducing Elbow

46. Tee

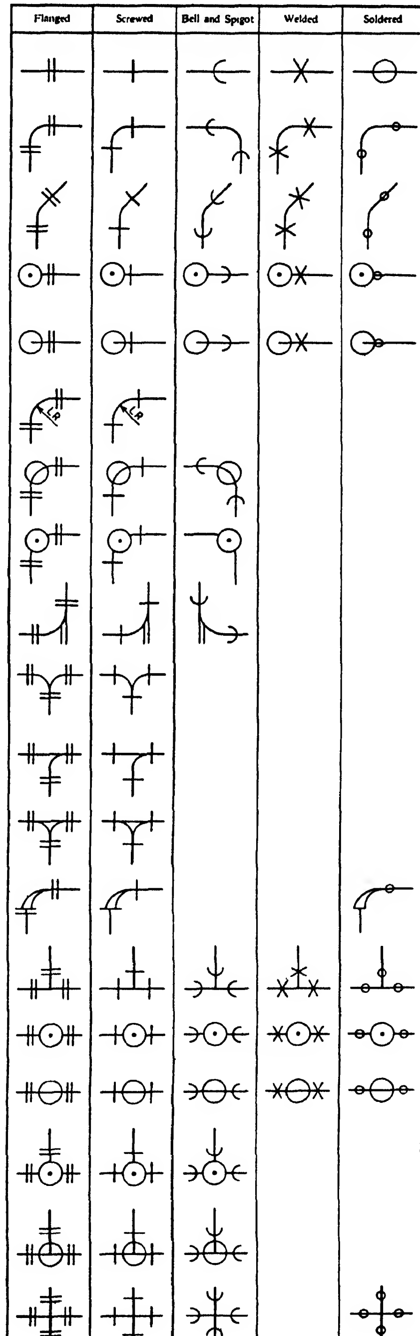
47. Tee—Outlet Up

48. Tee—Outlet Down

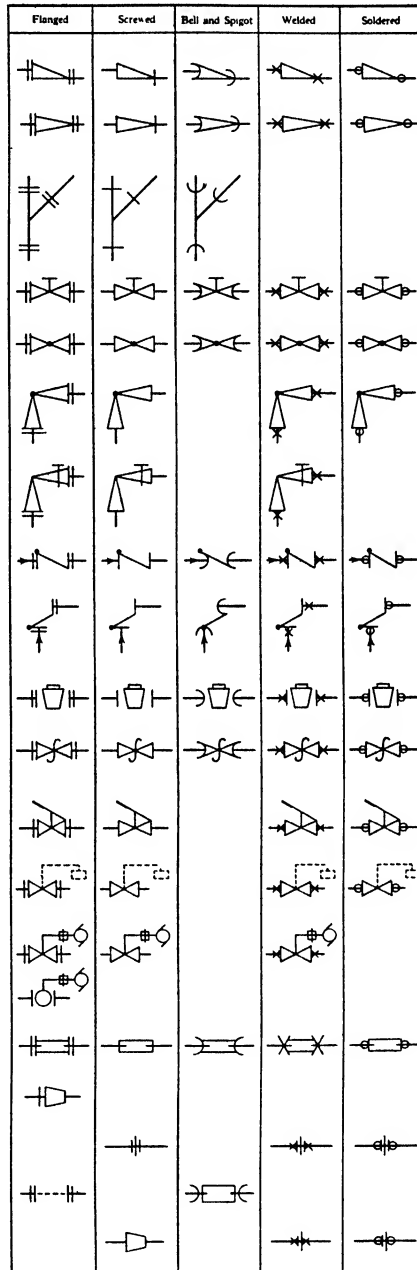
49. Side Outlet Tee—Outlet Up

50. Side Outlet Tee—Outlet Down

51. Cross



52. Eccentric Reducer
53. Reducer
54. Lateral
55. Gate Valve
56. Globe Valve
57. Angle Globe Valve
58. Angle Gate Valve
59. Check Valve
60. Angle Check Valve
61. Stop Cock
62 Safety Valve
63. Quick Opening Valve
64. Float Operating Valve
65. Motor Operated Gate Valve
66. Motor Operated Globe Valve
67. Expansion Joint Flanged
68. Reducing Flange
69. Union
70. Sleeve
71. Bushing



A.S.H.V.E. CODES

The following codes and standards relating to the design, installation, testing, rating, and maintenance of materials and equipment used for the heating, ventilation and air conditioning of buildings, have been adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Air Cleaning Devices	A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work[a]	June, 1933	A.S.H.V.E TRANSACTIONS, Vol. 39, 1933, p. 225
Air Conditioning	Code of Minimum Requirements for Comfort Air Conditioning[a]	January, 1938	A S H.V E. Reprint
Boilers (testing)	Standard and Short-Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2)[a]	June, 1929	A.S.H.V E. TRANSACTIONS, Vol. 35, 1929, p. 322
Boilers (testing)	A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code 3)[a] [b]	June, 1929	A.S.H.V.E TRANSACTIONS, Vol. 35, 1929, p 332
Boilers—Oil Fuel (testing)	A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[a]	June, 1932	A S H.V.E. TRANSACTIONS, Vol. 37, 1931, p 23
Boilers Stoker-Fired (testing)	A S.H.V.E Standard Code for Testing Stoker-Fired Steam Heating Boilers[a]	January, 1938	A.S H.V.E. Reprint
Boilers (rating)	A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[a]	January, 1929 Revised April, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p 42
Concealed Gravity Type Radiation	A.S.H.V E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Section)[a]	June, 1933 Revised January, 1935	A S H V E TRANSACTIONS, Vol. 39, 1933, p. 237
Convectors	A S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[a]	January, 1931 Revised January, 1935	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367
Ethics	Code of Ethics for Engineers	January, 1922	A.S.H.V.E. TRANSACTIONS, Vol 28, 1922, p. 6 (See frontispiece THE GUIDE, 1939)
Fans	Standard Test Code for Centrifugal and Axial Fans[a]	1938 Edition	A S H.V E. Reprint[c]

[a]Reprints available

[b]Originally adopted by the *National Boiler and Radiator Manufacturers Association*

[c]See A S.H.V E TRANSACTIONS, Vol 29, 1923, p 407; Vol 37, 1931, p 363

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Garages	Code for Heating and Ventilating Garages	June, 1929 Revised January, 1935	A S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 355 A.S.H.V.E. Reprint
Heat Transmission Through Walls	Standard Test Code for Heat Transmission through Walls[a]	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 253
Minimum Requirements	Code of Minimum Requirements for Heating and Ventilation of Buildings, Edition–1929	June, 1925	A.S.H.V.E. Codes
Radiators	Code for Testing Radiators[a]	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 18
Unit Heaters	Standard Code for Testing and Rating Steam Unit Heaters[a d]	January, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165
Unit Ventilators	A.S H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators[a]	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25
Vacuum Heating Pumps	A S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps[a]	June, 1934	A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 33
Ventilation	Report of Committee on Ventilation Standards[a]	August, 1932	A S.H.V.E. TRANSACTIONS, Vol 38, 1932, p 383

The following Codes and Standards have been endorsed or approved by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	SPONSORED BY	REFERENCE
Air Conditioning Equipment	Standard Method of Rating and Testing Air Conditioning Equipment[e]	*American Society of Refrigerating Engineers*	*American Society of Refrigerating Engineers,* New York, N. Y.
Chimneys	Standard Ordinance for Chimney Construction	*National Board of Fire Underwriters*	Chapter 14, THE GUIDE, 1931
Condensing Units	Standard Method of Rating and Testing Mechanical Condensing Units[e]	*American Society of Refrigerating Engineers*	*American Society of Refrigerating Engineers,* New York, N. Y.
Piping Systems	Identification of Piping Systems[f]	*American Society of Mechanical Engineers*	*Heating, Piping and Air Conditioning,* July, 1929
Warm Air Furnaces	Standard Code Regulating the Installation of Gravity Warm Air Furnaces in Residences	*National Warm Air Heating and Air Conditioning Association*	*National Warm Air Heating and Air Conditioning Association,* Columbus, Ohio

[d]Adopted jointly by the *Industrial Unit Heater Association*, and the A S H V.E.
[e]Proposed code prepared by Joint Committee of *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association, National Electric Manufacturers Association* and *Air Conditioning Manufacturers Association*
[f]Adopted November, 1928, Sponsored by (1) *American Society of Mechanical Engineers*, (2) *National Safety Council.*

TABLE 1. CIRCUMFERENCES AND AREAS OF CIRCLES

Diameter in Inches	Area		Circumference		Diameter in Inches	Area		Circumference	
	Sq In.	Sq Ft	Inches	Feet		Sq In.	Sq Ft	Inches	Feet
¼	0.049	0 0003	0.785	0.0652	28	615.8	4.276	87.97	7.3
½	0.196	0.0014	1.571	0.1309	28½	637.9	4.430	89.54	7 4
¾	0 442	0.0031	2.356	0.1964	29	660.52	4.587	91.11	7.5
1	0.785	0 0054	3.142	0.2618	29½	683.5	4.747	92.63	7.7
1¼	1.227	0 0085	3.927	0 3273	30	706.8	4.909	94.25	7.8
1½	1.767	0.0123	4 712	0 3927	31	754.8	5.241	97.39	8.11
1¾	2.405	0 0167	5.498	0 4582	32	804.3	5.585	100.5	8 37
2	3.142	0 0218	6.283	0 5236	33	855 3	5.940	103.7	8.6
2¼	3.976	0.0276	7 069	0 5891	34	907.9	6.305	106.8	8 9
2½	4.909	0 0341	7 854	0.6546	35	962.1	6.681	109.9	9 1
2¾	5.939	0.0412	8 639	0.7200	36	1018 0	7.069	113.1	9.42
3	7.069	0.0491	9 425	0.7854	37	1075 0	7 467	116 2	9 6
3¼	8.296	0.0576	10 21	0 8510	38	1134 0	7 876	119 4	9 94
3½	9.621	0 0668	10 99	0 9160	39	1195 0	8 296	122.5	10.21
3¾	11.04	0.0767	11 78	0 9818	40	1256 0	8 727	125.6	10 47
4	12.57	0.0873	12 57	1.047	41	1320.0	9.168	128 8	10 73
4¼	14.19	0.0986	13 35	1.113	42	1385.0	9 621	131.9	10 99
4½	15.90	0.1104	14 14	1 178	43	1452.0	10 08	135 1	11.26
4¾	17 72	0 1231	14 92	1 243	44	1521 0	10 56	138 2	11.52
5	19 64	0 1364	15 71	1 309	45	1590.0	11 04	141 4	11 78
5¼	21.65	0 1504	16 49	1 374	46	1662 0	11 54	144 5	12.04
5½	23.76	0 1650	17.28	1 440	47	1735 0	12.05	147.7	12.30
5¾	25 97	0 1840	18 06	1 505	48	1810.0	12 51	150.8	12.57
6	28 27	0 1964	18 85	1.571	49	1886 0	13 09	153 9	12.83
6¼	30 68	0.2131	19 64	1 637	50	1963.0	13.64	157 1	13 09
6½	33 18	0 2304	20 42	1 702	51	2043.0	14.19	160 2	13 35
6¾	35.79	0 2486	21 21	1 768	52	2124 0	14.75	163 4	13 61
7	38.49	0.2673	21.99	1 833	53	2206 0	15 32	166 5	13.88
7¼	41.28	0 2867	22 78	1 899	54	2290 0	15 90	169 6	14 14
7½	44.18	0 3068	23.56	1 964	55	2376 0	16.50	172 8	14 40
7¾	47 17	0 3276	24 35	2 029	56	2463 0	17.10	175 9	14 66
8	50.27	0 3491	25 13	2 094	57	2552.0	17 72	179 1	14 92
8¼	53 46	0.3713	25 92	2 160	58	2642 0	18 35	182 2	15 18
8½	56.75	0 3942	26 70	2 225	59	2734 0	18 99	185.4	15.45
8¾	60.13	0 4175	27 49	2 291	60	2827 0	19 63	188 5	15 71
9	63.62	0 4418	28.27	2.356	61	2922.0	20 29	191 6	15.97
9¼	67.20	0 4668	29.06	2.422	62	3019 0	20 97	194 8	16.23
9½	70 88	0.4923	29.85	2.488	63	3117.0	21 65	197 9	16 49
9¾	74 66	0.5185	30 63	2 553	64	3217 0	22.34	201.1	16 76
10	78.54	0 5454	31.42	2.618	65	3318 0	23 04	204.2	17 02
10½	86.59	0 6010	32.99	2.750	66	3421 0	23 76	207 3	17.28
11	95.03	0.6600	34.56	2.880	67	3526.0	24 48	210.5	17.54
11½	103.9	0 7215	36.13	3 011	68	3632.0	25 22	213 6	17 80
12	113.1	0 7854	37.70	3.142	69	3739.0	25 97	216 8	18.06
12½	122.7	0 8520	39 27	3 273	70	3848 0	26 73	219 9	18 33
13	132 7	0.9218	40 84	3.403	71	3959 0	27 49	223 1	18.59
13½	143.1	0 9937	42.41	3.535	72	4072 0	28 27	226 2	18.85
14	153.9	1.069	43.98	3.665	73	4185.0	29.07	229 3	19.11
14½	165.1	1.146	45.55	3.796	74	4301.0	29 87	232 5	19.37
15	176.7	1 227	47.12	3.927	75	4418.0	30.68	235.6	19 63
15½	188 7	1.310	48 69	4.058	76	4536 0	31.50	238 8	19 90
16	201.1	1.396	50.27	4.189	77	4657.0	32.34	241.9	20 16
16½	213.8	1.485	51.84	4.321	78	4778.0	33.18	245 0	20 42
17	226.9	1.576	53.41	4.451	79	4902.0	34 04	248 2	20 68
17½	240.5	1 670	54.98	4 582	80	5027.0	34 91	251.3	20 94
18	254 5	1.767	56.55	4.712	81	5153 0	35 78	254 5	21.21
18½	268.8	1.867	58.12	4.845	82	5281 0	36.67	257 6	21 47
19	283 5	1.969	59 69	4.974	83	5411 0	37 57	260 8	21.73
19½	298.6	2.074	61 26	5 105	84	5542 0	38 48	263 9	21.99
20	314 2	2 182	62 83	5 236	85	5675 0	39 41	267 0	22 25
20½	330.1	2 293	64.40	5 367	86	5809 0	40 34	270 2	22.51
21	346.4	2 405	65.97	5.498	87	5945 0	41 28	273.3	22.78
21½	361 1	2 508	67.54	5.629	88	6082 0	42.24	276 5	23 04
22	380 1	2 640	69 12	5 760	89	6221.0	43 20	279.6	23.30
22½	397 6	2.761	70 69	5.891	90	6362 0	44.18	282.7	23.56
23	415 5	2.885	72 26	6 021	91	6504.0	45.17	285.9	23.82
23½	433.7	3.012	73 83	6.153	92	6648.0	46.16	289.0	24.09
24	452.4	3.142	75.40	6.283	93	6793.0	47.17	292.2	24.35
24½	471.4	3 274	76 97	6 415	94	6940.0	48.19	295.3	24.61
25	490.9	3 409	78.54	6.545	95	7088.0	49.22	298.4	24.87
25½	510.7	3.547	80 11	6.676	96	7238.0	50.27	301.6	25.13
26	530 9	3 687	81.68	6 807	97	7390.0	51.32	304.7	25.39
26½	551 6	3.832	83.25	6.938	98	7543.0	52.38	307.9	25.66
27	572.6	3 976	84 82	7.069	99	7698 0	53 46	311.0	25.92
27½	593.9	4 125	86.39	7.199	100	7854.0	54.54	314.2	26 18

CATALOG DATA SECTION

The HEATING, VENTILATING AIR CONDITIONING GUIDE 1939

INDEX TO ADVERTISERS

PAGE 859

INDEX TO MODERN EQUIPMENT

PAGE 1137

In this Catalog Data Section of The Guide 184 manufacturers describe the most modern types of heating, ventilating and air conditioning equipment and materials—259 pages of valuable data, profusely illustrated.

Alphabetical arrangement of advertisers—on pages 859-864—permits ready reference to the product of a specific manufacturer.

Various types of apparatus and materials are grouped in their respective classes, and may be located readily by reference to the page headings—Boilers, Heaters, Insulation, Pumps, Valves, etc.—as on pages 865-1134.

A classified Index to Modern Equipment—on pages 1137-1160—contains complete listings of manufacturers whose products are described in the Catalog Data Section.

INDEX TO ADVERTISERS

HEATING VENTILATING AIR CONDITIONING GUIDE, 1939

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Y

American Moistening Company

ESTABLISHED 1888

Atlanta Ga
Boston Mass

Providence, R. I.

Charlotte, N C
Greenville S C

AMCO

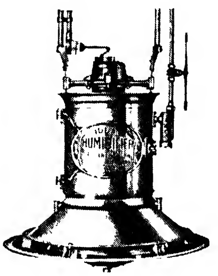
UNIT HUMIDIFYING AND AIR CONDITIONING EQUIPMENT

A few of many AMCO products with a Long Record of Dependable Performance

Sectional Humidifiers
Amtex Humidifiers
Hand Sprayers
Mine Sprays
Fabric and Paper Dampeners

Mechanical Psychrometers
Electro Psychrometers
Sling Psychrometers
Hygrometers

The Amco line of devices for the supply, maintenance and control of humidity is complete in its ability to meet any presented problem of applied humidification Used independently or as an adjunct to Central Station equipment, these devices automatically maintain any required humidity condition in a capable uniform performance

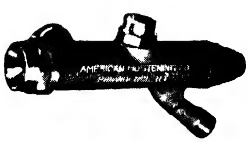


IDEAL HUMIDIFIERS—Senior Type

A high capacity unit for use where conditions require a great amount and good distribution of moisture Motor driven fan gives wide distribution of atomized spray Amco heads serve the triple purpose of humidifying, air washing and cooling

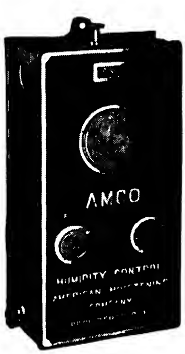
IDEAL HUMIDIFIERS—Junior Type

Similar in construction to Senior Type. Used where medium capacities are required.



AMCO ATOMIZER—No. 4

Quality and quantity of spray are maintained even under adverse conditions because this atomizer is automatically self-cleaning When the compressed air supply is shut off, either manually or in response to a humidity control, *both* air and water nozzles are thoroughly cleaned.



AMCO HUMIDITY CONTROLS

Compressed Air Operated

An extremely accurate and active device operated by compressed air which assures a regulation of humidity within exceedingly close ranges

AMCO HUMIDITY CONTROL

Electrically Operated

Similar in principle to the Compressed Air Type except that the hydroscopic element operates electrical contacts which control the units

American Blower Corporation

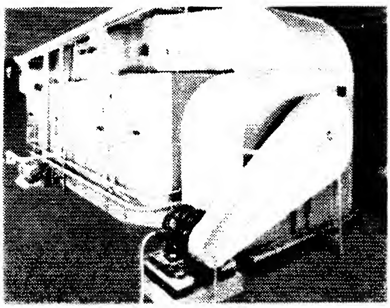
Division of American Radiator and Standard Sanitary Corporation

General Offices and Factory

Detroit, Mich.

Branches in All Principal Cities

AIR CONDITIONING — HUMIDIFYING — DEHUMIDIFYING — COOLING — VENTILATING — HEATING — VAPOR-ABSORPTION — DRYING — AIR WASHING AND PURIFICATION — EXHAUSTING EQUIPMENT AND MECHANICAL DRAFT APPARATUS.

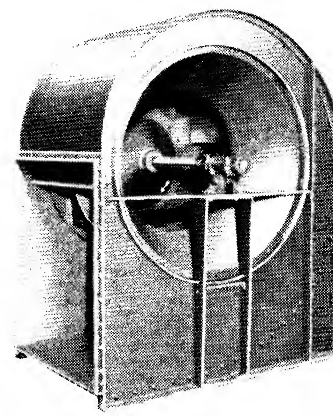


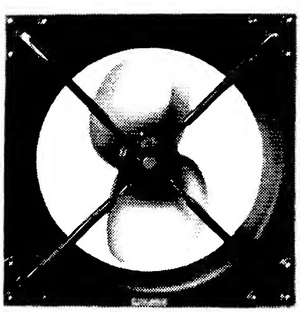
American Blower Central Air Conditioning Systems with Sirocco Fan

For more than 30 years, particularly adapted to heating, ventilating, air washing, purifying and cooling large buildings. Write for complete data.

American Blower Multi-Blade Fan

For heating, ventilating, cooling and air conditioning system. High volumetric and mechanical efficiency. Driven by belt, steam engine, motor or turbine. Capacities 30,000 to 500,000 cfm. Write for Bulletin No. A-403.





Ventura Ventilating Fan

For exhausting bad air, odors, steam, etc., in a wide range of buildings and industrial processes, restaurants, garages, etc. Write for Bulletin No. A-1913.

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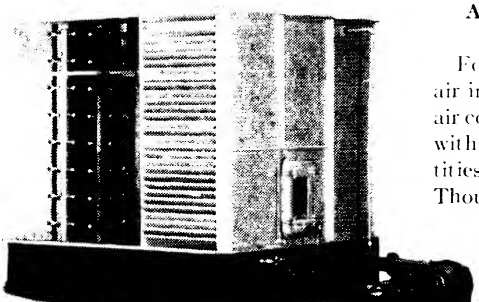


American Blower Dehumidifiers and Washers

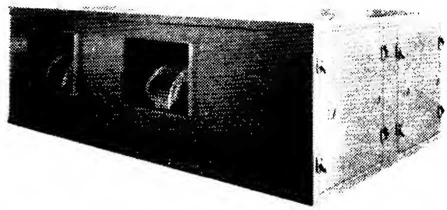
For dehumidifying, cooling and washing air in connection with central systems, for air conditioning, for process work or for use with a ventilating system where large quantities of fresh, washed air are required. Thousands of these dehumidifiers are already in operation in theatres, public buildings, office buildings, garages, hotels and industrial plants. Write for Special Bulletin No. 3523.

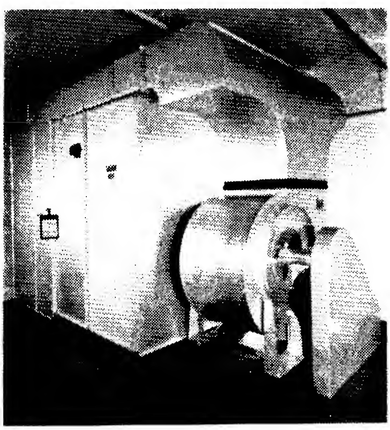
TYPES OF AMERICAN BLOWER CORPORATION AIR HANDLING AND CONDITIONING EQUIPMENT

All types of air handling and air conditioning equipment for industrial applications, process work, drying, cooling; also equipment for stores, offices, shops, public buildings, power plants, etc., and attic ventilation for homes.

American Blower Series B Conditioner

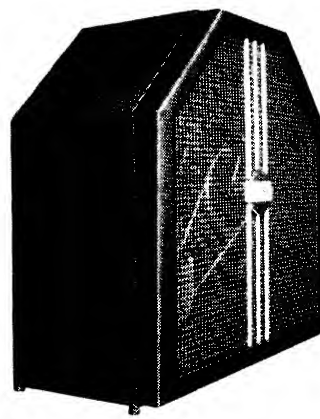
For installation in stores, restaurants, offices and other commercial establishments. May be installed in present buildings or new buildings for cooling, ventilating, dehumidifying as well as heating and humidification. Write for technical Bulletin No. 5127.





American Blower Series K Conditioner

For use with a duct system in a wide variety of air conditioning applications. For complete data on this conditioner write for Bulletin No. 3927.



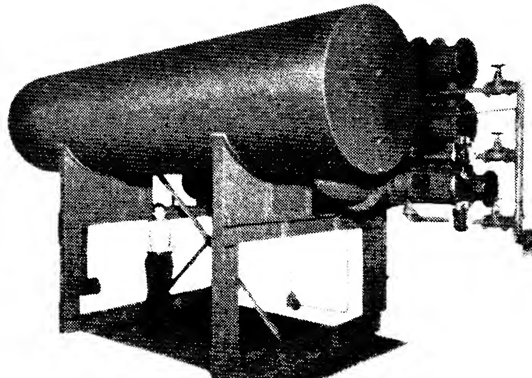
Ventura Home Conditioner

For comfort cooling by means of attic ventilation. Made in various sizes for all home needs. No refrigerating machine required. Thousands already in operation. Write for Bulletin No. A-3713.

•

Decalorator Steam Refrigeration Units

Furnished for air conditioning and process cooling. By means of a jet of steam and the flow of condensing water, they function to produce cool, refrigerated water at any temperature above 35 F. For complete data on steam refrigeration, send for Bulletin No. 3727.



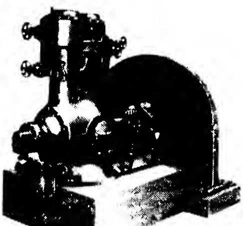
Baker Ice Machine Co., Inc.

Omaha, Nebr.

MANUFACTURERS OF INDUSTRIAL AND COMMERCIAL REFRIGERATION AND AIR CONDITIONING

Branch Factories: Fort Worth, Los Angeles, Seattle
Eastern Sales: New York City Central Sales: Chicago
Sales and Service in All Principal Cities Authority on Mechanical Cooling for Over 30 Years

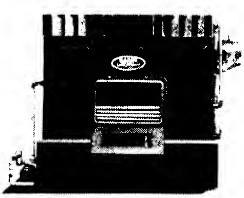
Get specifications for your requirements from the Baker line of equipment. Every machine and cooling assembly has been precision-manufactured and designed to offer maximum service, dependability and economy per dollar invested.



Baker Ammonia Compressor

Baker Ammonia Compressors

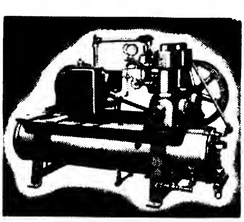
Available to 100 tons capacity, with synchronous, direct-connected or V-belt drive. Baker Compressors may be arranged in duplex or multiple installations for any desired capacity. Also equipped with double-suction, capacity reduction where conditions require utmost economy of operation.



Baker ColdStream Brine Spray Unit—forced draft type

Baker Cold-Stream Brine Spray Units Forced Draft

Designed for applications requiring uniform control of temperatures and relative humidity. Equipped with slow-speed blowers mounted on ball bearings. Fan speeds may be changed to suit air velocity requirements. Housing is of boiler plate construction.

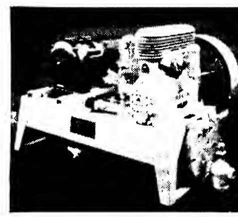


Baker Ammonia Type Self-Contained Unit

Baker Ammonia Type Self-Contained Units

Ranging in size from $\frac{1}{2}$-ton to 25-ton capacities, 24 different models are included in the complete Baker line of ammonia type self-contained units. Complete with motor, drive and control, mounted with condenser on rigid steel base. Two and four-cylinder models.

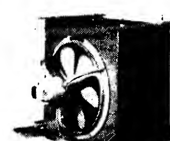
Baker Freon or Methyl Chloride Units



Baker Freon or Methyl-Chloride Unit

Baker offers a complete line of 77 models assembled in both single and dual mounted self-contained automatic units from $\frac{1}{4}$ hp to 60 hp capacity, two and four cylinder types. Both air cooled and water cooled models available.

Baker Fan Type ColdStream Units



Baker Fan Type ColdStream Unit

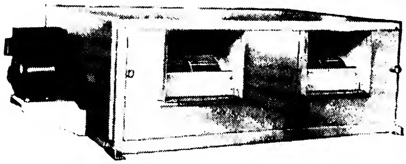
Rigidly constructed for compact, high-capacity, heavy duty service in refrigerating and air conditioning foods and other perishables requiring positive control of temperatures above the freezing point. Finned coil surfaces and air velocity designed for a correct combination of temperature and relative humidity.

Baker ColdStream Brine Spray Units Gravity Flow

Baker ColdStream Brine Spray Unit Gravity Flow Type

Especially designed for cooling and air conditioning service in rooms which require absorption of excessive quantities of heat and moisture, such as are found in meat processing plants and pre-cooling plants. Suction created by brine spray jets draws warm air into grille openings at top. Passing through these *atomized* brine sprays, air is cooled and dehydrated, forced downward and out through rust-proof moisture eliminators near bottom of unit.

Baker Ice Machine Co., Inc.
Omaha, Nebr.

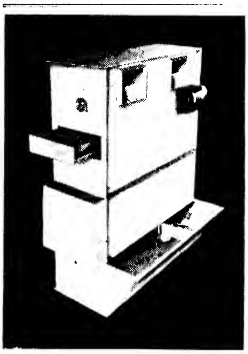


Baker ColdStream Air Conditioner

Baker ColdStream Air Conditioners

A wide range of capacities is available to suit every need. Condensed capacities for largest and smallest sizes:

Cfm	Water Btu	Direct Expansion Btu
3,000	65,200	64,500
12,000	433,000	484,000



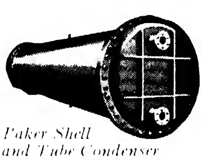
Baker Evaporative Condenser

Baker Evaporative Condensers

Compactly designed, eliminates expense of cooling towers with separate condensers and saves space. Drastically reduces water costs. Casing is of heavy metal, rigidly braced and thoroughly reinforced. Non-corrosive type eliminators. Outdoor units are weatherproofed.

Baker Shell and Tube Condensers

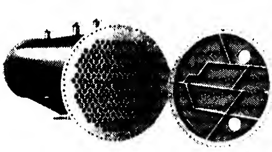
Made in all sizes up to 2500 sq ft of effective cooling surface. Vertical, horizontal, multi-pass or single pass types available, with diameters and tube lengths to fit any specification. Easily cleaned.



Baker Shell and Tube Condenser

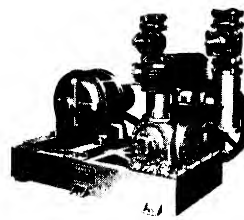
Baker Liquid Coolers

Designed to cool quickly large quantities of water or brine. Horizontal multipass shell and tube construction. Complete range of sizes. Cooling capacity up to 150 tons each. Easily cleaned.



Baker Liquid Cooler

Baker Four-Cylinder Freon Compression Units

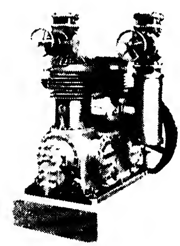


Baker Four-Cylinder Freon Compression Unit

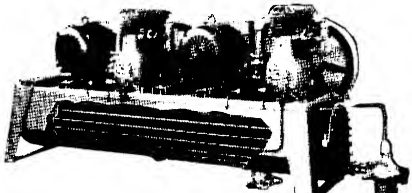
Baker 4-cylinder vertical enclosed type Freon compression unit, assembled on a rigid metal base with motor and automatic control. Available in 30 to 60 hp. Timken anti-friction roller bearings, balanced bellows crankshaft seal, and built-in removable cartridge-type oil filter. Automatic pressure-type temperature control and high-pressure cut-out (thermostat type also available). V-belt drive. Compactly built, all parts easily accessible.

Baker Booster Compressors

Especially designed for quick freezing and low temperature work. Two and four cylinder models. Arranged for single or dual assembly, direct connected or V-belt drive. Positive lubrication, Press-R-Seal, Timken tapered bearings.



Baker Booster Compressor



Baker Dual Condensing Unit

Baker Dual Condensing Units

Designed especially for variable heat load requirements. Dual 4-cylinder type water cooled Freon or Methyl Chloride unit, with automatic capacity control. Equipped with shell and tube type extra capacity condenser or for evaporative type condenser. Cut-away view shows position and length of tubes.

Clarage Fan Company

Kalamazoo, Michigan

Sales Engineering Offices in All Principal Cities

(Consult Telephone Directory)

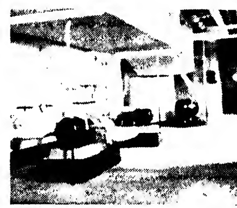
CLARAGE AIR HANDLING AND CONDITIONING EQUIPMENT

For Over a Quarter-Century Clarage has been a leading manufacturer of air handling and conditioning equipment. There is a Clarage fan or blower, conditioning unit or system to meet every need, from the simplest ventilating or cooling job to the most exacting temperature and humidity control installation.

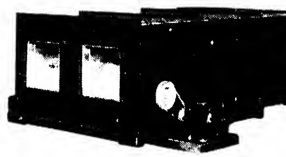
Whatever your ventilating, unit heating, cooling, drying, air cleaning, humidifying, dehumidifying or complete air conditioning problem, we can meet your requirements successfully and economically.

Clarage Experience covers every conceivable type of installation, commercial, industrial and public building. Clarage equipment is used in the largest industrial plants, office buildings, auditoriums, theatres, hotels, restaurants, retail stores, hospitals, churches and schools.

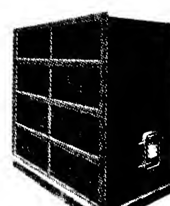
Architects, Engineers and Contractors find our service specially helpful. This Company is an independent manufacturer selling through regular trade channels, and cooperating fully with those who specify and those who install. Your inquiry for data on any Clarage product is invited. Write for Bulletins.



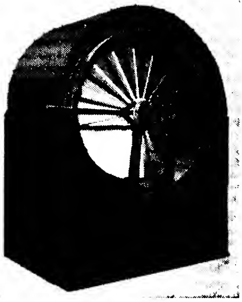
Clarage Systems for complete air conditioning in public buildings and industrial plants.



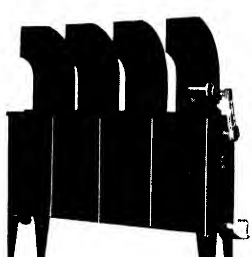
Multitherm Units for complete conditioning, summer cooling, or winter heating.



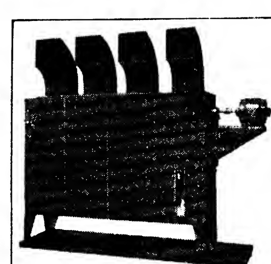
Unicoil Units for use in small air conditioning systems.



Clarage Fan with Vortex (constant speed) Volume Control for ventilation and air conditioning.



Unitherm Unit Heaters with Syncrotherm Temperature Control for factory heating.



Unitherm Unit Coolers for product cooling and refrigeration.

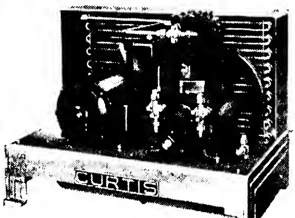
Curtis Refrigerating Machine Company

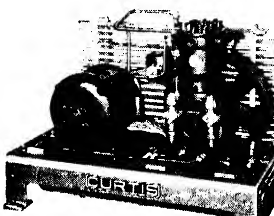
***Division of* Curtis Mfg. Company**

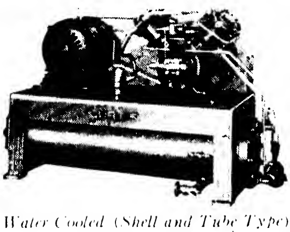
1959 Kienlen Avenue
St. Louis, Mo.

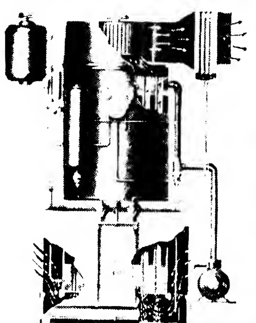
CURTIS REFRIGERATION AIR CONDITIONING AND COMMERCIAL

Est. 1854
New York Office
30 Vesey St.


Air Cooled Condensing Units—1/4 hp to 5 hp sizes.


Water Cooled (Counter-flow Type) 1/3 hp to 5 hp sizes.


Water Cooled (Shell and Tube Type) 3 hp up to 30 ton capacity.


Evaporative Condenser Receiver 3 to 20 ton capacity.

GENERAL INFORMATION ABOUT CURTIS

90 models comprise the Curtis line of condensing units—45 air-cooled units in sizes from 1/6 to 5 hp; 45 water-cooled units from 1/3 to 30 hp. Available for either Freon (F12) or Methyl Chloride.

MECHANICAL FEATURES OF CURTIS UNITS

Curtis compressors employ "Centro-Ring" positive pressure lubrication—only one moving part—no gears or plunger.

Timken Tapered Roller Bearings (models above 3/4 hp) . . . Cylinder Heads removable . . . Drop forged, alloy steel, heat-treated crank-shaft and connecting rods . . . Balanced sylphon bellows seal . . . Unusually large receivers and condensers assure efficiency.

TYPICAL CURTIS UNITS

Air-Cooled Condensing Units For commercial cooling. Extra large air-cooled condensers and condenser receivers.

Water-Cooled Condensing Units (Counter-flow Type)—For commercial and air conditioning work when water is free from salt or other mineral content.

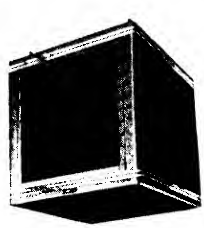
Water-Cooled (Shell and Tube Type) **Condensing Units**—With cleanable condensers having removable heads.

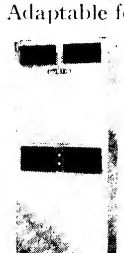
Evaporative Condenser Receiver Cleanable type, designed to provide economical condensing of refrigerant vapors for high water rates and high water temperature.

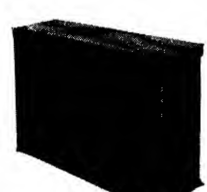
Floor Type Cooling Units—Attractive appearance, efficient operation. Two models one for comfort cooling, one for complete air conditioning.

Comfort Ceiling Type Units Each unit complete with filter, blower fan, efficient cooling unit. Furnished for cold and hot water circulation.

Store and Office Cooler A complete packaged air conditioning unit requiring only water and electrical connections to install. Cools, dehumidifies, circulates and filters the air. Adaptable for heating.


Comfort Ceiling Type Unit—3/4, 1 1/2 and 3 ton capacity.


Store and Office Cooler—3 and 5 ton capacity.


Floor Type Cooling Units—1/2 to 1 1/2 ton capacity.

Albany
Atlanta
Baltimore
Boston
Buffalo
Charlotte
Chicago
Cincinnati
Dallas
Detroit

Frick Company

(Incorporated)

Air Conditioning, Refrigerating and Ice-Making Equipment

Waynesboro, Penna.

Kansas City
Los Angeles
Memphis
New Orleans
New York
Oklahoma City
Palatka
Philadelphia
Pittsburgh
St. Louis
Seattle

Distributors in 150 Principal Cities

AIR CONDITIONING

We furnish complete air conditioning systems as well as refrigerating machinery for use with equipment furnished by others. Upwards of a thousand installations attest the value of the various Frick systems of air conditioning, some of which are patented, and of those made under the patents of the Auditorium Conditioning Corp. Ask for Bulletin 505, describing the four principal kinds of systems; also Bulletins 504 and 512, illustrating and listing typical jobs. Estimates cheerfully furnished.

The Philtower and Philcade Buildings at Tulsa are Air Conditioned with 1000 Tons of Frick Refrigeration

FRICK FREON-12 REFRIGERATION

Includes a complete line of enclosed type Freon-12 compressors. Large capacity, ample gas passages, pressure lubrication from internal pump, patented FLEXO-SEAL at shaft. Coils, coolers, condensers and controls for Freon-12 systems. Bulletin 508.

38 Schrafft's Restaurants Use a Total of 145 Frick Machines for Air Conditioning and Food Service

AMMONIA REFRIGERATION

Machines in all capacities from ½ ton up. Combined units, vertical enclosed type compressors, horizontal machines: complete high and low sides. Widely used for air conditioning. Bulletins 102 to 138.

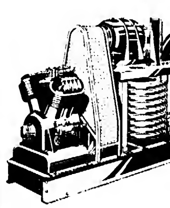
The Standard Carmel Company of Lancaster, Penna., has used Frick Ammonia Refrigeration for Air Conditioning Since 1910

CARBON DIOXIDE REFRIGERATION

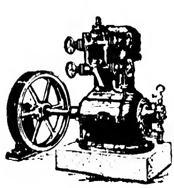
Six sizes of enclosed CO_2 compressors: smooth running, efficient and reliable machines; condensers, coolers, etc. Bulletin 118.

LOW PRESSURE REFRIGERATION

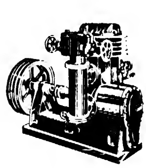
Commercial units in more than 50 sizes and types, with motors of ¼ to 30 hp. Charged with Freon-12 or methyl chloride. Air and water cooled condensers. Finned coils, fan and blower units, ice cube and beverage coolers, etc. Bulletins 97 and 98.



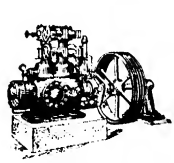
20-Ton Freon-12 Unit for Air Conditioning Work



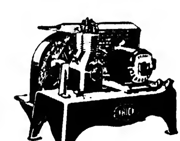
Enclosed Type Ammonia Compressor



Frick Enclosed Freon-12 Compressor



Enclosed Compressor for Carbon Dioxide



Low Pressure Refrigerating Units

Ingersoll-Rand Company

11 Broadway, New York City

Branches or Distributors the World Over

ATLANTA	CINCINNATI	DULUTH	KNOXVILLE	PICHER	SCRANTON
BIRMINGHAM	CLEVELAND	EL PASO	LOS ANGELES	PITTSBURGH	SEATTLE
BOSTON	DALLAS	HARTFORD	NEWARK	POTTSVILLE	ST. LOUIS
BUFFALO	DENVER	HOUSTON	NEW YORK	SALT LAKE CITY	TULSA
BUTTE	DETROIT	KANSAS CITY	PHILADELPHIA	SAN FRANCISCO	WASHINGTON
CHICAGO					

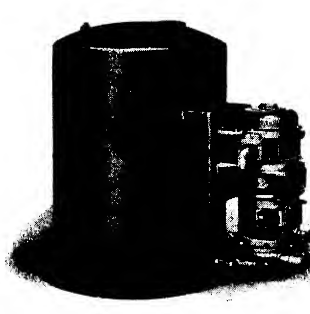
PUMPS

Motorpump

There is an efficient, reliable Ingersoll-Rand pump for every purpose. Single-stage centrifugal units range from 40 to 100,000 gpm, and multi-stage units (2 to 8 stages) range from 125 to 3000 gpm for pressures up to and over 3000 lb.

The "Motorpump" is ideal for general service everywhere, ranging in sizes from ¼ to 40 hp. Pumps and motors are built as one compact and efficient unit on a single shaft. They can be mounted in any position on the floor, wall or ceiling. Capacities range from 5 to 1000 gpm for heads as high as 240 ft single-stage; and from 20 to 275 gpm for heads up to 500 ft two-stage. Motors for any common current conditions; open, splash-proof, totally-enclosed, or explosion-proof types.

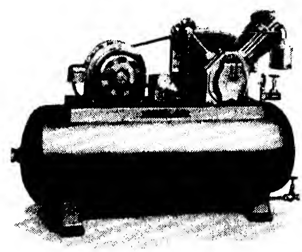
CONDENSATE RETURN UNITS



"Motorpump" Condensate Return Units are designed to return condensation from radiation systems, heaters, steam coils, etc.

The units consist of standard "Motorpumps" mounted on 15-, 30-, or 60-gal tanks and controlled by a float switch. They are suitable for use in schools, apartment houses, greenhouses, industrial plants, chemical plants, etc. They are often used to replace inefficient steam traps.

AIR COMPRESSORS



Type "30" Two-Stage Compressor

A "Type 30" Air Compressor is an excellent source of compressed air for use with regulating devices or control equipment, and for many other applications requiring small capacities. It is a self-contained plant, consisting of a compressor, driving motor, air receiver and automatic pressure control switch. The compressor is either single- or two-stage, using I-R steel valves. Cylinders and intercooler are air cooled. Capacities range from 1.2 to 102 cfm. Pressures range from 5 to 1000 lb per sq in.

Ingersoll-Rand manufactures more than 1000 sizes and types of compressors including a complete line of ammonia compressors.

OTHER I-R PRODUCTS

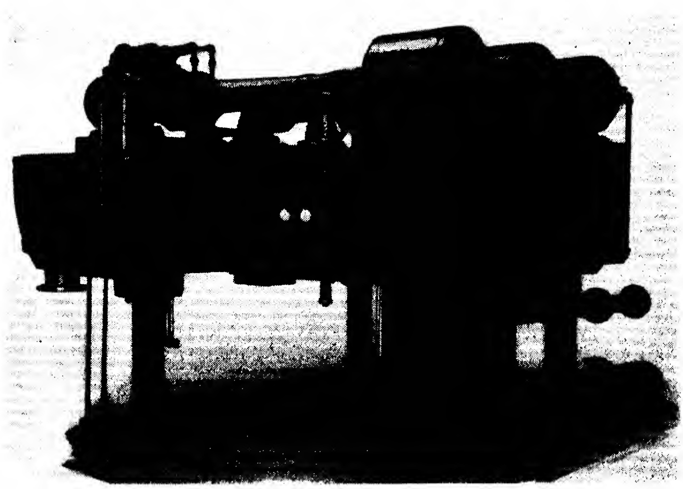
Surface condensers; counter-current and ejector-jet barometric condensers; steam-jet ejectors; vacuum pumps; centrifugal blowers; air aftercoolers and receivers; Diesel and gas engines; air and electric hoists; rock drills; and pneumatic tools of many kinds.

WATER-VAPOR REFRIGERATION

I-R Water-Vapor Refrigeration uses water as the only refrigerating medium. There is no refrigerant to purchase or replace. The system operates entirely under vacuum, and may be opened up for inspection whenever desired.

Water-Vapor Refrigeration gives low operating costs by saving refrigerant replacement and by economical operation at light as well as full loads. Of special value is its unusual ability to handle overloads. Full capacity is retained for the life of the machine.

I-R Units are ordinarily sold through reliable contractors for field installation as a part of air conditioning systems.



Cooling is effected by direct evaporation of water at high vacuum. Steam-jet boosters maintain the vacuum and discharge the evaporated water vapor into a surface condenser. (When desired, barometric type condenser units can be furnished). The heat carried away by the evaporation of the water vapor chills the main body of water as it is circulated through the unit.

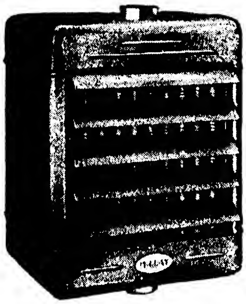
The I-R patented design has been specially developed to insure reliability and simplicity of operation. Steam-Jet Coolers are free from noise and vibration, and are easy to install and maintain.

Standard sizes are available for capacities of approximately 10 tons upward for chilled water temperatures of 35 F and higher. Units are built for steam pressures of 2 lb gauge and upward.

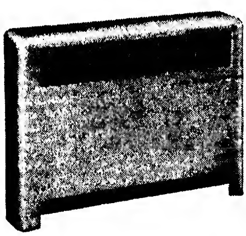
McQuay, Inc.

1600 Broadway, N.E., Minneapolis, Minn.

MANUFACTURERS OF
AIR CONDITIONING EQUIPMENT



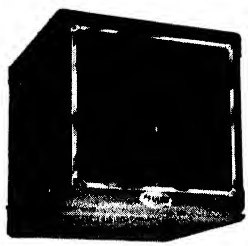
Unit Heaters. Full floating, all copper heating elements. Improved bond, between fin and tubes.
26 sizes



Concealed and Cabinet Copper Radiation (Convectors). Enclosure types, exposed floor and wall hung, fully or partially recessed. Removable panels and concealed. All copper heating elements.

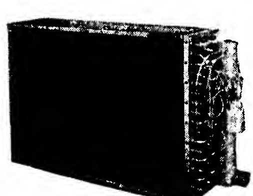
Suspended Cooling Unit. Blower Type. Companion to floor type. Combination cooling and heating.

Evaporator Coils. For all phases of commercial refrigeration. Tailor-made to fit specific requirements.

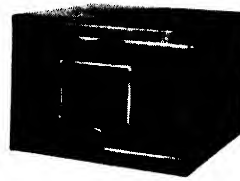


Comfort Coolers. For direct expansion or water as cooling medium; also as combination cooling and heating units. Capacities for any requirements.

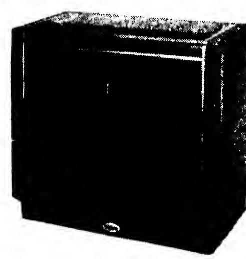
Unit Coolers. For truck refrigeration, storage rooms, etc. All refrigerants. Nine sizes.



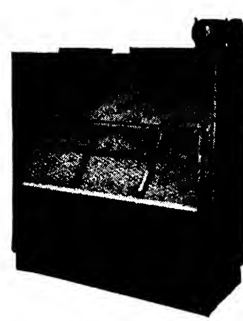
Air-Conditioning Coils. For central fan heating and cooling systems. For cooling and dehumidifying. Direct expansion refrigerants or water for cooling, steam or hot water for heating. Sizes for any air conditioning application.



Air-Conditioning Unit. Suspended Type. New line, low priced, combining high efficiency of centrifugal fans and air filters. Cold water or Freon cooling medium. Attractive cabinets, adaptable for fresh air connection.



Cabinet Room Coolers. Floor type for home or office. Attractive, compact. Water or direct expansion refrigerants.



Floor Type Cooling and Heating Units. Six sizes, 3 to 50 tons cooling capacities.

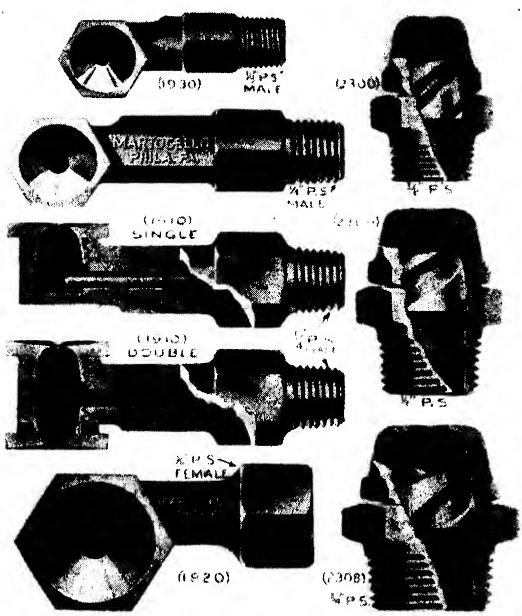
NEW DESCRIPTIVE BULLETINS ARE READY on all McQuay Products.

Jos. A. Martocello & Company

229-31 North 13th Street, **Philadelphia, Pa.**

ATOMIZING SPRAY NOZZLES

Martocello Atomizing Spray Nozzles contribute greatly to the general efficiency of air washing and air conditioning installations, and permit the complete apparatus to function at its maximum rated capacity.



Types of Martocello Spray Nozzles

Successful—efficient operation of air washing and air conditioning equipment depends largely upon having the proper design of spray nozzles.

Martocello Atomizing Spray Nozzles are used exclusively by many manufacturers because they give a uniformly fine, wide spray having a spread of 95 to 115 deg.

For obtaining maximum efficiency from our Standard Spray Nozzles, we recommend with orifices as indicated in the table below. Any reasonable range of capacities at various pressures can be obtained from the sizes shown.

We appreciate all difficult Air Conditioning problems being submitted to us for proper recommendations and results.

Sizes and Capacities

Pipe Size Inches	Part No.	Diam. Orifice Inches	Capacity, Gallons per Minute 5 lb	10 lb	15 lb	20 lb	25 lb	30 lb	35 lb	40 lb
1/8	1930	7/64	.22	.29	.34	.39	.44	.49	.54	.59
1/4	1910	13/64	.54	.77	.96	1.13	1.29	1.44	1.58	1.71
1/4	1910 Double	5/32	.86	1.18	1.48	1.76	2.02	2.24	2.44	2.63
3/8	1920	17/64	1.48	1.96	2.38	2.75	3.08	3.36	3.60	3.82
3/8	2300	7/32	1.98	2.63	3.15	3.62	4.05	4.44	4.80	5.13
1/2	2304	5/16	2.66	3.77	4.71	5.52	6.24	6.87	7.47	8.04
3/4	2308	11/32	3.59	4.87	5.92	6.83	7.62	8.33	8.98	9.60

Brass Forged Nozzles illustrated and also Cast Red Brass Nozzles in 1 in. to 2 in. Pipe Size carried in stock for prompt shipment.

Satisfaction Guaranteed

Parks-Cramer Company

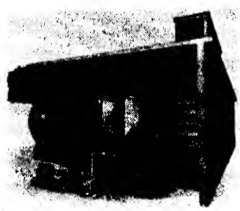
Fitchburg, Mass. **Charlotte, N. C.**

CERTIFIED CLIMATE

Complete Air Conditioning Systems including Heating, Cooling, Humidifying, De-humidifying, Air Changing, Refrigeration, Air Filtering and Air Washing

AUTOMATIC REGULATION

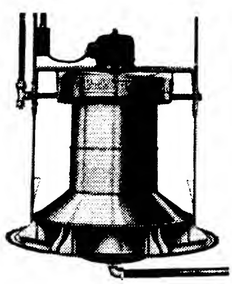
Merrill Process System of Hot Oil Circulation for Heating Industrial Materials



Central Station

Central Station Air Conditioning

A centrally located AIR WASHER supplying correct amount of moisture with positive pre-determined air change. Usually includes indirect radiation for heating —may include refrigeration and cooling. Features absolute control and centralized equipment. Helps in many industries, notably, Textiles (Cotton, Wool, Worsted, Silk, Rayon, Jute); Printing and Lithographing; Cigar, Cigarette and Tobacco; Clothing; Paper and Envelope; Leather and Shoes; Wood Products; Cereals; Storage of Perishables; Ceramics; Celluloid; Glassine Paper; Starch and Dextrine; Cement; Confectionery. Installations similar in design are effective in Hospitals, Art Galleries, Auditoriums, and Restaurants . . . Nozzles for Central Station Air Washers.



High Duty Humidifier

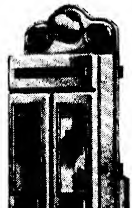
High Duty Humidifier

Water under pressure generates spray. Excess water returns to filter tank and recirculates. Evaporation per unit high; two sizes of heads each with three sizes of nozzles give flexible capacity for varying conditions. Circulation increased by individual motor-driven fan. Spray thoroughly diffused and distributed over wide area.

Turbo and Turbomatic Humidifiers

(not illustrated)

Efficient humidifiers of the atomizer type. For direct humidification and as humidity boosters for Central Station systems of all makes. Self-cleaning.



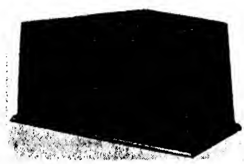
Psychrostat

Parks Automatic Airchanger

For use with High Duty or Turbo Humidifiers. Insures fixed humidity and maximum evaporative cooling.

Automatic Regulation

The Psychrostat for accuracy, durability, sensitivity. Hygrostat (not illustrated) where requirements are not so exacting. Psychrostat employs the principle of the Sling Psychrometer, used by U. S. Government in all Weather Bureau Stations. An Air Conditioning System is no better than its Regulation.



Pettifogger

The Pettifogger

A compact humidifier for offices, stores, storerooms, testing laboratories, and other isolated departments. Entirely self-contained in attractive lacquered copper casing. Permanently though flexibly connected to water and electrical supplies. Automatic control. Adjustable capacity. Reduces dust. Neutralizes drying effect of heating. Conditions textiles and other hygroscopic substances for testing purposes.

Research Corporation

405 Lexington Avenue, **New York City**

RESEARCH SYSTEM OF AIR CONDITIONING

(Patented and Patents Pending)

The Research Corporation System of Air Conditioning features a direct method of chemical dehumidification, and may be applied to either comfort control or industrial processing. It functions either with or without the use of refrigeration, depending upon operating conditions.

In the working unit, or Calorider, a moisture-absorbing salt solution is sprayed over copper extended surface coils through which cooling water circulates.

This salt solution is automatically kept at a definite concentration by heat supplied at relatively low temperatures (e.g., by waste steam).

The system makes possible the independent and yet simultaneous control of temperature and humidity over a wide range, at extremely low operating cost.

Advantages of Research Corporation System of Air Conditioning

1. Installation costs are moderate, operating costs are cut 50 to 80 per cent.
2. Cooling water at 70 deg gives conditioned air with dew-point as low as 22 deg.
3. Maintenance is simple, cheap and infrequent. No reciprocating machinery, no gas to leak. The only moving parts are fans, pumps and their motors.
4. Noise and vibration limited to that of main air supply blower.
5. Fully automatic control, all year 'round if desired; relative humidity may be held ± 3 per cent at any point between 15 and 50 per cent.
6. Equipment available in sizes of 1000, 3000, 6000, 9000, 12000 and 18000 cfm.

COEY MULTISTAGE COOLING TOWERS

(Patented and Patents Pending)

"Water Saved is Money Earned"

The Coey Multistage Cooling Tower is constructed of a copper-bearing steel shell and corrosion resisting supporting members, Red Gulf Cypress wood baffles, a non-overloading reverse blade centrifugal fan rotor, and the Coey Spray Eliminator of copper-bearing steel or aluminum.

Table of Capacities and Sizes

Model No.	Nominal Water Flow Gpm	Motor Hp	Height above beams*	Diam	Operating Wgt Pounds*	Mech. Refrig. Tons	Steam jet Refrig. Tons	Steam Condensing at 27 in. Vac. 115 F Lb per hour
10 or 10-A	100	2.0	9'- 0"	5'- 0"	4500	35	15	1,125
20-R	225	5.0	11'- 5"	8'- 5"	9000	75	35	2,500
35 or 35-R	375	7.5	14'- 2"	11'- 3"	15500	125	60	4,200
50 or 50-R	550	10.0	16'- 9"	13'- 3"	21500	185	90	6,200
85 or 85-R	850	15.0	18'- 8"	15'- 0"	30000	285	135	9,550
110 or 110-R	1100	20.0	21'- 7"	17'- 9"	43000	375	175	12,400
150 or 150-R	1400	25.0	22'-10"	20'- 0"	56000	465	230	15,700

Compact, light in weight, spray free and quiet in operation, it is desirable for installation anywhere, but is especially well-suited for roof and basement installations in congested districts.

Controlled Cooling. Sprayless Operation. Minimum Noise Level. Architectural Harmony.

Complete information on Research Corporation products gladly sent upon request.

Other Equipment: Cottrell Electrical Precipitation Systems—Multiclone Dust Collectors—Impax Separators—Cottrell Royster Deodorizers—Royster Stoves for High Temperature Heat Exchange.

Servel, Inc.

Electric Refrigeration and Air Conditioning Division

Evansville, Indiana

AIR CONDITIONING

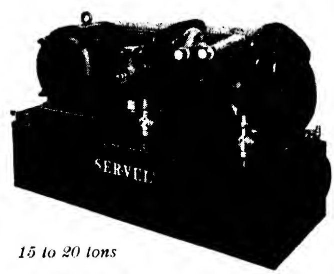
Servel specializes in the manufacture of refrigerating machines for the air conditioning industry. Eliminating side lines and accessories, Servel concentrates its engineering and manufacturing facilities on this one all-important element in every complete system. Backed by 17 years of experience in building heavy-duty low-pressure refrigerating machines, Servel units offer every modern feature, plus a record of proven dependability.

SERVEL'S NEW PRODUCTS FOR 1939

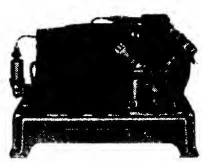
Servel offers a fully integrated line of machine units, designed as a "family." This line is the result of a research program inaugurated early in 1935. Since that time, dozens of new materials, design features, manufacturing processes and practical selling advantages have been developed. All units have been moulded to the modern trend. While special stress has been placed on quiet operation and compactness, there has been no sacrifice of high capacity and overall efficiency.

Remote Machines—For conventional central fan systems or room coolers, Servel offers over twenty models, ranging from ½ ton to 20 tons and including air-cooled, water-cooled and evaporative condenser types. Air-cooled models feature shrouded condensers with leaf-type "Streamair" fans. The larger water-cooled units have the economical counter-flow shell and tube condensers developed by Servel two years ago.

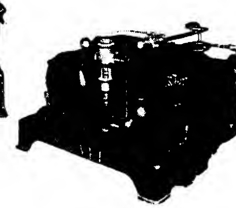
Machines for Self-Contained Units—For manufacturers or assemblers of self-contained conditioners, Servel offers compact assemblies of compressors, condensers and auxiliaries suited to this type of work from ½ ton to 10 tons.



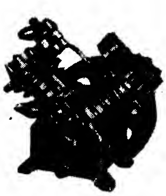
15 to 20 tons



¾ ton



4 to 7 tons



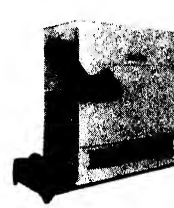
Typical Compressor

COMPRESSORS

Servel offers nine distinct compressors for use with units of ⅓ ton to 20 tons capacity. All except the smallest are four or eight-cylinder compressors, insuring absolute balance and freedom from vibration. Manufacturers will find these compressors adaptable to almost any assembly program. Servel's engineers will be glad to supply technical data, specifications and other details to responsible firms.

EVAPORATIVE CONDENSERS

For sections of the country where water rates are high, or where there is a shortage of water supply or sewage facilities, Servel offers five economical evaporative condensers. Ranging in size from 5 tons to 20 tons, these evaporative condensers are shipped complete, including coil, blower, pump, receiver, eliminator, by-pass connection, motors, drives and control. Each model is designed to handle full capacity, with condensing temperature within 25 F of entering wet-bulb.



The Trane Company

2021 Cameron Avenue, **La Crosse, Wisconsin**

MANUFACTURERS OF HEATING, COOLING AND AIR CONDITIONING EQUIPMENT

Over 70 U.S. Branch Offices

Albany, N. Y., Altoona, Pa., Amarillo, Tex., Appleton, Wis., Atlanta, Ga., Aurora, Ill., Baltimore, Md., Birmingham, Ala., Boston, Mass., Brooklyn, N. Y., Buffalo, N. Y., Canton, Ohio, Charleston, W. Va., Chattanooga, Tenn., Chicago, Ill., Cincinnati, Ohio., Clarksburg, W. Va., Cleveland, Ohio, Columbus, Ohio, Dallas, Tex., Davenport, Ia., Dayton, Ohio, Denver, Col., Des Moines, Ia., Detroit, Mich., Flint, Mich., Gainesville, Fla., Grand Rapids, Mich., Greensboro, N. C., Greenville, S. C., Harrisburg, Pa., Houston, Tex., Indianapolis, Ind., Jackson, Miss., Juneau, Alaska, Kalamazoo, Mich., Kansas City, Mo., Knoxville, Tenn., LaCrosse Wis., Lake Charles, La., Little Rock, Ark., Livingston, Mont., Los Angeles, Calif., Louisville, Ky., Memphis, Tenn., Miami, Fla., Milwaukee, Wis., Missoula, Mont., New Haven, Conn., Newark, N. J., New Orleans, La., New York City, Oklahoma City, Okla., Omaha, Neb., Peoria, Ill., Philadelphia, Pa., Phoenix, Ariz., Pittsburgh, Pa., Portland, Ore., Portsmouth, Ohio, Providence, R. I., Richmond, Va., Rochester, N. Y., St. Louis, Mo., St. Paul, Minn., Salt Lake City, Utah, San Antonio, Tex., San Francisco, Calif., Seattle, Wash., Sioux City, Ia., South Bend, Ind., Spokane, Wash., Syracuse, N. Y., Toledo, Ohio, Washington, D. C., Westchester, N. Y., Wilkes-Barre, Pa., Zanesville, Ohio.

In Canada: Trane Company of Canada, Ltd., Toronto, Ont.

NON-FERROUS CONVECTORS

The Trane Non-Ferrous Convector provides instant, even heat to any room space. Constructed in fifteen basic types and several hundred sizes, the Trane Convector may be placed on the floor, partially or totally concealed in the wall, or hung completely exposed on the wall. This heating unit operates efficiently on all types of steam and hot water systems. Can be installed within range of cast iron radiator prices.

TRANE AIR

The Tranemark of High Quality in Heating, Cooling and Air Conditioning

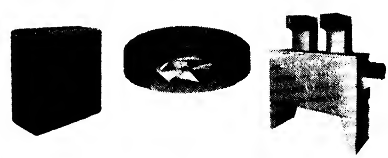
TRANE AIR CONDITIONING MANUAL

A comprehensive handbook and textbook covering the fundamentals of the science of air conditioning. For the engineer and architect. Straightforward. Unbiased. Price: $5.00.

HEATING SPECIALTIES

The Trane Heating Specialty line consists of Low, Medium, and High Pressure Bellows Traps; Thermostatic Drip Traps; Quick and Float Vents; Vent Traps; Hermetic, Packless and Bellows Angle Valves; High and Low Pressure Float Traps; Direct Return Traps; Inverted Bucket Traps; Dirt Strainers; and Damper Regulators.

Trane Heating Specialties are precision built and ruggedly constructed for outstanding performance for any steam system. The heart of Trane Heating Specialties is the famed Trane Bellows. Trane is one of the few heating specialty manufacturers who actually fabricate this important part.



UNIT HEATERS

Projection Unit Heaters—A draw-through propeller type unit heater that taps the reservoir of heat at the ceiling, projecting it down to the living zone. It may be used at heights from 10 to 60 ft. Available in single or multiple fan units in 6 sizes. (Illustrated center, above).

Propeller Type Unit Heaters—A standard unit heater for all types of heating. Available in 60 sizes for steam pressures up to 150 lb or positive circulation hot water systems at 160 deg or higher. Incorporates refinements that eliminate strain and stress caused by expansion, and that direct heat to the floor line. (Left).

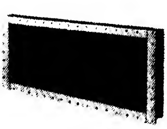
Torridor Blower Type Unit Heaters—A blower type draw-through heater for large space heating with a minimum of units. Their high velocity makes them desirable for duct work application. Available in belt or direct driven, standard or by-pass control units, floor, ceiling and wall types; 96 sizes. (Right).

AIR CONDITIONING EQUIPMENT

Climate Changers—The complete year around air conditioner for the home, apartment and small building. Hot water or steam heating. Cooling with any standard medium. Humidification. Air circulation. Dehumidification. In 12 sizes in horizontal or vertical models.

Commercial Air Conditioners—A compact unit for commercial application. Same features as the Climate Changer. A wide range of sizes and capacities in floor and suspended models.

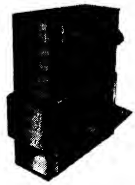
DeLuxe Air Conditioners—For use where attractiveness must be combined with functions of year around air conditioning. Floor Type and Hotel and Office Models. Floor type Unit is a vertical unit of compact design available in 6 cabinet sizes. The Hotel and Office Unit is a horizontal unit for ceiling suspension. Primarily a summer air conditioner.



HEATING AND COOLING COILS

There is a Trane Extended Surface Coil to meet all heating or cooling applications, commercial or industrial.

1. **Type E** Blast Coils for heating and cooling using low and high pressure steam, hot or cold water. Especially designed for duct work installations.
2. **Type C** though similar to Type E is not encased. Side channels provided for structural strength. 1 and 2 row sections and many sizes. Designed primarily for installation in self-contained units.
3. **Type S** for cooling with clean cold water. Available in single and double row serpentine for varying water supplies. In 2, 3, 4, and 6 row depths and a multitude of sizes.
4. **Type R** for cooling with water where tube cleaning may be necessary. In 4, 6, and 8 row depths and a variety of sizes.
5. **Direct Expansion Coils** for cooling with standard refrigerants. 2, 3, 4, and 6 row depths and many sizes.
6. **Welded Header Coils** for steam pressures up to 250 lb. Particularly advantageous for drying applications. 2-row depths and a variety of sizes.
7. **Booster Coils** for booster type heating, ventilating and air conditioning in a central system.



OTHER AIR CONDITIONING EQUIPMENT

1. The Trane Evaporative Condenser to condense refrigerants in any compressor system.
2. The Trane Air Washer for humidification in the average commercial heating and ventilating system.
3. The Trane Type X Propeller Comfort Cooler, an inexpensive quiet propeller fan draw-through unit. Intended for service with water as the cooling medium.
4. The Trane Product Cooler a compact unit used with refrigeration system provides proper temperatures for product storage.
5. Railroad Air Conditioning Equipment includes the Trane Electro Mechanical System for undercar installation and sub-cooler and railroad evaporative condensers to make other systems function more effectively.
6. Bus Air Conditioning Equipment for light weight unobtrusive installations.

UNIT VENTILATORS

Intended primarily for the school room, the Trane Unit Ventilator may be used wherever the combination of positive ventilation and heating is required. The unit is a draw-through type, attractively finished. Available in a variety of sizes and types in class room and auditorium models.

GAS EQUIPMENT

Trane Gas Equipment consists of Unit Heaters and Winter Air Conditioners. The Unit Heater is an automatic unit, light in weight, one of the speediest heat makers ever developed. Can be used with natural or manufactured gas. The Winter Air Conditioners are factory-assembled for duct or free delivery. Humidity, heating, air cleansing, and circulation obtained. May be equipped for summer cooling.

ALLIED EQUIPMENT

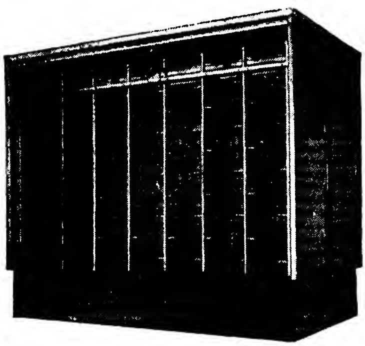
Includes Condensation and Circulating and Booster Pumps, Compressors, Temperature Control Valves to maintain set temperatures on convectors or radiators, and Washer and Atomizing Target Type Spray Nozzles.

United States Air Conditioning Corporation

A Complete Line of Air Conditioning Equipment

2121 Kenedy St., N.E.
Minneapolis, Minn.
Branch Offices or Agents in Principal Cities

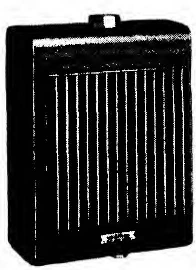
U. S. AIRCO Air Washers



Single and double stage air washers, all sizes, from 2500 cfm to 100,000 cfm.

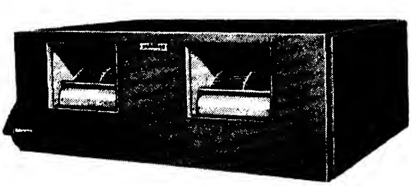
U. S. AIRCO Unit Heaters

Series 39 Unit Heater with U. S. AIRCO patented Deflecto-Grille, horizontal and vertical blades both adjustable for perfect control of air volume and air distribution.



Also Standard Model Unit Heaters with adjustable horizontal louvres.

U. S. AIRCO Blower Type Unit Heaters

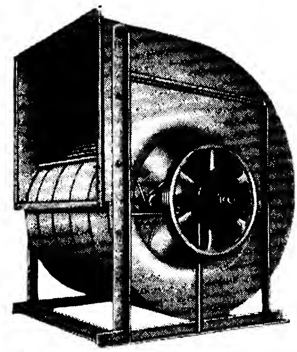


Ceiling suspension and floor models. Available with heating and cooling coils, humidifiers and filters.

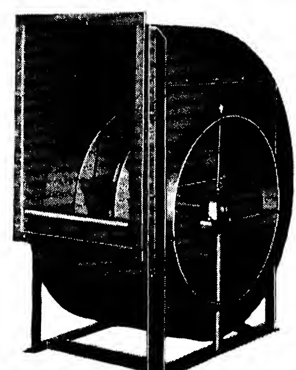
Also blast heat cores.

U. S. AIRCO Blowers

Single inlet single width and double inlet double width Blowers for both supply and exhaust. Sizes from 300 cfm to 100,000 cfm.



Type A Blower, with backwardly curved blade impeller. Both single and double inlet. Sizes from 1,000 to 70,000 cfm.



Also light duty Blowers and Blower-Filter Units for furnaces and self-contained air conditioners.

Also Propeller (Exhaust) Fans.

U. S. AIRCO Unit Coolers

Unit Coolers for cold water or direct expansion. Range of sizes.

Send for catalog showing complete line of U. S. AIRCO Equipment.

The Vilter Manufacturing Company

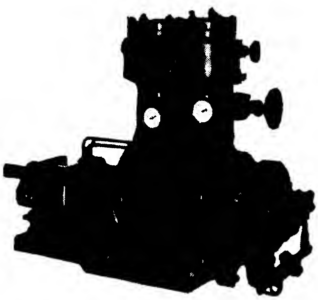
Since 1867

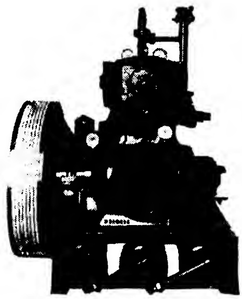
Milwaukee, Wisconsin

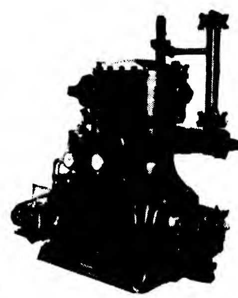
AIR CONDITIONING EQUIPMENT FOR INDUSTRIAL OR COMFORT COOLING

COMPRESSORS OF MODERN DESIGN

Ammonia **"Freon 12"** **Methyl Chloride**







Compressors by Vilter are the result of nearly seventy years of research, development and experience gained through thousands of installations in sizes from one ton to several hundred tons. These installations cover a range of applications from the most simple comfort cooling installation to the exacting requirements of temperature and humidity control necessary to the successful operation of certain industries.

The Vilter "Freon 12" Compressors embody many outstanding new features two of which are of major importance and exclusive—a new, Vilter developed, shaft seal which prevents leakage, and the elimination of certain friction factors which result in extremely low relative horsepower per ton.

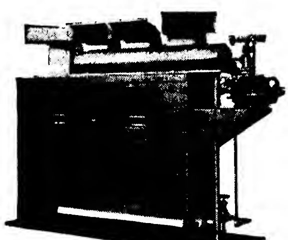
Vilter Air Washers—designed for industrial air conditioning. Positive control of humidity, temperature, and circulation of air. Automatic or hand operated. Eliminators are incorporated for entrained moisture removal. Odors and dust are removed by water sprays. No filter replacement. Low cost operation. Good for low or high temperature cooling.

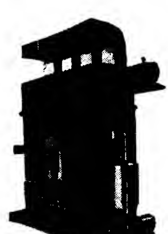
Vilter Mono-Unit Air Conditioners—are built in a complete range of sizes and types from small ceiling units to large floor units.

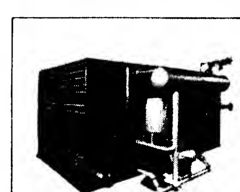
For refrigeration and air conditioning projects Vilter can supply the equipment.

UNIT AIR CONDITIONERS

Dry Coil **Spray Type** **Air Washer**







Bulletins describing this Vilter Equipment are available for the asking.

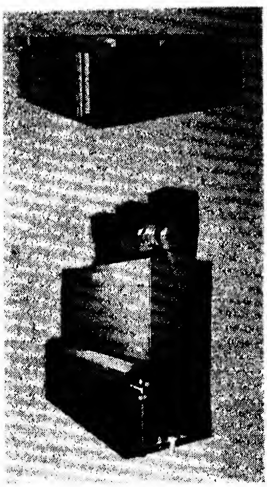
Whether the application requires a standard unit conditioner of the dry coil or spray type or a large central system of the Washer type, Vilter Equipment of sturdy construction and ample capacity is available.

York Ice Machinery Corporation

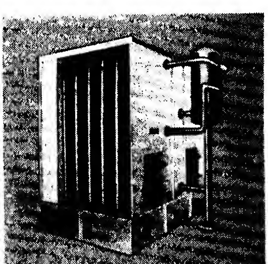
York, Pennsylvania

Factory Branches and Distributor Engineering and Sales Offices throughout the World.

Air Conditioning and Refrigeration for maintaining proper atmospheric conditions for human comfort and industrial processes. Installations of unit and central systems in a complete range of capacities and types for every design requirement.



York Air Conditioners



York Standard Dehumidifier with Coils

York Water Cooling System

Air Conditioning Units: A complete line of finned coil, dry coil wetted surface and spray type sectional air conditioners for horizontal or vertical applications, designed to facilitate installation and the distribution of air. Standard units can be equipped with by-pass feature and arranged for cooling and dehumidifying, heating and humidifying, for year-round comfort.

Finned Coils For installations where capacities or system arrangement prevent the use of standard York air conditioning units, York Fin Sections are especially applicable. Corrugated fin plates are pressed on seamless copper tubes, metal bonded and hot-dip coated, construction being identical with the heating and cooling surfaces manufactured for human comfort air conditioners.

Dehumidifiers—With or Without Coils. For central station systems where a large volume of air is to be handled and where control of humidity is an essential requirement, the York dehumidifier is especially applicable. Construction features insure a minimum space demand and maximum performance conditions. Standard washers, and those equipped with direct expansion coils, are available in a full range of capacities for human comfort or industrial installation. Air washers can be furnished also for use as indoor condensing water cooling towers when specified.

The York Economizer—A combined forced-draft cooling tower and refrigerant condenser, is avilable for installations where prohibitive water costs or inadequate drainage facilities preclude the use of a water cooled condenser. Standard factory constructed and built-up units may be used singly or in multiple for applications of any specified capacity. Economizers for use with the refrigerant Freon-12 are furnished, as standard, with a liquid sub-cooling coil.

Condensing and Water Cooling Systems—Standard systems are available for every application requirement up to 1,000 hp capacity using a single compressor. Self-contained units up to 150 hp feature the YORK line. These units are furnished with water cooled condensers or without condensers for economizer applications.

Automatic or manual capacity reduction by-pass valves can be provided for economical operation at reduced load.

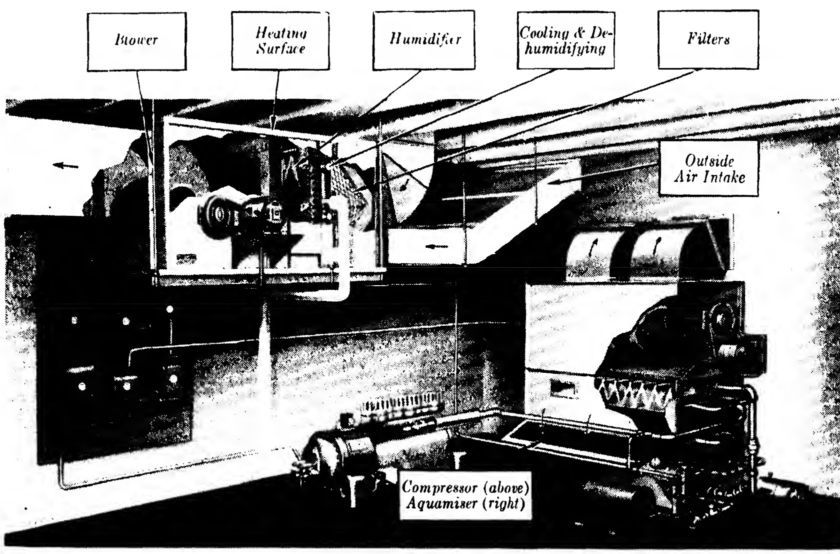
All materials and manufacturing methods employed in the construction of the York Freon-12 Condensing and Water Cooling Systems conform to the high standards and efficient operating characteristics of all YORK products and carry performance guarantees based on Refrigeration Manufacturers' Association and Air Conditioning Manufacturers' Association ratings.

Westinghouse Electric & Manufacturing Co.

Springfield, Mass.

Sales, engineering and service available through Authorized Engineering Contractors in all principal cities

UNIFIED Air Conditioning Equipment for Commercial Building Applications



APPLICATION—Westinghouse equipment provides air conditioning for every building application. Every unit is engineered by Westinghouse and matched in capacity and performance to operate in a Unified System. Equipment is applied in carefully engineered installations, with responsible maintenance service to assure continuing efficiency.

FEATURES

CONDENSING UNITS AND COMPRESSORS—Available in capacities from 1 to 100 tons. Exclusive Westinghouse Hermetically-sealed construction—no shaft seals, belts, pulleys or other visible moving parts. Compact and light in weight, they require unusually small floor space and may be installed without special foundations.

HEAT TRANSFER SURFACES—Cooling surfaces for Freon or cold water—heating surfaces for steam, hot water or vapor, in sizes to match all other Westinghouse equipment. Tubes and fins of pure Lake copper, soldered together for permanent contact and efficiency. All joints in Freon surfaces are silver-soldered.

AIR CONDITIONING UNITS—Horizontal and vertical types in sizes to match all Westinghouse compressors, cooling and/or heating surfaces. These units incorporate blower, heat transfer surfaces, humidifiers, dehumidifiers and filters.

EVAPORATIVE CONDENSERS—Westinghouse Aquamisers, combining the performance of water-cooled condenser, air-cooled condenser and cooling tower in a single compact unit that reduces water consumption by 95 to 98 per cent.

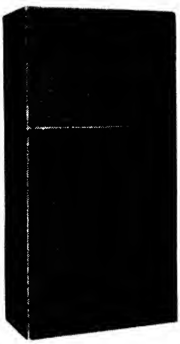
ELECTRICAL EQUIPMENT—Starters, circuit breakers, switches, conduit, etc., for every requirement.

HOW TO SELECT— Fit equipment to meet the total Btu load according to capacities under local conditions as listed on Westinghouse performance data sheets. Consult your Westinghouse Engineering Contractor or write Westinghouse, Air Conditioning, Dept. 9004, Springfield, Mass., for data sheets.

WHERE TO BUY—Consult classified telephone directory or nearest Westinghouse district office for name of Authorized Engineering Contractor.

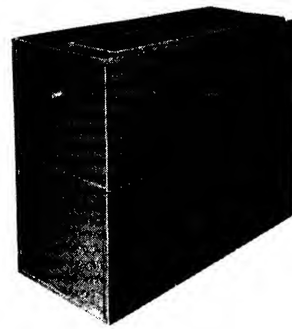
SELF-CONTAINED UNITS

Capacities from 1 to 15 tons refrigeration



Compact, attractive self-contained units from 1 to 6 tons capacity, for location directly in space to be air-conditioned.

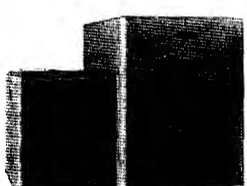
Water-cooled Mobilaire room cooling unit (above) Air-cooled Mobilaire (below)



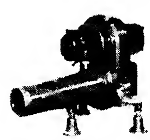
Complete, self-contained unit of 15 tons capacity, only 6 ft 6 in. long by 5 ft 4 in. high and 2 ft 10 in. wide, for use with duct system.

HOME HEATING AND AIR CONDITIONING

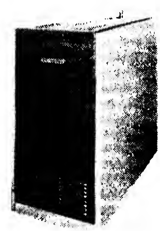
A complete line of attractive, efficient heating and air conditioning equipment including gravity warm air furnaces (pipes and pipeless types), coal-fired air conditioning units, boiler-burner units, conversion oil burners, oil and gas-fired automatic air conditioning units, oil and gas-fired gravity warm air furnaces, blower-filter units, Mobilaire room coolers and summer cooling units for use with duct systems. Full range of capacities meets all residential application requirements.



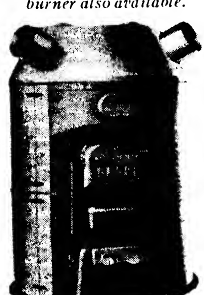
Series 1100 oil-fired winter air conditioning unit, capacities from 101,000 to 331,000 Btu at register.



Westinghouse Spiralaire conversion oil burner, capacities to 5 gph. Rotary wall burner also available.



Boiler-burner unit for steam or hot water systems, oil or gas-fired.



Steel and cast-iron gravity warm air furnaces in a total of 22 sizes.

RU-90 summer cooling unit for use with winter air conditioning units to form a year-round system.

Worthington Pump and Machinery Corporation

CARBONDALE

Carbondale Division

WORTHINGTON

General Offices: HARRISON, NEW JERSEY

ATLANTA	CINCINNATI	DETROIT	LOS ANGELES	PITTSBURGH	SEATTLE
BOSTON	CLEVELAND	EL PASO	NEW ORLEANS	ST. LOUIS	TULSA
BUFFALO	DALLAS	HOUSTON	NEW YORK	ST. PAUL	
CHICAGO	DENVER	KANSAS CITY	PHILADELPHIA	SAN FRANCISCO	WASHINGTON

Representatives in Principal Cities of Foreign Countries CA9-1

REFRIGERATION SYSTEMS FOR AIR CONDITIONING IN COMFORT COOLING OR INDUSTRIAL PROCESS

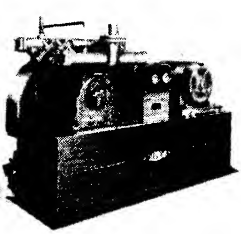
Complete refrigerating systems for use with "Freon-12", Methyl Chloride, Ammonia, or Carbon Dioxide, either direct-expansion of water cooling applications. A complete line or refrigeration compressors, permitting impartial recommendations. A nation-wide organization of Dealer-Distributors in major cities to provide sales and engineering service and plan complete air conditioning systems of the central or unit type. Architects, Engineers, and Contractors are invited to consult with us. Write to Harrison, N. J., or any branch office, for bulletins on these products.

Small Self-contained Units

"Freon-12" or methyl chloride commercial units; motors up to 25 hp; ratings up to 25 tons.

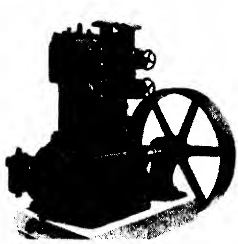
Overall Dimensions
Smallest: 22 in. long, 16 in. wide, 17 in. high
Largest: 98 in. long, 40 in. wide, 41 in. high

Medium Self-contained Units



"Freon-12" or methyl chloride units, equipped with capacity control; patented Feather Valves; motors of 30, 40 and 50 hp. Eight-cylinder units; motors of 75, 100 and 125 hp.

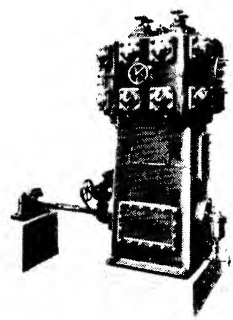
Vertical Ammonia Compressors



Pressure-lubricated; roller main bearings; safety heads; patented Feather Valves; belt drive, or direct-connected to electric motor, Diesel or gas engine; ratings from 2 to 160 tons in one unit.

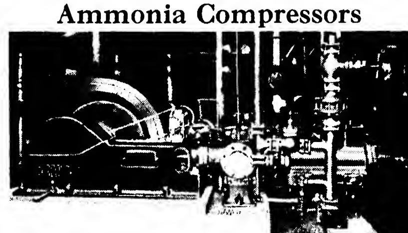
Overall Dimensions (Without Motor)
Smallest: 2 ft 3 in. long, 2 ft ½ in. wide, 3 ft 3¼ in. high
Largest: 6 ft 7½ in. long, 5 ft wide, 7 ft 7½ in. high

Vertical Duplex Double-acting Compressors



"Freon-12" or ammonia; large tonnage compressors; force-feed lubrication; roller main bearings. Crankcase sealed from cylinders, preventing contamination of oil by refrigerant. Equipped with patented Feather Valves; automatic capacity control features. Crosshead incorporated in enclosed crankcase.

Horizontal Ammonia Compressors



Single and duplex; single-stage and two-stage; belt drive, or direct-connected to electric motor, Diesel, gas or steam engine; patented Feather Valves; ratings from 60 to 750 tons. Automatic capacity control features are easily applied. Space requirements vary greatly depending upon type and drive. Compressor requirements above 250 tons are best met with these machines.

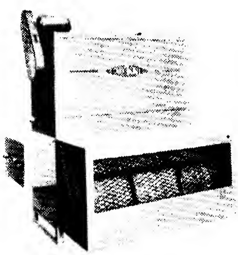
Worthington Pump and Machinery Corporation, Carbondale Division

Carbon Dioxide Compressors

A series of convenient types and sizes for every requirement is available.

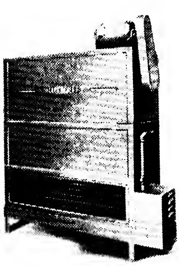
Air Conditioning Units

For Direct Expansion or Water Circulation



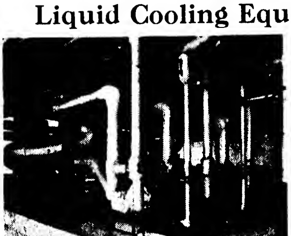
Vertical and horizontal; 500 to 15,000 cfm; large air passages; slow speed, quiet rugged fans; heavy welded steel frames; separable sections; readily accessible. The design permits a wide degree of flexibility in installation arrangements.

Shower Condensers



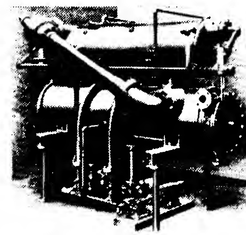
A combined condenser, receiver, and modified cooling tower, in one assembly, for "Freon-12" or methyl chloride systems; 5 to 50 tons refrigeration; built in separable sections; all parts easily accessible. Saves 90 to 95 per cent in cost of water.

Liquid Cooling Equipment



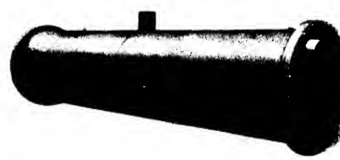
Various designs of horizontal single and multi-pass types, for a wide range of services; also vertical types. Chillers for oil dewaxing. Single and double-pipe for milk, wort, chemicals, etc. Cold liquid circulating systems.

Steam Jet Water Cooling Systems



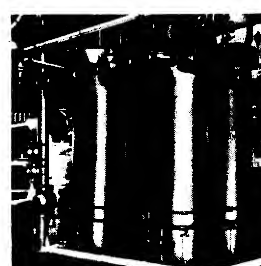
15 to 600 tons. Either surface or barometric condenser may be furnished.

Horizontal Condensers



Atmospheric drip type, for warm corrosive waters. Double-pipe for closed systems, can be retubed without shutting down. Multi-pass for closed systems and space saving.

Vertical Condensers



The Carbondale "Spira-Flo" possesses the advantages of both the atmospheric and double-pipe types. Usable with any kind of water. Compact; installed indoors, outdoors, or on the roof. Easily cleaned.

Miscellaneous

High and low side equipment for every purpose:

Coils	Valves
Air Coolers	Purgers
Separators	Tanks
Receivers	Traps
Controls	Connections
Pumps	Fittings

Space requirements and other data on request.

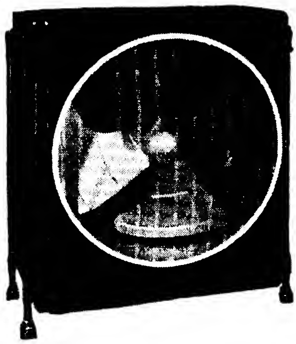
Air Controls, Inc.

Div. of

The Cleveland Heater Co.

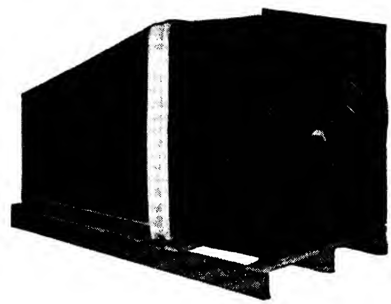
1933 West 114th Street, **Cleveland, Ohio**

Manufacturers of REX-AIRATE Air Circulators, REX AIR-PAK Blower Filter Units, REX A C Blowers and Blower Parts.



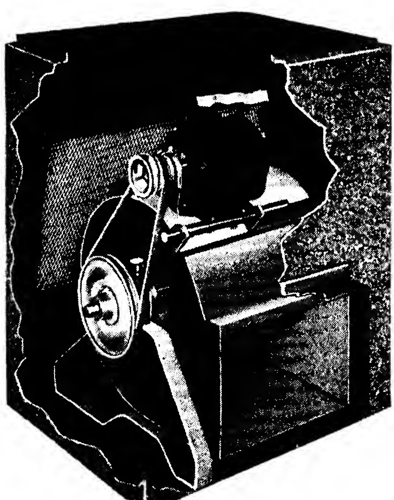
REX-AIRATE VENTILATORS

REX-AIRATE Summer Comfort Cooling units are furnished in a wide range of styles and sizes for every type of commercial cooling and ventilating. Quiet and economical in operation. Model S shown, stands 42 in. high. Capacity: 7,000 cfm.



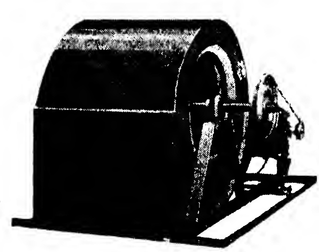
REX-AIRATE HOME COOLER

A highly effective home cooling unit by means of attic ventilation. Complete (no extras to buy) with automatic ceiling shutters, ceiling moulding, metal vent-box and canvas connector. Available in sizes for every type home.



REX-AIR-PAK PACKAGE FURNACE BLOWER

Model JR-100– A complete package blower filter unit for the small home owner and home renter. Easily detached from one furnace and quickly installed to another. New streamlined cabinet. Four-speed pulley provides various air deliveries from 650 to 1000 cfm. Available in many colors. Other models in larger sizes.



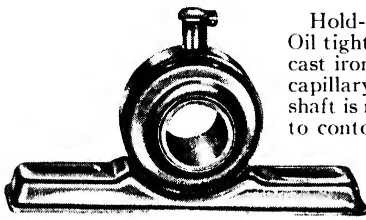
REX A C Blower parts such as blower wheels, housings, bearings, four-speed, etc., can be furnished to furnace manufacturers and sheet metal contractors.

Write us for bulletins describing REX equipment illustrated on this page.

The Lau Blower Company

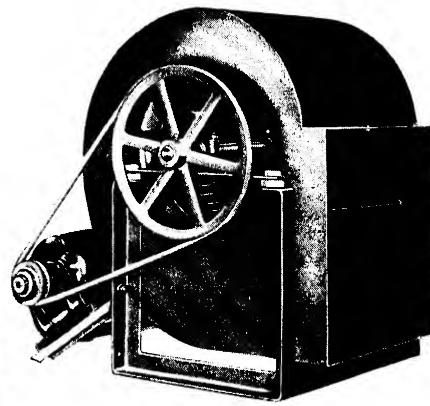
Monument Ave. at Barney, **Dayton, Ohio**

Manufacturers of Furnace Blowers, Blower Wheels, Housings, Pulleys, Pillow Blocks and Complete Assemblies



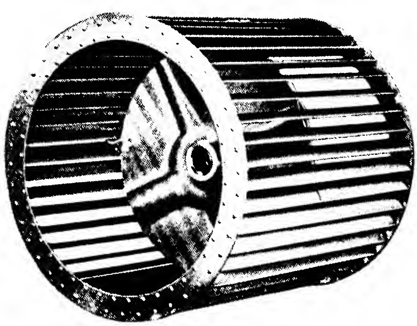
Self-Aligning Pillow Blocks

Hold-down bolts cannot affect the freedom of the bearing. Oil tight steel housing; reservoir holds twice as much oil as cast iron housings; Durex bushing feeds oil to the shaft by capillary action and maintains a constant oil film even when shaft is not rotating. Spherical surface of bearing conforms to contour of housing, providing a universal joint action.



100 Series Assembly

A complete blower assembly for manufacturers who fabricate their own casings (also available with top motor mounting) 8-sizes; variable speed drive; automatic belt-tightening device; automatic cut-out on motor.



Blower Wheels and Housings

Squirrel cage, forward curve, multi-blade type wheels. Double inlet, double width (or single inlet, single width). Dynamically balanced; sizes $4\frac{1}{2}$-in. to 25-in. Blower housings available in 10 standard sizes—special sizes on request.

700 Series Package Unit Blower

Steps-up efficiency of coal, gas and oil-fired furnaces. Complete with filters, blower cabinet, variable speed drive, blower and full size access doors on both sides. Motor and drive assembly are reversible . . . may be placed on the most convenient side—on the job. 8 sizes; knock-down construction.

Blo-Ette Package Unit

A blower that will remedy the many thousands of existing unsatisfactory gravity jobs at a price so low that it is well within the reach of everyone. Top motor mounting; automatic cut-out on motor; leak-proof filter frames; 16 x 25-in. filters; high speed—low pressure; large size access door. 3 sizes.

Schwitzer-Cummins Company

Indianapolis, Indiana, U. S. A.

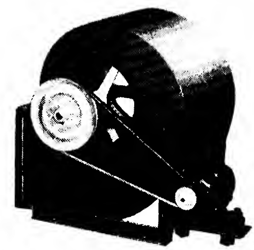
BLOWER FANS
VENTILATING FANS
HOME-VENTILATORS

HY-DUTY

BLOWER-FILTERS
HUMIDIFIERS

HY-DUTY BLOWERS

A superior line of blower fans from 4½ in. to 25 in. dia.—with or without motors and drives—available with either top or bottom horizontal, top or bottom vertical outlets—efficient, thoroughly balanced and silenced.

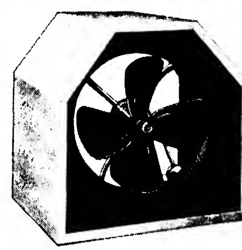


ATTIC VENTILATORS

Made with HY-DUTY propeller type fans and HY-DUTY blower fans. Models for every type installation including complicated home arrangements.

BLOWER-FILTER UNITS

STOKOLAIR, de luxe unit with AIR-MAZE all-metal filter and summer comfort damper door, and HY-DUTY models for the competitive priced field. Six models—many useful exclusive features.



BLOWER-FILTER HUMIDIFIERS

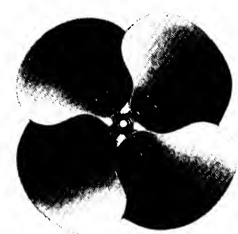
An advanced design embodying a dual system of humidity control and all-metal indestructible filters.

HY-DUTY BLOWER WHEELS

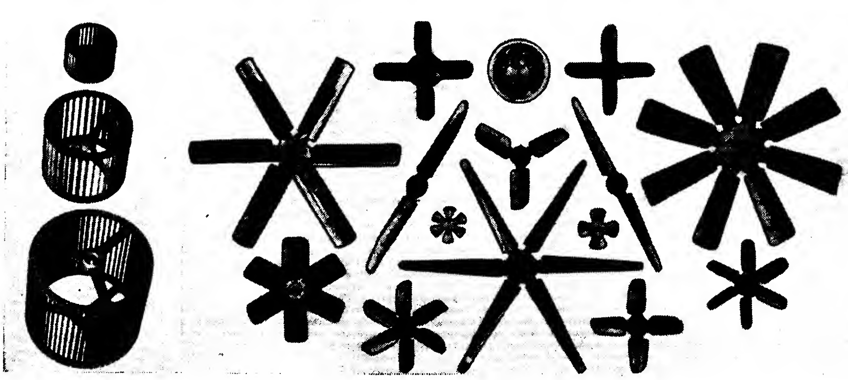
Multi-blade type, diameters from 4 in. to 50 in., furnished in single inlet and double inlet types.

KIDNEY BLADE TYPE FANS

Quiet and efficient. A full range of sizes.



Blower Wheels and Propeller Fans for Every Purpose



(See also Page 1111)

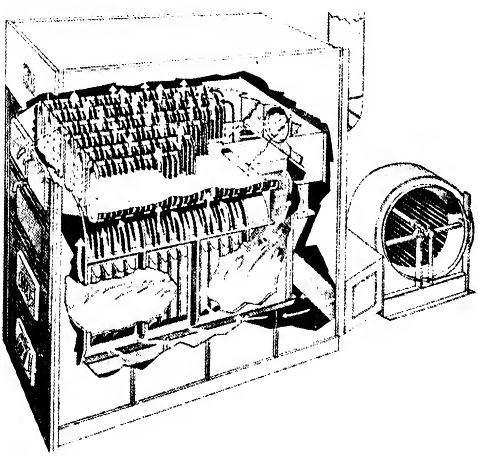
Acme Heating & Ventilating Co., Inc.

4224 Lowe Avenue, **Chicago, Ill.**

THE ACME HEATER—"It's in the Fins"

The Acme Heater has been designed by experienced heating engineers, and is constructed by expert craftsmen. It combines the best practice in the design of direct transmission heaters, with marked improvements resulting from practical experience.

Burns Any Kind of Fuel: A direct-transmission heater, such as the Acme, is not dependent upon the kind of fuel used—any type of fuel may be burned. Suitable grates may be provided so that bituminous, semi-bituminous, anthracite coal, or other solid fuels may be used with equal efficiency. Replacement of grates and linings by proper refractory material permits the use of automatic stokers or oil burning equipment.



Phantom View of Acme Heater Showing Flow of Gases and Air Travel.

Physical Data—Large Series

Size No.	Dimensions Lgth	Width	Ht	Grate sq ft	Heat Surf. sq ft	Free Area sq ft Min.	Free Area sq ft Max.	Wt. Lb	Max. Capacity Btu
7	6′-6″	4′-0″	7′-0″	10.31	260	6.55	10.25	5900	900,000
8	8′-1″	4′-0″	7′-0″	11.91	340	7.73	12.50	7000	1,100,000
9	9′-8″	4′-0″	7′-0″	13.06	430	8.91	14.75	8000	1,300,000
10	11′-3″	4′-0″	7′-0″	14.43	500	15.82	22.62	9300	1,500,000

Junior Series

2	4′-6″	3′-6″	5′-8″	3.9	136	4.7	4.7	3200	350,000
3	6′-0″	3′-6″	5′-8″	6.1	183	5.9	6.9	4800	527,000
4	7′-6″	3′-6″	5′-8″	7.2	230	7.1	9.1	5000	634,000
5	9′-0″	3′-6″	5′-8″	9.3	280	8.3	11.3	6000	800,000

Note: For automatic firing add 10% to ratings given.

Large Combustion Chamber: Provides ample space for ignition of gases of combustion, regardless of kind of fuel used. The unusually large combustion chamber, acting as "primary" heating surface, effects efficient transfer of heat, because of the great temperature difference between the burning gases inside the chamber and the air passing over the outside surface.

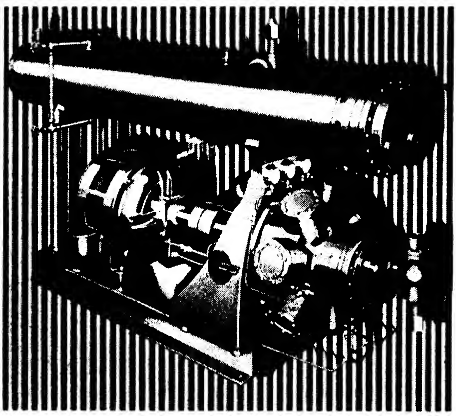
Efficient Radiator Section: Although the heating surface of the combustion chamber is large and efficient, still more heat must be extracted to obtain satisfactory overall efficiency. The "phantom view" as shown reveals how the gases of combustion enter the rear smoke chamber, flow to the front of the heater, and return again to the smoke-box. The gases are held in intimate contact with the heating surface, six times the length of the heater, before they are permitted to escape.

High Ratio of Heating Surface to Grate Area: The radiator tubes are covered with extended surfaces, or fins, typical of those used on indirect heating coils. The long, oval tubes of the radiator provide an exceptionally large heating surface which, combined with the surface of the combustion chamber, affords a remarkably high ratio of heating surface to grate area.

Balanced Construction of the Acme Heater provides ample free area and allows proper velocity of the air to be heated. Moreover, this air is brought into direct contact with as much heating surface as possible, resulting in the Acme of Efficiency.

AIRTEMP
DIVISION OF
CHRYSLER CORPORATION • DAYTON, OHIO

CHRYSLER'S AIRTEMP RADIAL COMPRESSOR



Sizes range from 10 Hp to 75 Hp capacity for individual unit or multiple installation. In self-contained Condensing Unit models for city water or cooling tower application or Compressor Unit models for use with evaporative type condensers.

Engineered Especially For Air Conditioning.

- Practically no vibration . . . an inherent characteristic of radial design . . . all parts balanced.
- Direct connected . . . no belts, no flywheel. Operates at standard motor speeds.
- Economical—high volumetric efficiency, large gas passages, large valves and ports, and low friction insures low operating costs at all loads.
- Forced-feed lubrication for smooth running and long life.
- Automatic starting unloader permits starting without load.
- Automatic capacity regulator keeps machine constantly balanced to varying load requirements while running at constant speed . . . economical. Operates at peak efficiency under all loads.
- Shipped completely assembled . . . easily installed.

Write for Bulletin No. L-336.

AIRTEMP "ALL-IN-ONE" AIR CONDITIONER

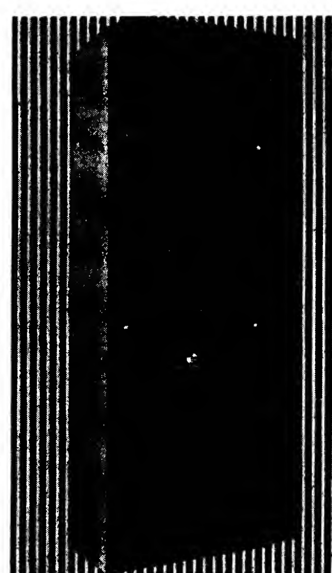
The Airtemp "All-In-One" is a complete system, all in one cabinet . . . cools and dehumidifies, filters and circulates, with positive ventilation; for free discharge or duct distribution of the conditioned air. Heating coil and humidifier can be furnished for connection to existing heating systems.

Compact—Entirely self-contained and all working parts compactly enclosed within a finished metal cabinet. Two sizes and capacities for single or multiple installation meet all requirements for dependable low-cost air conditioning in stores, shops, offices etc.

Model No.	Hp Cap.	Free Air Delivery cfm	Approx. Dimens. (inches)		
			L	W	H
3 SC	3	1200	33	20	89½
5 SC	5	2000	34	20	97½

Efficient—Unusually quiet, trouble-free operation assured because of the Airtemp hermetically-sealed Radial Compressor which is suspended from single rubber mounting.

Readily Installed—Completely assembled and factory tested . . . delivered ready for installation. **Approved by Underwriters Laboratories, Inc.**



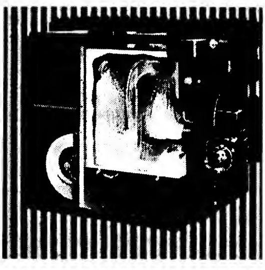
AIRTEMP

DIVISION OF

CHRYSLER CORPORATION • DAYTON, OHIO

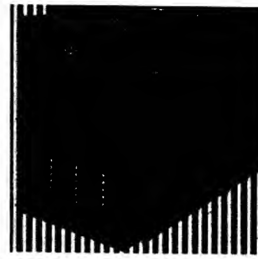
AIRTEMP DIRECT-FIRED WINTER AIR CONDITIONERS

Quality and Dependability at Low Cost



Cut-away view of model O-125 Winter Air Conditioner.

A complete line of compact, dependable winter air conditioners that heat, humidify, filter and circulate the air at cost of heating alone. These units are designed around a one-piece copper bearing steel furnace body that is gas tight and compactly enclosed in an insulated finished metal cabinet. Light in weight, economical in floor space, with either oil or gas fired models.



Model O-125 Winter Air Conditioner with enclosure Jacket.

OIL BURNING MODELS

Model No.	Btu/hr. Output* Rating at Bonnet	Floor Dimens. (inches) L	Floor Dimens. (inches) W	Approx. Net Weight Lb
0-125	100,000	40	28	438
0-163	130,000	45¼	35	625
0-200	160,000	49¼	37	710
0-250	200,000	55	41	848

*Nominal Rating 80 per cent Efficiency.

GAS BURNING MODELS

Model No.	*AGA* Rating Input* Btu/hr-	Floor Dimens. (inches) L	Floor Dimens. (inches) W	Approx. Net Weight Lb
G- 87	87,000	18	26	310
G-125	125,000	40	28	418
G-163	163,000	45¼	35	605
G-200	200,000	49¼	37	685
G-250	250,000	55	41	828

*Nominal Output 80 per cent of Input.

AIRTEMP BOILERS

Maximum efficiency and low operating costs are built into Airtemp Boilers. Matched coordinated design of boiler and burner, ample heating surface to insure full rated capacity and quick heating, fire chamber surrounded on all sides including bottom by water sections and jacket insulated with rigid sheet asbestos surfaced with reflective aluminum foil.

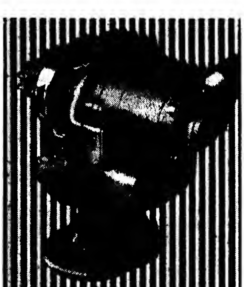
OIL BURNING

Steam Boiler Model No.	Water Boiler Model No.	Installed* Sq Ft Radiation Steam	Installed* Sq Ft Radiation Water
OS-2	OW-2	280	445
OS-3	OW-3	490	735
OS-4	OW-4	665	1000
OS-5	OW-5	840	1260
OS-6	OW-6	1010	1540
OS-7	OW-7	1200	1780

GAS BURNING

Steam Boiler Model No.	Water Boiler Model No.	Installed* Sq Ft Radiation Steam	Installed* Sq Ft Radiation Water
GS-2	GW-2	280	445
GS-3	GW-3	438	700
GS-4	GW-4	585	935
GS-5	GW-5	735	1175

*Installed Rad. calculated equal to 70 per cent total EDR.



AIRTEMP OIL BURNERS

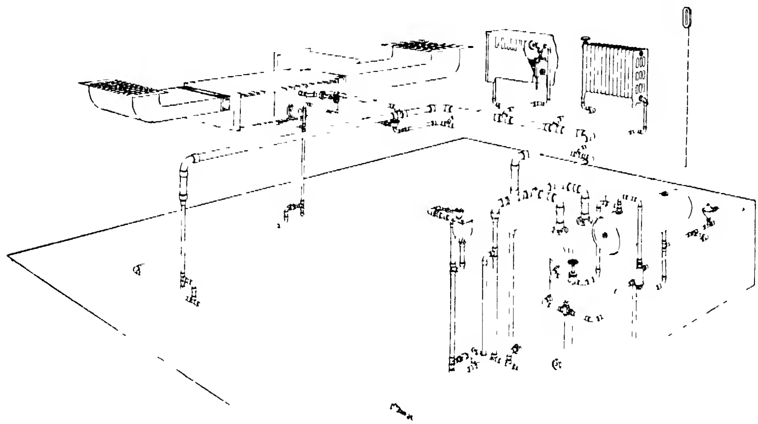
A high pressure, atomizer type burner for use in Winter Air Conditioners, Boilers, or Conversion work. Burns No. 3 fuel oil with adjustable oil pressures and air quantities. The quality and dependability of this burner is unique because of the High Velocity Blender which properly mixes the air and oil to give complete combustion, coupled with a Focused Flame tailored to fit the combustion chamber. Automatic fuel control insures smooth, quiet operation and control of oil to the last drop at nozzle tip. Long Life flexible Duprene coupling minimizes wear on pump and motor. Available in four sizes ranging from 1 to 9.5 gallons of oil per hour.

AMERICAN RADIATOR COMPANY

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

40 West 40th Street, New York, N. Y.

NEW AMERICAN RADIATOR CONDITIONING SYSTEMS



MODEL K
controlled warmth
system

CONTROLLED WARMTH SYSTEMS

The Arco Model K Vapo-Orifice System is an engineered and coordinated unit which operates to insure equal distribution of warmth On each radiator there is installed a special inlet valve with an adjustable orifice, by means of which flow of steam to each radiator can be accurately calibrated in accordance with the capacity of the radiator. The adjustment can be made while the system is in operation

Pressure is so controlled that no more steam is admitted to the radiators than they are capable of condensing Therefore no thermostatic traps are required on the return connections of the radiators A thermostatic air eliminator is provided If the pressure, through inadvertence, should exceed the design pressure, steam will enter the return lines, close the vent port of the air eliminator, and the system will automatically equalize itself, insuring the return of condensate to the boiler.

When a zone system is used, the zone valves prevent the self-equalizing action and it is necessary to install an Arco Condensation Pump to insure the return of condensate to the boiler

Simplicity of design and moderate cost make possible the installation of this system for little more than the cost of a One Pipe Steam System It can be used in any size building and has been installed in buildings requiring up to 60,000 square feet of radiation

"Standard Specifications" and "Design and Installation Guide" including all details are available at American Radiator Company branch offices

USED WITH MODEL K SYSTEM

Arco No. 900 Adjustable Orifice Valve—Orifice may be adjusted permanently to "meter" the correct amount of vapor to each radiator according to its capacity, to effect balanced heating

AMERICAN RADIATOR COMPANY

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

40 West 40th Street, New York, N. Y.

NEW AMERICAN RADIATOR CONDITIONING SYSTEMS

In this new kind of home conditioning presented by American Radiator Company, the heating operates independently of the other functions of air conditioning. This permits operation of the conditioning even when the heating is off, or of heating alone when no conditioning is necessary. It simplifies duct work, too, since ducts do not carry the heating load.

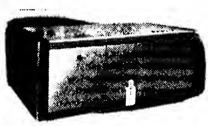
The illustration on page 898 shows a typical system. The Conditioning Unit is suspended from the ceiling of the basement on rubber dampers. Air is filtered as it enters, then brought to a comfortable temperature by tempering coils. A spray humidifier provides correct moisture content. A Sirocco Blower in the unit silently forces the conditioned air through the house.

A radiator system—steam, hot water or vapor—provides heat. There are new controls, new valves, new vents which improve heat distribution. Arco Pipe and Fittings of pure wrought copper connect boiler, radiators and hot water supply system. Because of low conductivity they cut down heat loss; they are rustproof, leakproof, and corrosion resisting.

The system is simple to install. There is a minimum of duct work . . . outlets and return grilles being generally recommended only for the first floor, in as few rooms as is advisable. Introduced even at only one point, the conditioned air will naturally permeate the entire house.

Another advantage of the system is in modernization work as the air conditioning unit is easily added to existing radiator systems.

ARCO AIR CONDITIONERS

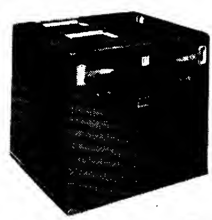


ARCO AIR CONDITIONERS 101-B, 201-B

Designed to be part of American Radiator Conditioning Systems, Arco Air Conditioners add Fresh Air Ventilation, Humidification, Air Cleaning, Circulation to any radiator heated home or small building. Needing only simplified air mains, Arco Air Conditioners are especially suitable for modernization work.

ARCO AIR FILTER

Replacement type. Viscous coated. Provides 90 deg change in air flow. Available in many sizes.

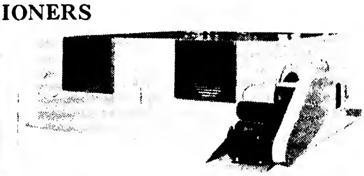


ARCO AIR CONDITIONER 1101-B

Combines in one jacket No. 11 Oil Burning Boiler, Arco Air Conditioner and Taco Heater for domestic hot water.

ARCO AIR CONDITIONER 301-B

For larger homes—provides heat as well as air conditioning in conjunction with steam or hot water boiler.



ARCO AIR CONDITIONER SERIES "RH"

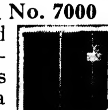
The Arco Conditioner, Series "RH," is a quiet, compact and economical air conditioning unit for residential and smaller commercial buildings. The inherent simplicity and advanced features of design result in a unit flexible in application and constructed of the best materials which will deliver the benefits of air conditioning at a cost comparable to that of an ordinary heating system.

ARCO HUMIDIFIER No. 8000

Adds correct moisture to radiator heated homes. Needs only one 3 in. main. Easily connected to any boiler.

ARCO HUMIDIFIER No. 7000

A simple device, designed to be installed on the basement wall or ceiling. Blows humidified air through a short duct and outlet grille into the rooms above.



(See also American Radiator Co. pages 940-941 and Subsidiaries)

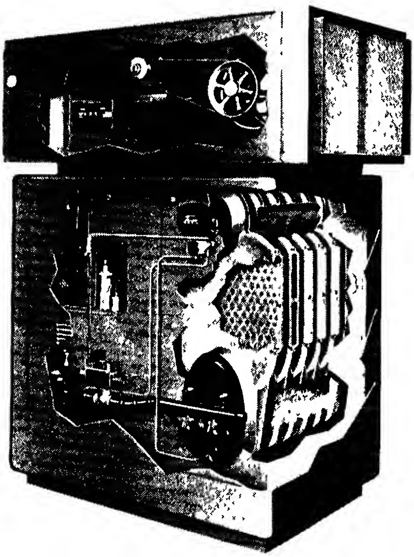
American Gas Products

Division of American Radiator Company

40 West 40th Street, **New York, N. Y.**

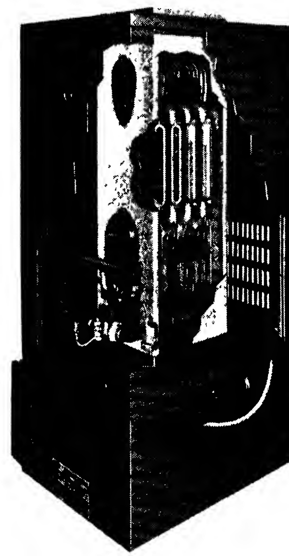
Gas Fired Air Conditioners

AGP AIR CONDITIONERS FEATURE:



Counter-Flow Type—Type 2-FE

- Space Saving Design.
- Factory Assembled Fan-Motor Unit
- Factory Assembled Heating Section of cored cast iron.
- Single Combustion Chamber
- Single Thermostatic Pilot
- Sirocco Type AC Fan
- High Efficiency
- Quiet Operation



Parallel-Flow Type—Type 1-FP

RATINGS—DIMENSIONS—DATA

Conditioner Number	AGP Guaranteed Ratings: AGA Rating: Input Btu/hour	AGA Rating: Output at Bonnet Btu/hour	Output at Registers Btu/hour	Conditioned Space Cu Ft	Maximum Fan Cfm at 65 F	Cfm at Discharge	Approximate Shipping Weight Lb
2-FE- 4-100	90,000	72,000	64 800	10,600	660	785	1210
2-FE- 4- 80	100,000	80,000	72,000	14,700	920	1060	1210
2-FE- 5-100	112,500	90,000	81,000	13,300	830	990	1290
2-FE- 5- 80	125,000	100,000	90,000	18,400	1150	1325	1355
2-FE- 6-100	135,000	108,000	97,200	15,800	990	1180	1355
2-FE- 6- 80	150,000	120,000	108,000	22,000	1375	1585	1420
2-FE- 8-100	180,000	144,000	129,600	21,100	1320	1570	1760
2-FE- 8- 80	200,000	160,000	144,000	29,400	1840	2120	1775
2-FE-10-100	225,000	180,000	162,000	26,600	1660	1980	2275
2-FE-10- 80	250,000	200,000	180,000	36,600	2290	2640	2315
2-FE-12-100	270,000	216,000	194,400	31,700	1980	2360	2450
2-FE-12- 80	300,000	240,000	216,000	44,000	2750	3170	2500
1-FP- 4	56,000	44,800	40,300	8,300	520	600	510
1-FP- 6	85,000	68,000	61,200	12,600	790	910	627
1-FP- 8	100,000	80,000	72,000	14,800	930	1070	813
1-FP-12	150,000	120,000	108,000	22,100	1380	1590	1013

NOTE *AGP Guaranteed Ratings* are based on thermal efficiency of not less than 80% of the rated *AGP* Btu Input. *Register Output* is based on 10% deduction for normal average heat loss in ducts *Conditioned Space* indicates the maximum cubic feet of the space which can be conditioned at rated fan CFM to insure quiet operation and adequate air circulation.

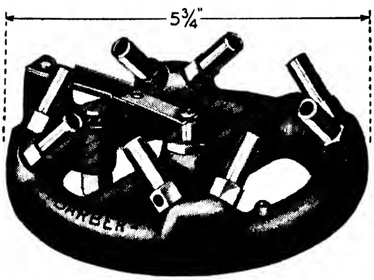
(*See also Page 951*)

The Barber Gas Burner Company

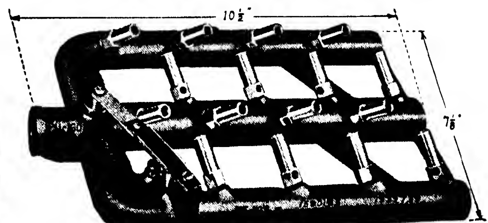
3704 Superior Ave., **Cleveland, Ohio**

Address Michigan Inquiries to THE BARBER GAS BURNER CO., OF MICH., 4475 Cass Ave., Detroit, Mich.

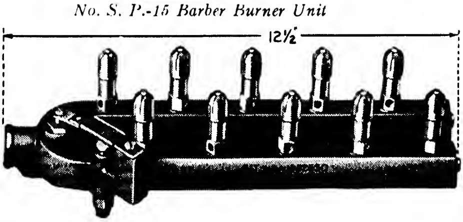
Barber Automatic Jet Gas Burners for Heating and Air Conditioning Equipment and Numerous Gas-burning Appliances have won leadership through 20 years of research and manufacture. Used for heating thousands of homes and buildings throughout this country and Canada. Adopted by leading makers of gas appliances The exclusive Barber jet principle of combustion, other basic advantages of design and numerous recent improvements, have kept Barber Burners apace with modern heating and air conditioning practice. Shown are only a few items from Barber's complete line. Write for No. 38-A catalog and price list.



No. S. P.-15 Barber Burner Unit



No. P.U.-160 Burner Unit

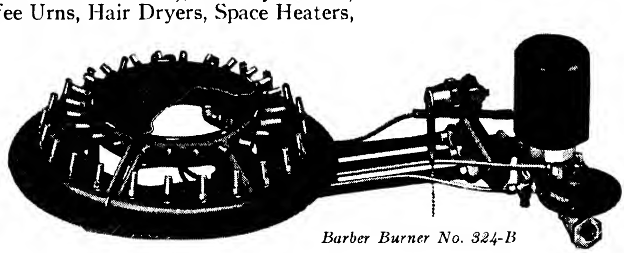


No. C. U.-90 Barber Burner with Safety Pilot

Barber Gas Pressure Regulators A.G.A. Approved

Made in the following sizes: ¼", ⅜", ½", ¾", 1", 1¼", 1½", 2".

BARBER BURNERS AND REGULATORS are Adaptable to: Air Conditioning Equipment, High Pressure Boilers (Tubular and Tubeless), Bakery Ovens, Garage Heaters, Coffee Urns, Hair Dryers, Space Heaters, Floor Furnaces, Clothes Dryers, Water Heaters, Confectioners' Stoves, Vulcanizing Machines, Pressing Machine Boilers, Japanning Ovens, Core Ovens, Banana Room Heaters, Other Appliances.



Barber Burner No. 324-B

Conversion Burner for Round Furnaces or Boilers

Round Burners are adjustable as to diameter, on the job, to fit practically all grate sizes. Also to fit grates of oblong furnaces and boilers. Supplied with Baltimore Safety Pilot. Listed in the *A.G.A.* Directory of Approved Appliances. No. 324-B equipped with automatic controls with motor gas valve. Available in "A" series with magnetic gas valve control, "S" series with quick acting gas valve control (for buildings equipped with automatic heat control), and "M" series with manual control.

Gas Burner Specialists offering Engineering Department and Laboratory facilities for Gas Burner problems.

Delco-Frigidaire Conditioning Division

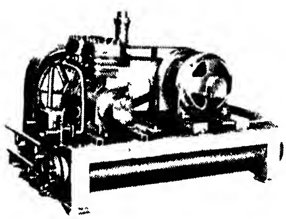
General Motors Sales Corporation

GM
GENERAL MOTORS

Dayton, Ohio

A complete line of air conditioning and automatic heating products, including: Condensing units, coils, self-contained room conditioners, self-contained store conditioners, central system units, remote units, suspended units, evaporative condensers, oil burners, coal stokers, oil and gas fired boilers, oil and gas fired winter air conditioners, split system units, automatic water heaters. Write to Delco-Frigidaire, Dayton, Ohio for latest information and detailed specifications, or consult local Delco-Frigidaire distributor whose address is listed in the classified section of your telephone directory.

CENTRAL SYSTEMS



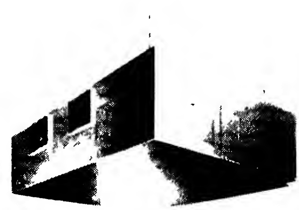
10 hp Condensing Unit

COMPRESSORS

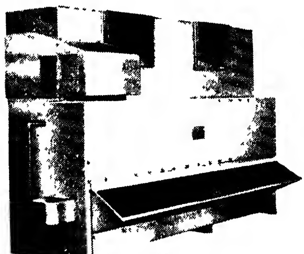
Frigidaire automotive type air conditioning Condensing Units, which produce more cooling per dollar of water cost and per kilowatt of power consumed, are available from 1/3 to 50 hp in size, for use with evaporative condensers, water towers or city water. Air cooled models available up to 3 hp

CENTRAL PLANT UNITS

Frigidaire Central-Duct type Air Conditioners make available the advantages of a properly engineered unit, built under strict factory control, to meet the requirements of central-duct air conditioning systems. Available in vertical and horizontal models with a nominal capacity range of 6-24 tons Heating coil optional for winter use.



6 Ton Central System Unit

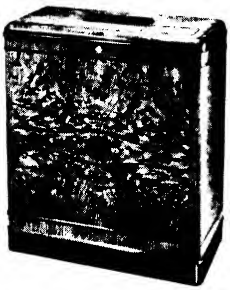


40 Ton Evaporative Condenser

EVAPORATIVE CONDENSERS

Frigidaire Evaporative Condensers are designed for use with Frigidaire air conditioning condensing units Their use reduces necessary water consumption, and makes it possible to provide economical and efficient installations in localities where water is scarce, of poor quality, available only at high rates, or where use is restricted. Available in six sizes with a nominal capacity range of 3-40 tons

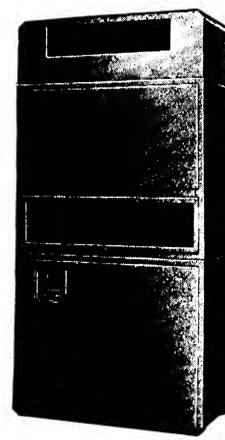
UNIT AIR CONDITIONING



RSA Room Conditioner

ROOM CONDITIONERS

Frigidaire self-contained Room Conditioners, models RSA (air cooled) and SC-81 (water cooled) are designed for single room installations. Portable model RSA powered by famous Meter-Miser mechanism; can be used year 'round for filtering, circulation and ventilating in addition to true summer air conditioning; needs only power and window connection. Model SC-81 needs power, water and drain connections. Thermostatic control furnished as standard equipment on both models. Both models nominally rated at 2/3 ton capacity.



3-Ton Store Unit

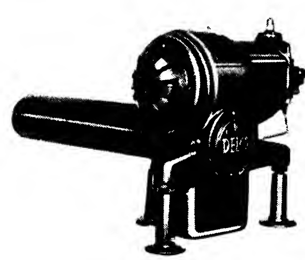
STORE CONDITIONERS

Frigidaire self-contained Store Conditioners are completely self-contained, incorporating the condensing unit, fans, coils, filters and controls within compact, attractive cabinets. Frigidaire Store Units provide cooling, dehumidification, filtering and air circulation. A heating coil may be added to furnish a positive circulation of warm filtered air for winter service, if desired. Directional grilles on the 3 and 5 ton models provide proper distribution to meet the space requirements. Models are available with nominal ratings of 3, 5 and 10 tons. These can be installed either within conditioned area (except 10 ton model), or in adjoining space. Units are ideal for short leases, as they may be removed and reinstalled in new locations in but a few hours.

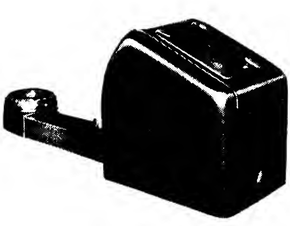
AUTOMATIC HEATING

OIL BURNERS

Delco Oil Burners employ the highly efficient pressure atomizing method of breaking the liquid fuel into fine particles for complete combustion. In the Delco Oil Burner with the Rotopower unit, all rotating parts are built as an integral unit on the motor shaft. Available in 5 sizes in standard voltage characteristics with combustion rates from 1-30 gal per hour or a capacity range from 440-12,000 sq ft of steam, EDR.



Residential Burner



Typical Stoker

COAL STOKERS

Delco Stokers are designed to provide automatic firing for coal-fired domestic heating plants. Are of underfeed, screw type with intermittent coal feed. Two 30-pound and one 50-pound stokers, burning bituminous coal, make up the line. Automatic controls, sectional tuyeres; Coal Control; automatic air control, rubber-lined corrosion-resistant hopper, hopper gas eliminator and sound insulated mechanism in deluxe models minimize the time and effort of attention and provide uninterrupted heating service.

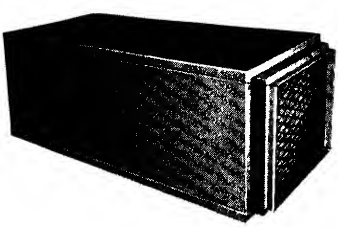
AUTOMATIC HEATING

AUTOMATIC BOILERS

Delco Automatic Boilers coordinate the Delco oil burner or Delco gas burner with a boiler of special design and construction for application on hot water, steam, or vapor-vacuum heating systems. Most oil-fired models incorporate the famous Delco Oil Burner with the Rotopower Unit Boilers are honey-combed with water-filled fins When sections are fitted together, fins form a series of passes, exposing a maximum of water-backed surface to the heated gases. Two models, the DB3 and DB4 incorporate the exclusive Quik-Action Heat Transmitter which provides quick, radiant heating and renders the previous slow heating fire clay refractory lining of the combustion chamber unnecessary Five oil fired automatic boilers, with capacities ranging from 350-1,335 sq ft of steam, EDR, and one gas fired model with a capacity of 800 sq ft of steam, EDR, are available

DB-3 Boiler

"SPLIT SYSTEMS"

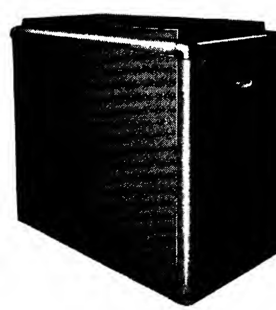


HC-20 Conditioner

Delco Residential Air Conditioning Units are furnished in two sizes and styles, the HC-20 and HC-40, and are particularly designed for use with Delco automatic boilers. Are of horizontal, suspended type and operate in conjunction with a duct system. Units are designed for four major types of applications. (*1*) "Split System" heating and humidifying units in connection with an external source of steam or hot water, together with separate radiators; (*2*) as "Indirect" heating and humidifying units for use with external source of heat, where no radiation is used, (*3*) as year 'round units, requiring external heating and cooling sources, (*4*) HC-20 may also be used as a humidifier, by substituting a tempering coil.

CONDITIONAIRS

The Delco Conditionair is a compact, completely automatic unit, oil or gas fired, which provides true winter air conditioning by circulating cleaned, humidified and properly heated air Model DAO Delco oil Conditionair incorporates the new, exclusive Quik-Action Heat Transmitter Air flow resistance reduced to a minimum by tear drop design, heat transferred to the flowing air from a large heating surface, dotted with heat projectors, moisture then added by pan, cascade or spray type humidifier. Cooling attachment can easily be added for year 'round use In Delco gas conditionairs there is a sufficient range in sizes to permit selection of the proper unit for applications ranging from a small six room house (85,000 Btu heat loss or less) to a large mansion (255,000 Btu heat loss or less). Delco Conditionairs, oil fired, range in size from 85,000 Btu heat loss to 280,000 Btu heat loss



Small Conditionair

The Excelsior Steel Furnace Co.

118 S. Clinton Street
Chicago, Ill.

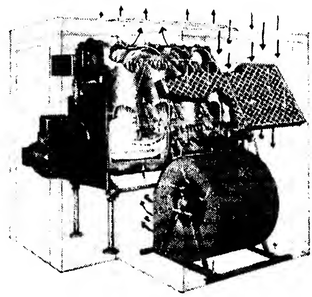
BRANCHES

BROOKLYN, N. Y. — ST. PAUL, MINN. — KANSAS CITY, MO.

EXCELSIOR WARM AIR HEATING EQUIPMENT

UNIQUE OIL BURNING AIR CONDITIONER

Cast iron Heating Element provides greater radiating surfaces with increased heating efficiency.



Made in three sizes:

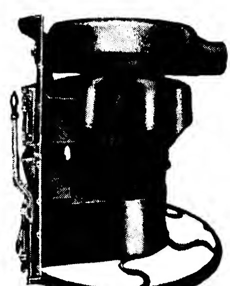
B1—121,000 Btu.
670 to 1300 cfm.

B2—141,000 Btu.
1160 to 2100 cfm.

B3—161,000 Btu.
1160 to 2100 cfm.

Unequalled for economy, heating efficiency and quietness.

EXCELSIOR WARM AIR FURNACES



ERA

Cast iron. Made in six sizes, rated 336 to 869 sq in. warm air pipe capacity.

SUPERLIFE

Chrome-alloy cast iron of similar design with 20 year guarantee.

FAMOUS

Cast iron. Made in three sizes, rated 595 to 843 sq in. warm air pipe capacity.

STEEL FURNACES

Two models ranging from 461 to 1217 sq in. warm air pipe capacity.

FAMOUS EXL-AIR

Square cased Units with blower, blower switch and automatic humidifier.

Made in three sizes, rated from 111,600 to 158,700 Btu and 670 to 2100 cfm. Standard model permits side stoker installation.

EXCELSIOR FORCED AIR DUCTS AND FITTINGS

Standardized, prefabricated ducts and fittings for forced air and air conditioning installations.

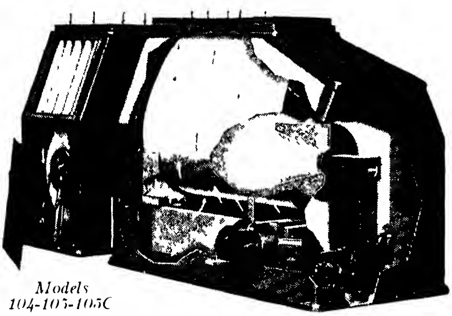
Full information on our complete line of Warm Air Heating Equipment upon request.

Gar Wood Industries, Inc.

AIR CONDITIONING DIVISION

7924 Riopelle St., **Detroit, Mich.**

Licensed Distributors in All Principal Cities



Models 104-105-105C

TEMPERED-AIRE UNIT

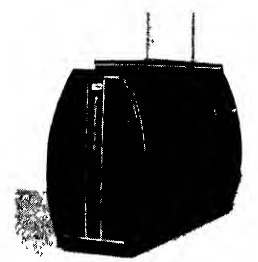
The Tempered-Aire heating and air-conditioning equipment made in six (6) capacities, consists of filters, blower, burner, humidifier, and oil furnace engineered into one compact, coordinated system

Clean, humidified, warmed air is delivered from the unit to the duct system and then uniformly distributed to all parts of the building

Air is cleaned by dry cloth filters and circulated by a multiple-blade blower fan The heating unit is a down-draft, warm-air furnace equipped with an integral pressure atomizing type oil burner The humidifier is a steam tube within the firebox, assuring simple and prompt humidification

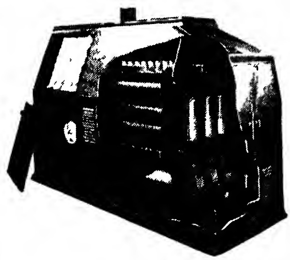
The new line of Tempered-Aire units (Models 101, 201, 301) have many improvements including the new Model "O" oil burner and new cabinet design (see cut opposite) The oil burner starts, operates and cuts off smoothly even with widely varying draft conditions Combustion is clean, quiet and efficient, due to the improved manner in which the air and oil are mixed to make a sunburst shaped flame CO_2 is exceptionally high, and even greater economy than previous models is assured.

Models No 101 201 301



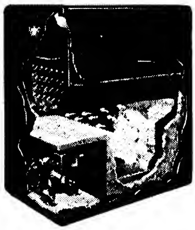
Tempered-Aire Ratings and Dimensions	No 101	No 201	No 301	No 104	No 105	No 105-C
Btu I Hour at Bonnet	100,000	135,000	200,000	225,000	300,000	400,000
Btu I Hour at Grilles	80,000	110,000	160,000	185 000	245 000	
Air Delivery CFM	1,000	1,350	2,000	2,250	3,000	4,000
Air Temperature at Bonnet	160°	160°	160°	160°	160°	160°
Oil Rate Gallons I Hour	1 00	1 35	2 00	2 25	3 00	4 00
Heating Surface, Sq Ft	54½	64	86¼	176	220	220
Filter Area, Sq Ft	30	30	40	60	77	77
Motor Horse Power-Burner	1/6	1/6	1/6	1/6	1/6	1/4
Motor Horse Power-Blower	1/5	1/5	1/3	1/2	3/4	1
Over all Length, Inches	77½	82¾	92¾	140¼	156¼	156¼
Over all Width, Inches	32	32	32	38	38	38
Over all Height, Inches	54	54	54	64	64	64

GAS FIRED TEMPERED-AIRE



Built in a single unit, it provides air filtering, blower circulation, humidifying and heating. Furnace is made of heavy gauge copper bearing steel Thermostatically operated. Safety controls give full protection. Approved by Amer. Gas Assoc.

	No 90	No 120
AGA Input, Btu per hour	135,000	180,000
AGA Output, Btu per hour	101,250	135,000
Grille Delivery Btu per hour	90,000	120,000
Air Delivery CFM	900-1425	1200-1900
Filter Surface Sq Ft	34	34
Blower Motor Size	1/5 hp	1/5hp
Blower Motor Current Consump	250 Watts	250 Watts
Overall Length, Inches	96 9/16	96 9/16
Overall Width, Inches	40	40



MODEL "R" BOILER-BURNER UNIT

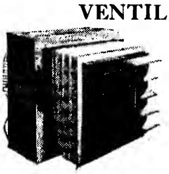
A compact firetube steam or hot water heating boiler, with an integral oil burner.

Boiler built of heavy rust-resisting boiler plate, electrically welded. The combustion chamber walls are carried clear to the crown sheet. The result is a considerably higher temperature of fire than is found in the conventional boiler, and better combustion of No. 3 oil.

Since there are no water legs, the bulk of the heating surface is concentrated in the firetubes, or secondary heating surface, resulting in maximum extraction of heat from the hot gas which leaves the firebox and consequent low stack temperature, high efficiency and operating economy.

Burner Model	R500 H	R750 H	R1000 D	R1400 D	R1800 D
Firing Rate—Gal per Hour	1.65	2.50	3.25	4.50	6.00
Maximum Net Steam Load—Sq Ft	500	750	1000	1400	1800
Maximum Net Hot Water Load—Sq Ft	800	1200	1600	2240	2880
Maximum Gross Steam Load—Sq Ft	750	1125	1500	2100	2700
Maximum Gross Hot Water Load—Sq Ft	1200	1800	2400	3360	4320
Heating Surface—SqFt	52	68	84	118	154
Overall width—Jacket	$30\frac{3}{16}$	$30\frac{3}{16}$	$37\frac{13}{16}$	$37\frac{13}{16}$	$37\frac{13}{16}$
Overall Length—Jacket	53	65	$59\frac{3}{4}$	66	80

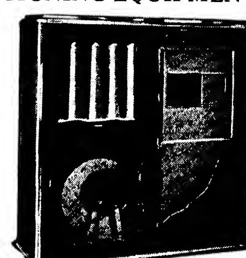
VENTILATOR



A silent operating ventilator—attic or commercial use. Ventilator consists of belt-driven propeller type fan; electric motor with overload protection; metal housing; automatic shutter; insect screen; canvas connection and clamps; suspension springs.

8 Sizes: 20 in. to 60 in. CFM: 3500 to 30,000.

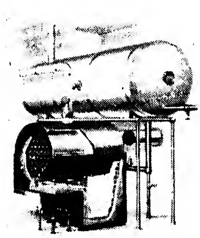
INDIRECT AIR CONDITIONING EQUIPMENT



Made in eight (8) sizes. Consists of a compact cabinet unit containing cloth air filter and vapor mist filter, blower, humidifying chamber, and indirect, blast heater.

While designed particularly for use with Gar Wood Boiler-Burner Units (steam, vapor, or hot water), the air conditioning unit may be used effectively with any adequate existing plant.

OIL-FIRED WATER HEATERS



Commercial Type—An attractive, quiet operating balanced unit, in 2 sizes. Burns No. 3 fuel oil. Fully automatic. Separate tank.

Capacities: 200 and 300 gal per hour 100 F temperature rise. Overall widths: 28¼ in.; lengths: 49 in.–61 in.

Domestic Types—Model S50—coil type heater for use with separate storage tank. Cap. 50 gal per hour 100 F temperature rise. Model S40 has storage tank integral. Cap. 40 gal per hour 100 F temperature rise. Both types have vertical rotary burner with only one moving part and for use with No. 2 or lighter oil.

Model S50

Model S40

CONVERSION OIL BURNERS

Highly efficient for installation in existing heating plants. Pressure atomizing type. Simple in design. Handle No. 3 fuel oil. Sturdy, quiet, easily accessible. Fit any shape of firebox.

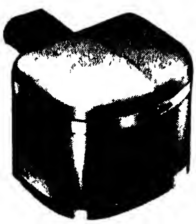
Model "W"
412-2500 sq ft net steam radiation.

Model "K"
412-2500 sq ft net steam radiation.

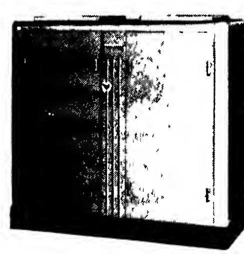
Model "H"
Up to 625 sq ft net steam radiation.

GENERAL ELECTRIC
COMPANY
AIR CONDITIONING PRODUCTS
Air Conditioning Department, Bloomfield, N. J.

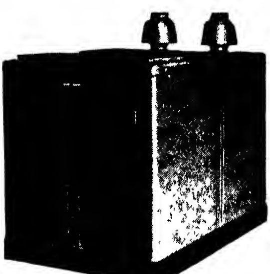
G-E Oil Furnace—Four sizes, LA-3, LA-4, LA-6 and LA-7 designed for steam, vapor, or hot water radiator systems and for indirect heating with air conditioners Boiler output, LA-3, 417 sq ft steam; LA-4, 555 sq ft steam; LA-6, 665 sq ft steam, LA-7, 1,250 sq ft steam. Steel boilers constructed in accordance *A S M E* boiler code Approved by Underwriters' Laboratory, Inc. Fully coordinated-boiler, burner, domestic hot water, controls in one enclosed unit made and guaranteed by General Electric. Low standby losses Automatic Night and Day temperature control available with all units. G-E Water Circulator available with hot water models.



G-E Oil Burner- Conversion unit for existing or built-up heating systems Consists of motor-compressor, fan, burner head, and controls enclosed in an attractive gray metal jacket. Flexibility makes it applicable to varying sizes of boilers and furnaces Tailored flame to conform to fire-box giving maximum efficiency. Models available for steam, vapor, hot water, and warm air systems Easily installed and serviced Oil rates from $\frac{3}{4}$ to 3 gallons per hour



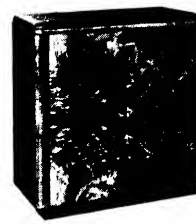
G-E Warm Air Conditioner, Oil Fired—Three sizes, LB-3, LB-4 and LB-6 consisting of combustion-heat transfer unit, centrifugal fan, humidifier, filters, controls, and necessary air, oil, water and electrical connections, all enclosed in an attractive chrome-trim gray cabinet. Of direct fired type, developed especially for residential air conditioning, it circulates clean, warm, moistened air through ducts. Automatic Night and Day temperature control available with both units Total outputs are. 100,000, 133,000 and 160,000 Btu per hour respectively.



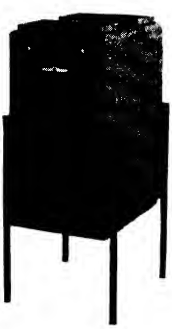
G-E Warm Air Conditioners, Gas Fired—Conventional and vertical models Consists of combustion heat transfer unit, gas burner, centrifugal fan, humidifier, controls, and necessary gas, water, and electrical connections enclosed in attractive cabinet. Direct-fired air conditioner, developed especially for residential and small commercial air conditioning, which circulates clean, warm, moisened air through the conditioned space Numerous sizes available to meet heating requirements from 35,250—216,000 Btu per hour, output, with air flows ranging from 400—2700 cfm. Approved by *A G.A*.

G-E Gas Furnaces—Designed for steam, vapor or hot water radiator systems and for indirect heating with air conditioners. Type LM ratings from 320 to 1440 sq ft steam; Type LK ratings from 660 to 1760 sq ft steam; Type LC ratings from 1980 to 9680 sq ft steam. Automatic pressure, low water and temperature limit control. Gas regulation is gas operated to assure positive action. Cast iron sectional boilers meet *A.S.M.E.* boiler code. Approved by *A.G.A.*

G-E Unit Air Conditioner—Type FD-30—Consists of an enclosed sound-proofed 3 hp condensing unit, water cooled condenser, four row cooling coil, fan, filters, adjustable air discharge, and controls all enclosed in a golden bronze cabinet. The unit is designed to be installed quickly and easily within the air conditioned space of a general office or small commercial establishment. Outstanding features include simplicity in design, easy accessibility, quiet operation, and simple installation permitting easy relocation.

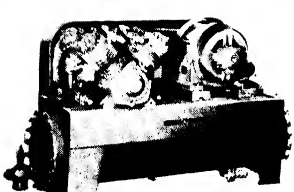


G-E Unit Room Air Conditioner—Type AF-1—This unit is designed for comfort cooling applications where cooling, dehumidifying, circulating, ventilating and cleaning of air are desirable. The air conditioner includes a condensing unit, cooling coil, fans, filters, air or evaporative cooled condenser and controls, all enclosed in an attractive walnut cabinet. The unit features high cooling capacity, low operating cost, ease of installation and pleasing appearance.



G-E Air Conditioner for Winter—Type HW-1 designed for winter air conditioning of radiator heated homes. Includes filters, humidifier, tempering coil and radial flow aphonic fan.

G-E Air Circulator—Types HV1B, HV-1D (illustrated), HV2A, and HV2B for attic ventilation, air circulation and exhaust applications. Type HV1D also available with pedestal mounting for air circulation.



G-E Condensing Units—Available in sizes from 1 hp through 50 hp. Several air cooled models in small sizes; water cooled models with shell-and-coil and truly cleanable shell-and-tube condensers. Efficient design provides high cooling capacities. Designed especially for air-conditioning application as part of complete G-E air conditioning systems.

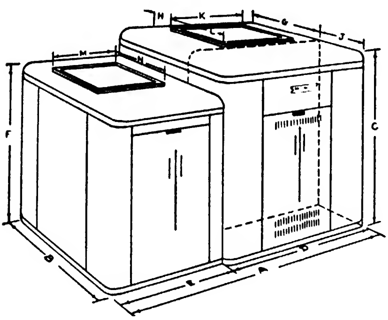
G-E Central Plant Air Conditioners—A complete line of factory designed air conditioners for summer, winter or year 'round applications. Types HD-50, 100, 200 and 300 units include aphonic radial flow fan, filters, humidifiers, cooling coils and heating coils in combinations which meet a wide range of air conditioning functions and required capacities. The larger sizes, Types HD-400, 500,600, and 700 include filters, humidifiers, cooling coils, heating coils, to meet requirements of large, single and multi-zone systems.

(*See also Pages 1044-1045*)

The Henry Furnace & Foundry Company

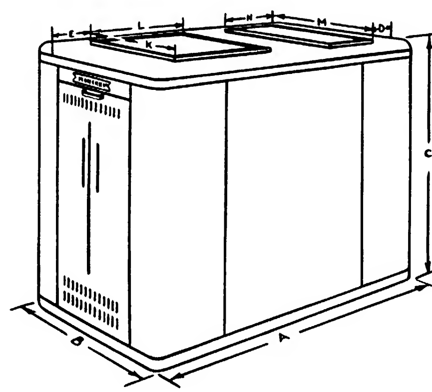
Manufacturers of MONCRIEF Furnaces & Air Conditioning Systems

3471 East 49th Street **Cleveland, Ohio** Phone Michigan 9322



Aristocrat
Oil Fired Air Conditioner

Moncrief Aristocrat and "Special" Oil Fired Winter Air Conditioners, made to operate with any standard Gun Type Burner, and Gas Fired Winter Air Conditioners are available in numerous sizes and capacities. Write for particulars.



C B F
Gas Fired Air Conditioner
Approved by American Gas Association

"SPECIAL" OIL FIRED AIR CONDITIONERS

Unit No.	75	100
Input Gallons per hour	.76	1 02
Size of casing		
Width	28″	28″
Depth	71″	71″
Height	60⅜″	60⅜″
Blower size	110	112
Motor Hp	1/6	¼
Heating surface—sq in	6120	6120
Cfm	915	1225
No filters 13 x 25	2	2
Btu Del at Register	75,000	100,000

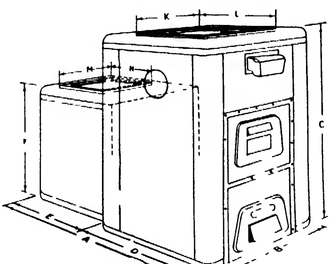
SIZES, CAPACITIES—Aristocrat Oil-Fired Units.

A. C Unit Size No	Casing Dimensions									Plenums: Warm Air		Plenums: Cold Air		Btu Per hour at Reg	Cfm	Blower Size	Blower Motor Hp	Sq In Heating Surface	No of Filters	Size of Filters	Shipping Weight (Approx.)	Input Gal Per Hour
	A	B	C	D	E	F	G	H	J	K	L	M	N									
125	76⅝	51⅜	60⅜	48½	28	51¾	38⅜	5 3/16	18	20	28	20	28	125,000	1525	112	¼	8925	4	16x25	1240	1 28
150	76⅝	51⅜	60⅜	48½	28	51¾	38⅜	5 3/16	18	24	28	24	28	150,000	1830	114	¼	8925	4	16x25	1265	1 53
175	80½	56¼	60⅜	52½	28	51¾	43¼	5⅝	18	24	32	24	32	175,000	2200	114	⅓	11088	4	16x25	1365	1 8
200	80½	56¼	60⅜	52½	28	51¾	43¼	3⅝	18	24	36	24	36	200,000	2440	114	½	11088	4	16x25	1455	2 04
275	97	64½	64	59	38	51¾	51½	5¾	18	30	40	30	40	275,000	3350	214	¾	13248	6	16x25	2010	2 8

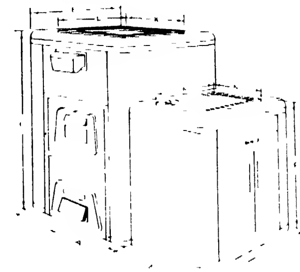
SIZES, CAPACITIES—Gas-Fired C.B.F Units.

Unit No	Casing Dimensions					Plenum Size Inches: Warm Air		Plenum Size Inches: Cold Air		Btu Input	Btu at Register	Blower Size	Motor Hp	No of Filters	Filter Size	Cfm Required	Sq In Heating Surface	Gas Line	No of Burners	Flue Size
	A	B	C	D	E	K	L	M	N											
G-8	75	20	54	1½	13	14	32	16	14	80,000	61,200	110	1/6	2	16x25	748	5410	1″	2	6″
G-12	75	26	54	1½	13	19	32	22	15	120,000	91,800	112	¼	3	16x25	1124	8115	1″	3	6″
G-16	75	34	54	1½	12	26	34	30	12	160,000	122 400	112	¼	4	16x25	1500	10820	1″	4	7″
G-20	75	41	54	1½	12	32	34	37	12	200,000	153,000	114	⅓	4	16x25	1870	13525	1″	5	8″
G-24	75	49	54	1½	12	40	34	44	12	240,000	183,600	114	½	5	16x25	2244	16230	1″	6	8″
G-32	75	68	54	1½	12	52	34	60	12	320,000	244,800	214	¾	6	16x25	2994	21640	1¼″	8	9″
G-40	75	78¼	54	1½	12	66	34	72	12	400,000	306,000	214	¾	8	16x25	3740	27050	1¼″	10	10″
G-48	75	94½	54	1½	12	80	34	88	12	480,000	367,200	214	1	9	16x25	4480	32460	1¼″	12	11″

Moncrief Aristocrat Coal-Fired Winter Air Conditioners are made with either cast or steel heating elements, in a wide range of sizes. Equipment includes metal floor, blower with motor, filters, controls, damper motor, thermostat and Thermo-Drip Automatic Humidifier, duplex roller-bearing grates—except with triangular grates, at no extra cost (on cast only). If to be used with stoker, grates and controls—except blower control—may be omitted, stokers being equipped with control. Fine-grain crackle finish—green, trimmed with chromium plate



For Coal with Cast Heating Unit



For Coal with Steel Heating Unit

With Cast Heating Element.

Unit Size No	Casing Dimensions—inches						Plenum Size Warm air		Plenum Size Cold air		Blower Size	Motor Hp	Sq In Heating Surface	Grate Area Sq In	Cfm Required	No of Filters	Forced Air Register Btu Coal Hand-Fired	Forced Air Register Btu Stoker Fired
	A	B	C	D	E	F	K	L	M	N								
4020	66¼	40¼	62¾	38¼	28	50¾	30	28	30	18	110	¼	5021	214	967	2	73,800	85,200
4422	68¼	42¼	62¾	38¼	30	50¾	32	28	32	20	112	¼	5609	269	1157	3	88,300	101,800
4824	70¼	44¼	62¾	40¼	30	50¾	34	30	34	20	112	¼	6010	326	1330	3	101,600	117,300
5226	78¼	46¼	66	44¼	34	50¾	36	34	36	22	112	⅓	6753	397	1580	4	120,400	139,000
5628	82¼	48¼	66	48¼	34	50¾	38	38	38	24	114	⅓	7579	472	1840	5	140,300	162,000

With Steel Heating Element.

Unit Size No	Casing Dimensions—inches							Plenum Size Warm air		Plenum Size Cold air		Blower Size	Motor Hp	Sq In Heating Surface	Grate Area Sq In	Cfm Required	No of Filters	Forced Air Register Btu Coal Hand-Fired	Forced Air Register Btu Stoker Fired
	A	B	C	D	E	F	G	K	L	M	N								
2054	68¼	40¼	62¾	40¼	28	50¾	35¼	30	30	30	16	110	¼	6226	214	1070	2	81,600	94,200
2254	70¼	44¼	62¾	40¼	30	50¾	39¾	30	34	34	18	112	¼	6499	269	1232	3	94,000	108,400
2456	74¼	46¼	66	40¼	34	50¾	41¼	30	36	36	20	112	¼	6960	330	1424	4	108,700	125,500
2758	74¼	51¼	67¼	40¼	34	50¾	46¼	30	42	42	20	114	⅓	8086	434	1815	5	138,600	160,000
3060	86¼	54¼	69	48¼	38	50¾	49¼	36	42	42	24	212	⅓	10041	552	2240	5	171,000	197,300
3060J	90¼	54¼	69	52¼	38	50¾	49¼	40	42	42	24	212	½	11480	552	2362	5	180,300	208,000
3462	90¼	60¼	71	48¼	42	50¾	55¼	36	48	48	24	214	¾	11245	730	2800	6	213,800	246,700
3462J	96½	60¼	71	54¼	42	50¾	55¼	42	48	48	24	214	¾	12386	730	2895	6	221,200	255,200

Heating capacities are based on a 7 lb combustion rate using 12,000 Btu coal and efficiencies of —65 per cent at bonnet, 85 per cent register for fan coal hand fired—75 per cent at bonnet, 85 per cent at register for oil or stoker fired

If 6 lb combustion rate is desired for prolonged firing period deduct approximately 15 per cent from forced air Btu ratings Maximum velocity through filters, 200 feet per minute.

The Anthracite Winter Air Conditioner consists of a MONCRIEF Unit and a Miles Combustion Modulator.

The outside temperature is correlated with the warm air temperature to affect a Graduated Room Temperature, 68 deg in mild weather and 78 deg in zero weather.

Constant Modulated Heat Supply eliminates COLD 70 and complicated control equipment. Write for particulars

NOTE: Made with either the Moncrief Cast or Steel Aristocrat Air Conditioner.

Kelvinator

Division of Nash-Kelvinator Corporation

SUMMER AND WINTER AIR CONDITIONING

Factories in Detroit, Michigan and London, Ontario

Factory Division Offices in

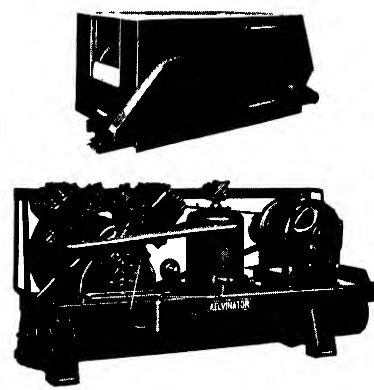
BOSTON	ATLANTA	CHICAGO	DALLAS
NEW YORK	CINCINNATI	ST. LOUIS	SAN FRANCISCO

Distributors in more than 200 Cities

FACTORY BUILT AIR CONDITIONING EQUIPMENT FOR EVERY REQUIREMENT

No matter what the requirements are, from a self-contained unit for air conditioning a single room to central system equipment for complete building installations, Kelvinator perfected equipment will exactly fit the need.

Evaporative Condensing Units. These factory-built units are available in all standard sizes up to 50 hp. Installation costs are saved because of unit construction of evaporative condenser. Use of spray discs instead of spray nozzles eliminates need for pumps, lowers power costs, and avoids a common cause of service interruptions Condenser can be installed in any unused over-head space above condensing unit or in machinery room Factory-built, assembled and tested before shipment, it is ready for duty as soon as it is mounted in position and connected.


Kelvinator Evaporative Condensing Unit and 20 hp Compressor

Construction features include. an all-galvannealed steel cabinet built on a heavy angle-iron frame, with removable panels to make the interior easily accessible for cleaning: the quiet-operating, centrifugal-type fan has ample capacity for circulating and cooling; condenser coils are fin type with wide spacing that facilitates cleaning; the brass spray discs which distribute a finely divided shower over the condenser coil, insure a fixed quantity of water at all times without requiring periodic adjustment and cleaning as when a pump and spray nozzles are used.

Kelvinator Condensing Units. Water-cooled types are available in all standard sizes up to 50 hp; air-cooled types from ¼ to 5 hp, inclusive

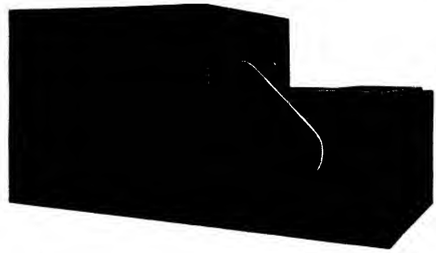
The use of air-cooled condensing units in sizes as large as 5 hp is made possible by the Kelvinator refrigerant-cooled head. Large condenser capacities in the air-cooled models make efficient operation possible with high temperature condenser air.

All water-cooled units have water-cooled compressor cylinder heads. Formation of carbonized oil is prevented by maintaining the temperature of the cylinder head and valve plate below the carbonization temperature of the oil. Water-cooled models of 10 hp and larger are equipped with shell and copper tube cleanable condensers. The condenser water passages are so designed that water pressure drop is kept at a minimum, assuring efficient operation in communities having low water pressures.

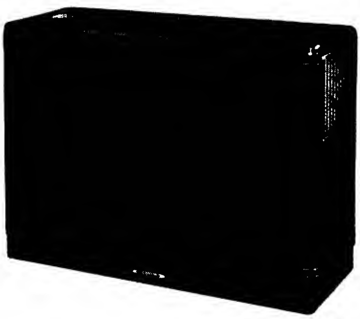
Kelvinator precision manufacture insures high operating efficiency.

Central System Air Conditioning Units. Completely-controlled air conditioning system; with automatic cooling, dehumidifying, filtering, and air distribution. Kelvinator heating and humidifying equipment may be installed at any time. Designed, engineered, and completely factory-built and factory-tested, for high efficiency and low operating cost.

Capacities: 10 to 40 tons I.M.E. per day.



Kelvinator Central System Air Conditioning Unit.

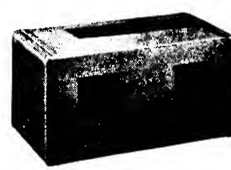


Kelvinator Air-Cooled Room Cooler; completely self-contained.

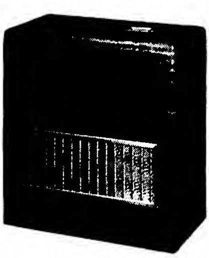
Self-Contained Air Conditioners for Single Room Installations. Air-cooled models in ½ hp and 1 hp sizes. Water-cooled models in ½ hp, 1 hp and 1½ hp sizes. Provide cooling, dehumidifying, cleaning, and circulation. Outside air intake standard on air-cooled models, winter-heating coils optional. All models can be equipped with thermostat for automatic operation.

Kelvinator Room Coolers can be moved to a new location as easily as other office or household equipment.

Suspended Type Air Conditioning Units. For 0.7 ton to 10 ton installation in locations where floor or room space is limited. Fully automatic control. Complete cooling, dehumidifying, filtering, and air distribution. Heating coil and humidifier can be included in sizes of 5 tons and larger. This equipment is designed for installation right in the room to be air conditioned. Eliminates costly duct work; ideal for business firms holding short-term leases.



Kelvinator Type C Suspended Air Conditioner; condensing unit remote.



Kelvinator Floor-Type Air Conditioner; condenser remote.

Floor-Type Cabinet Air Conditioning Units. For use with Freon or circulating cold water as the refrigerant. Heating and humidification for winter air conditioning, and outside-air intake, with filter, can be provided. Capacities: 0.55, 0.80, 1.10, and 1.60 tons.

These units require about the same space and cost little more than the radiator they may replace. They are particularly suitable for air conditioning bedrooms, hotel guest rooms, hospitals and office suites and other existing construction where duct installation expense would be prohibitive.

Kelvinator Store Air Conditioner

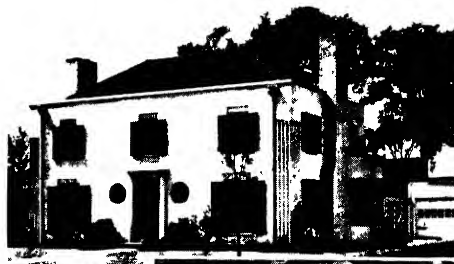
Kelvinator Store Air Conditioner. Designed to provide simplified, low-cost air conditioning in a self-contained unit for the average business establishment. Can be easily adapted to a variety of different situations. Where air conditioning is desired for several adjoining rooms, the unit can be installed at a convenient, centrally located point and ducts run to the spaces to be air conditioned. Simplified provision for air return can be made by use of air grilles or louvers in inner office doors.

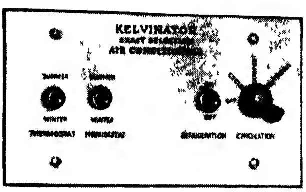
Completely self-contained, factory-built, it can be installed easily and quickly; requires no expensive installation and can be readily moved to new quarters.

Residential Air Conditioning and Automatic Heating Equipment

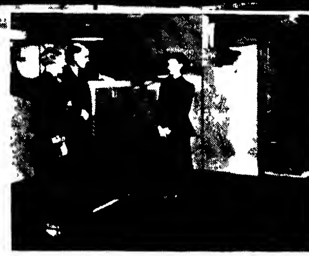
25 years of experience in manufacturing controlled temperature appliances is back of Kelvinator's year-'round air conditioning and automatic heating equipment Developed in the Kelvinator Research Residence, it has been proved in hundreds of Kelvin Homes throughout the country. These homes provide new scientific advances in year-'round comfort and household conveniences for families of moderate income

Year-'Round Air Conditioning System. Kelvinator's residential air conditioning system is of the indirect type. The air is heated by a steam coil in the conditioner Steam is supplied by an automatic boiler This boiler also furnishes heat to radiators for heating those parts of the house where conditioned air is not required



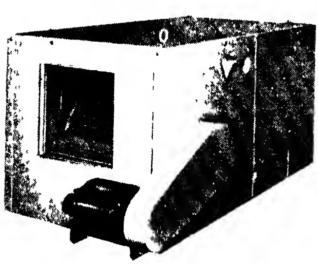


Kelvinator Exact Selection Control Panel assures "Finger-Tip" Selection of Conditions to be maintained



The illustration shows the automatic oil-fired boiler (left) the air conditioner (above at center), and the condensing unit for summer cooling (inside the small room at right)

The Conditioner. The conditioner contains a heating coil, cooling coil, filter, humidifier and centrifugal fan. Steam for the heating coil is provided by an automatic boiler, and refrigerant for the cooling coil is supplied by a Freon mechanical cooling unit Heating and cooling coils are provided in a variety of sizes to suit all climatic conditions

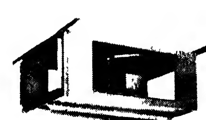


Kelvinator Type RC Conditioner

The attractively finished conditioner is resiliently mounted and is insulated to insure quiet operation The operation of the conditioner is automatically controlled by the thermostat in summer, and by the thermostat and humidistat in winter

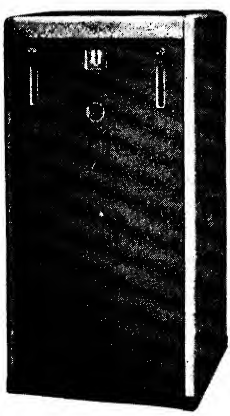
Four sizes with air deliveries up to 2000 cfm provide for conditioning any size home Multiple units can be used for zoned systems in large residences

The Comfort Damper. The comfort damper, controlled by the Selector, permits outside air to be brought to the conditioner, filtered and circulated throughout the house Mechanical cooling load is reduced and operating costs are cut in summer through the introduction of cool, night air and discharge of warm air



The comfort damper takes full advantage of the coolness of night air

Controls. Kelvinator residential air conditioning features the finger tip Exact Selection control. However, desired temperature and humidity may be automatically maintained with the thermostat and humidistat. Beside these a conveniently located Selector panel contains switches by which conditions to be maintained may be selected. Simply placing the three toggle switches in the up position establishes the system in summer operation; placing them in the down position selects winter operation. The rotary switch provides for (1) a completely off position for the air circulation system; (2) a position in which the air circulation cycles under control of the thermostat; (3) comfort position which controls the comfort damper, stopping recirculation and allowing the introduction of 100 per cent outside air circulation. With this Selector it is not necessary to call a service man to change from summer to winter or winter to summer operation.



Kel-O-Flame Oil-Fired Boiler.

Oil-Fired Unit Boilers. The Kel-O-Flame oil-fired unit boilers are provided in a range of capacities from 103,000 Btu/hour to 252,000 Btu/hour to efficiently meet the automatic heating requirements of small as well as large residences. Employing a completely coordinated burner design with a balanced heat-absorbing element, the Kel-O-Flame unit boiler has set a new standard of performance and efficiency.

The specially designed burner produces a quiet, non-luminous flame in direct contact with the primary surface of the boiler. Finned secondary heating surfaces extract the remaining usable heat from the flue gases. The low water line permits its installation in low-ceiling basements. Jacket is thoroughly insulated with asbestos cell. All working parts are easily accessible for adjustment or service.

Each unit is complete with fully automatic controls, including a low-water cut-off for steam boilers, safety stack switch, safety limit control, summer hot water control, and for hot water boilers a circulator relay and flow control valve. Provision is made for automatically supplying domestic hot water summer and winter.

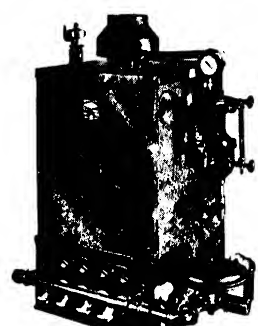
As the boilers are entirely factory-built, hazards of field assembly and installation are eliminated.

Gas-Fired Unit Boilers. Six sizes, for hot water or steam heating, are available, capacities up to 294,000 Btu/hour.

Quick steaming is achieved by the use of special venturi mixing throats interconnecting with diamond-shaped horizontal water tubes. Heat transfer surfaces are unusually large in proportion to water content of the boiler, frictional resistance within the boiler is materially reduced.

Dependable operation of pilots and burners is assured by a special down-draft diverter, which constantly regulates the draft through the boiler. An automatic combination valve supplies gas under constant pressure whenever the thermostat calls for heat.

Jacket is fully insulated. Complete automatic control equipment is provided, including a low-water cut-off, automatic electric or gas-failure shut off, and an electric limit control.



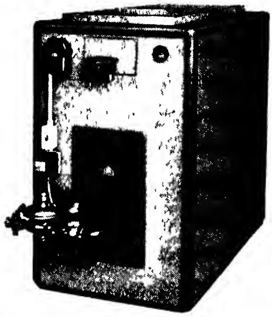
Kelvinator Gas-Fired Unit Boiler.

Kelvinator Oil-Fired Conditioner.

Oil-Fired Conditioner. 4 lines with capacities from 100,000 Btu/hour to 250,000 Btu/hour. This conditioner filters, heats, humidifies and circulates the air to accomplish complete winter air conditioning, and to provide automatically a more comfortable atmosphere in the home. Kelvinator Conditioners are appropriate in their size and appearance for any home requirements. Completely factory-built for high efficiency and low operating cost.

Gas-Fired Conditioner. Available in 3 lines, 4 models in each. Capacities from 90,000 Btu/hour to 200,000 Btu/hour. The Kelvinator gas-fired conditioner is a completely automatic heating system making use of gas. This conditioner maintains uniformly comfortable temperatures and also accomplishes a complete winter air conditioning effect by cleaning, humidifying and circulating the conditioned air throughout the house.

The gas-burning unit is of the atmospheric, individual port, up-shot type with automatic gas valve pressure regulator. Safety pilot prevents opening of main control valve in the event that flame should be extinguished. This insures absolutely safe operation.



Kelvinator Gas-Fired Conditioner

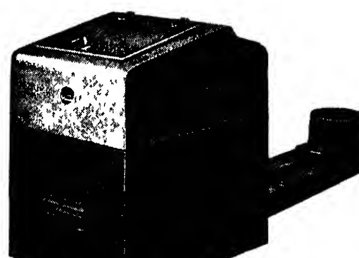
Kelvinator Oil Burners. Kelvinator conversion oil burners are especially developed to insure quiet, dependable, and efficient performance in conventional heating plants

Kelvinator Conversion Oil Burner

They employ the pressure-atomizing principle. By means of a scientifically designed turbulator and air deflector, thorough mixture of the air with the atomized oil is achieved. This results in complete burning of the fuel within the combustion chamber, and makes the full amount of heat available from low-cost, heavy grades of domestic fuel oil.

Seven models of burners may be fired at oil rates from 1.25 to 25 gal an hour with capacities up to 2,270,000 Btu/hour.

Kelvinator Coal Stokers. Kelvinator Automatic Coal Stokers introduce important improvements in the automatic firing of bituminous fuels. They utilize the proved underfeed principle. Gases, ordinarily wasted, are scientifically released and thoroughly burned over a slow-burning coke fire. Combustion is complete and smokeless.



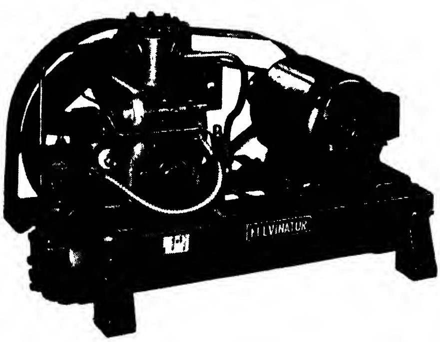
Kelvinator Automatic Coal Stoker

Burner is made of nickel-steel tuyere segments, with special air vents which distribute air uniformly to the fuel bed. A controlled air-damper regulates air flow automatically, and holds the fire down when heat is not required

Coal feed is by a cast steel tapered worm, full floating in a seamless steel tube. Transmission gears are drop-forged steel, case-hardened and ground, operating in a continuous bath of oil. An easily accessible adjustment provides the choice of 5 different speeds Feed is intermittent, providing the very necessary poking action in the fire bed A special coal agitator is available for use with fuels that have a tendency to cake.

Construction provides easy accessibility of all working parts for adjustment or service. Coal hopper of smaller, household size models, holds more than a day's supply of coal. Hopper cover is gas tight, preventing any possibility of blow back of smoke or dust

Five sizes of Kelvinator stokers are available with maximum feeds of 35 to 150 lb per hour and capacities of 210,000 to 900,000 Btu/hour.



Kelvinator water-cooled condensing unit for summer cooling.

Kelvinator Condensing Unit

See paragraph on Kelvinator Condensing Units under Commercial Air Conditioning Section.

THE MEYER FURNACE COMPANY

PEORIA, ILLINOIS

Manufacturers of Domestic Heating and Air Conditioning Units for Coal, Gas and Oil Burning

Branches and Distributors
Kansas City, Mo.
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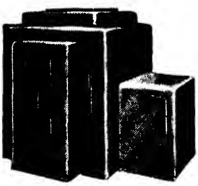
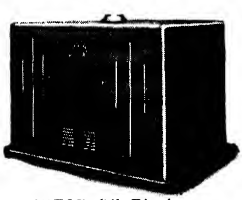


The WEIR Air Conditioner for solid fuels (hand or stoker fired) embodying the famous WEIR Steel Furnace is a complete unit for winter air conditioning, including air cleansing, humidifying and forced circulation as well as heating. Encased in furniture steel with baked enamel finish. Automatic controls for dampers, blower, humidifier.

WEIR Conditioned Air Unit · *WEIR Gravity Heater*

No.	Grate Surface (Sq Ft)	Ratio Htg. to Grate Surface	Smoke Outlet Diam. (In.)	Gravity Circulation				Fan Circulation		
				Casing Dimen.		Rated Output		Casing Dimen. (In.)	Air Delivery (CFM)	Rated Output at register (Btu. Hour)
				Round (In.)	Rect'lar (In.)	At Reg. (Btu. Hour)	Pipe Area (Sq In.)			
621	1.26	41.2	9	48	.....	54,400	400	......	....	....
624	1.78	33.9	10	52	47x50	73,600	541	47x90	1200	92,000
628	2.32	29.2	10	54	50x52	94,100	692	50x99	1600	118,000
630	3.08	26.4	10	58	54x56	119,000	875	54x103	2000	148,000
633	3.82	22.7	10	65	56x64	138,000	1015	56x110	2300	172,000
636	4.74	19.4	10	67	56x66	160,000	1180	56x118	2700	200,000
540	6.25	19.3	12	..	.....	....	....	60x106	4000	264,000
544	7.60	18.5	12	..	.....	....	....	64x114	5000	316,000



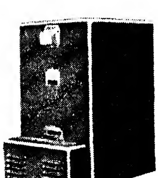
WEIR Oil Fired Air Conditioner

The WEIR Oil-Fired Air Conditioner does a complete job of winter air conditioning. Designed for oil fuel and forced circulation. Completely self-contained in *low* compact casing enclosing burner and blower as well as heater and all controls, yet with everything easily accessible.

The MEYER Gas-Fired Air Conditioner automatically provides completely controlled winter air conditioning. Efficient in performance, compact in design, and modern in appearance. Heavy gauge welded steel heating section; die-formed furniture steel casing. *A.G.A.* approved.

MEYER Gas Fired Air Conditioner

No.	Input at Burner (Btu/Hour)	Output at Bonnet (Btu/Hour)	Vent Diam. (In.)	Dimensions			Air Delivery ¼ In. S.P. (CFM)	Motor Size (HP)
				W. (In.)	L. (In.)	H. (In.)		
WEIR Oil-Fired Air Conditioner								
125-A	185,000	125,000	8	48	68	48	1700	1/4
175-A	260,000	175,000	8	56	68	48	2300	1/3
225-A	335,000	225,000	8	63	68	48	3000	1/2
MEYER Gas-Fired Air Conditioner								
E-10	110,000	88,000	5	32	69	45	1200	1/4
E-15	165,000	132,000	6	41	69	45	1800	1/3
E-20	220,000	176,000	7	53	69	45	2400	1/3
E-30	330,000	264,000	9	69	69	45	3600	1/2
E-45	495,000	396,000	10	97	69	45	5400	3/4
MEYER Gravity Gas Furnace								
C-100	100,000	75,000	5	38	38	60	Complete descriptive literature, including data on Summer cooling, upon request	
C-120	120,000	90,000	6	42	42	67		
C-150	150,000	112,500	6	42	42	67		



MEYER Gravity Gas Furnace

The MEYER Gas Furnace—Efficient—Economical—All Steel, welded heating section, die-formed insulated casing. *A.G.A.* approval.

Branches
St Louis
Memphis
Omaha
Minneapolis
Salt Lake City
Dallas

L. J. Mueller Furnace Co.

Established 1857

2009 W. Oklahoma Ave., **Milwaukee, Wis.**

Branches
Los Angeles
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Baltimore
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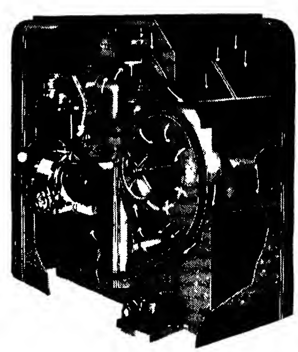
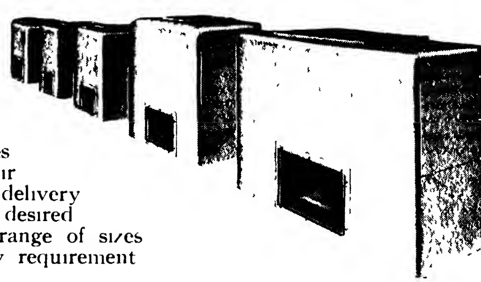
Chicago

SERIES "O" OIL-FIRED AIR CONDITIONING UNIT



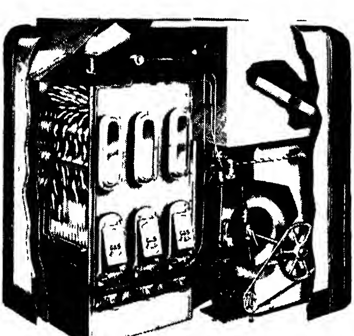
The basic, patented design of this Mueller unit secures a new standard of efficiency and fuel economy The air passes over heating surface not once, but three times Velocity and impingement create a rate of heat transfer impossible in previous designs The silent fan supplies generous volumes of warm, humidified, filtered air to secure uniform temperatures The Mueller Oil Burner is of pressure atomizing type, specially designed for use with this unit Details of construction and method of operation of the complete unit are shown at the right Available in three sizes to handle most residential requirements

CLIMATOR FAN-FILTER UNITS



For positive distribution of air in old or new construction, wherever air is to be moved, there is a Mueller Climator Fan-Filter unit available

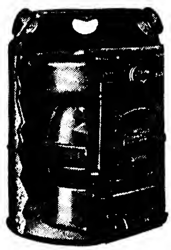
Fans are designed in accordance with latest aerodynamic principles Filter area is adequate to handle the air requirements Fans have ample air delivery capacity for the addition of cooling, if desired Climator units are available in a range of sizes capable of handling practically any requirement



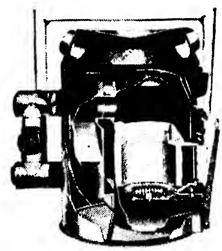
CLIMATROL AIR CONDITIONING FURNACE

Enclosed in the compact, ultra-modern cabinet are the furnace, fan, humidifier and filters, together with controls for automatic operation. Heat is supplied by Mueller gas-fired "Heat-Speeder" sections The powerful Mueller fan provides positive circulation of conditioned air to all rooms Cut-away view shows fire travel through one of the sections, also the complete assembly of fan, motor, humidifier, filters and controls Available in five sizes

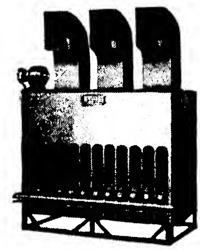
Mueller Heaters For All Fuels
A Complete Line for All Purposes



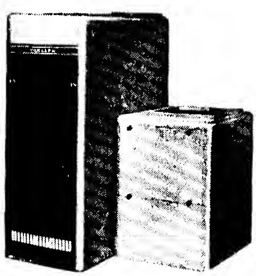
Return Flue all-cast Furnace. 18 in. to 30 in. firepots, single and double firedoor styles. Available in round, galvanized or square, lacquered casings.



Mueller Steel Furnace. Riveted and welded. Extra heavy construction. Burns any fuel. Seven sizes, 20 in. to 34 in. drums. Square or round casing.



Gas Era Unit Heater. Available in sizes from six to twenty sections, with *AGA* output ratings from 216,000 to 720,000 Btu per hour.



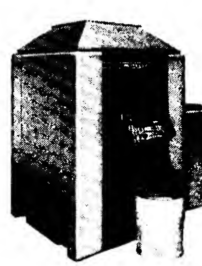
Gas Era Cast Iron air conditioning furnace. *AGA* input rating, 65,000 Btu per hour, per section. Wide range of sizes and air delivery capacities.

Gas-fired air conditioning furnace. *AGA* input rating, 45,000 Btu per hour, per section. Wide range of air deliveries.

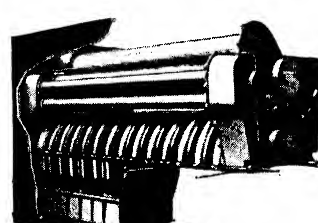
Muelleraire unit. Gas-fired with fan, filters, and humidifier. 5 sizes—*AGA* input ratings, 72,000 to 180,000 Btu per hour.

Series "A" Gas Boiler. *AGA* ratings, 180 to 1,260 sq ft steam; 290 to 2,015 hot water.

Series 60 oil-fired air conditioning furnace. Burner optional. Two sizes, 110,000 and 165,000 Btu per hour.



Series "SA" stoker-fired furnace, with fan-filter unit. Any stoker may be used. Capacities, 110,000 and 175,000 Btu.



Horizontal Tubular Heaters, for schools, churches and other large buildings. Three sizes, with capacity range from 1,188,000 to 1,390,000 Btu per hour.

Complete catalogs on each of above units available upon request.

Williams Oil-O-Matic Heating Corporation

Manufacturers of Automatic and Manually Controlled Fuel Oil Burners

Bloomington, Illinois

Service to Architects and Builders

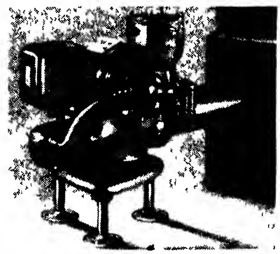
CHICAGO, ILL., 641 N Michigan Avenue — NEW YORK, N Y, 2014 Graybar Building

For Williams Ice-O-Matic Refrigeration Equipment, see File Index

A Complete Line

Williams Oil-O-Matic offers six Oil-O-Matic burner models—a genuine Williams Oil-O-Matic for every size and type of house, for every apartment, public building or commercial structure Also, the new Williams HP-3 (high pressure) model

Complete boiler burner and furnace-burner units, a new type of oil-burning automatic water heater, an oil-burning range burner for heavy duty ranges are available



WILLIAMS OIL-O-MATIC HEATING

Oil-O-Matic Water Heaters

Made in 3 sizes covering ordinary home use, large home or commercial use, and heavy duty requirement. The Oil-O-Matic oil burner, combustion chamber, water reservoir, and all automatic controls are combined in one compact unit. Horizontal tank permits the use of a unique triple flame travel which transfers all possible heat to the water. See specifications lower left.

Heating Capacities of the Various Williams Oil-O-Matic Burners

Model	Domestic Shipping Weight, Lb.	Length, in.	Width, in.	Height, in.	H.P.	R.P.M.	Gals Fuel Oil per Operating Hour Minimum	Maximum
K-150	125	29	15¼	19¾	1/10	1800	½	1½
K-3	145	30	15⅝	20¾	1/10	1800	1	3
K-4/5	170	33½	20¼	22¼	1/5	1800	2½	4½
K-7	175	32½	22¼	21¾	1/5	1800	4	7
J-1800	255	45	20	24	1/2	1800	8	15
JJ	295	50	32	24	1	1800	12	25
HP-3 (high pressure)	115	30	17	18	1/10	1800	1.35	3

Standard draft pipe 18 in 12-in length draft pipe optional

Standard electric current is 110-volt, 60-cycle In case odd frequency motors are used, the maximum capacity of the burner will be reduced in direct proportion to the rpm of the motor The minimum capacity remains the same

Water Heaters

WHA*	600	57	22½	28	1/10	1800		½
WHB†	885	75	23	28	1/10	1800	½	1
WHC‡	1385	90	29	34	1/10	1800	1	1¾

*Output: 90 F rise, 60 gal per hour. †Output 90 F rise, 120 gal per hour ‡Output 90 F rise, 210 gal per hour.

How to Decide Size of Burner

For low pressure domestic boilers, 1 gal of fuel oil per hour (140,000 Btu) is required for approximately:

300 sq ft of steam radiation or its equivalent.

480 sq ft of hot water radiation or its equivalent.

70,000 Btu when using hot air furnace ratings.

24 sq ft steam boiler heating surface (or 2.2 hp).

For exact detail data, see Oil-O-Matic Installation and Service Manual.

Complete Boiler-Burner and Furnace-Burner Units

Oil-O-Matic burners are also used as an integral part of the following boiler-burner and furnace-burner units:

Abram-Cox Oil-O-Matic, Evans Oil-O-Matic, International Oil-O-Matic, Kruse Oil-O-Matic, Orr & Sembower Oil-O-Matic, Waterman-Waterbury Oil-O-Matic

Boiler-burner units automatically provide domestic hot water supply the year 'round. Indirect heating coils may be included for either instantaneous or storage tank systems Furnace-burner units provide winter air conditioning

Horizontal Boiler-Burner Unit

A complete Oil-O-Matic product, consisting of a boiler and Oil-O-Matic oil burner It is furnished for both steam and hot water systems The welded steel tank used meets *A.S M.E* code specifications for operating up to 15 lb pressure Boiler capacities are: 610 sq ft equivalent direct steam radiation; 980 sq ft equivalent direct hot water radiation.

Underwriters' Listing

Listed as standard by Underwriters' Laboratories, Inc. Also approved by all important codes and governing bodies.

Fuel Oil Range Burners

Williams Oil-O-Matic fuel oil burner for use with heavy duty ranges in restaurants, hotels, hospitals, steamships, dining cars, resorts and clubs Furnished as an integral part of the Oil-O-Matic South Bend Heavy Duty Range.

Engineering Service

Available to architects. See *A. I. A.* File No. 30 G-1.

Oakite Products, Inc.

22 Thames Street, **New York, N. Y.**

Branch Offices and Representatives in All Principal Cities of the U. S.

OAKITE AIREFINER

PATENT APPLIED FOR

Established 1909

A new material that prevents corrosion, controls bacteria growth, prevents slime and algae accumulations. Materials also available for rust and scale removal, cleaning and deodorizing.

Bacteria, Slime and Algae Control

Does your air conditioning equipment use a recirculating water supply? Increase the efficiency of it by preventing the growth of bacteria, slime and algae in the circulating water. This is accomplished easily, inexpensively with a new development of the Oakite Research Laboratories, known as Oakite Airefiner. It is a dry, non-volatile white powder, completely soluble in water to provide safe, odorless, non-toxic solutions. Used in extremely low concentrations, one pound to 300 to 500 gal of water, this powerful bactericidal agent controls the growth of bacteria, slime and algae. Equipment is kept free of these accumulations, and from the unpleasant odors which they create.

Preventing Corrosion

Oakite Airefiner has been specially compounded to also incorporate sufficient alkalinity to counteract the development of acidity in the water due to extraction of sulphur dioxide and carbon dioxide gases from the air. If not prevented, this acid condition causes rapid corrosion of eliminators, air wash chambers and other metallic surfaces with which the untreated water comes in contact.

Cleaning and Deodorizing

Users of Oakite Airefiner enjoy a number of other definite advantages Increased wetting power of water containing Oakite Airefiner has resulted in actual removal of more dirt from the washed air. Eliminators, spray heads and water feed lines are kept free of scale and similar corrosion as well as of bacterial accumulations, slime and algae. The washed air is freed of practically all the matter which is responsible for objectionable odors in re-used air.

Cleaning Filter Screens

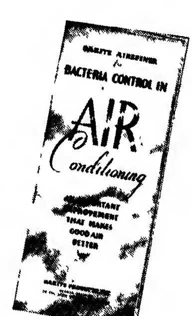
Where filters are of the re-usable and washable type, Oakite cleaning methods provide a quick, safe, complete removal of the accumulated dust and dirt. No injury is done to the glass, metallic or similar material which comprises the filtering surface.

Rust and Scale Removal

Removal of rust and scale from feed water pipes, water cooling chambers or on other metallic surfaces can be accomplished safely and economically with Oakite Compound No 32 This is an acidic material especially prepared to provide absolutely uniform scale and rust dissolving action without having any harmful effect on the sound underlying metal. Specific recommendations on work of this type will be sent upon request.

Nation-Wide Service

Oakite materials are supplied through a nation-wide group of service representatives who are thoroughly experienced in their application. Users are thus assured of employing the materials in the way to get the full benefits they provide



Write for FREE, interesting fact-filled booklet on the use of Oakite Airefiner and the other Oakite materials designed specifically for helping you get the maximum life and performance from your air conditioning equipment. No obligation, of course.

The Air-Maze Corporation

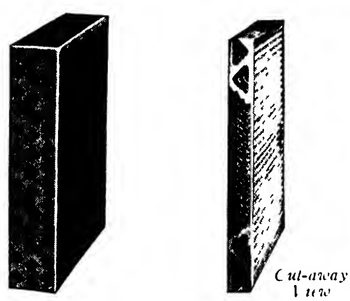
5202 Harvard Avenue, **Cleveland, Ohio**

ENGINEERS AND MANUFACTURERS OF AIR FILTERS EXCLUSIVELY

DIRECT FACTORY REPRESENTATIVES IN ALL INDUSTRIAL AREAS

DISTRIBUTORS IN PRINCIPAL CITIES AND TOWNS THROUGHOUT THE UNITED STATES

During more than a decade devoted exclusively to air filter engineering and manufacturing, a great deal about the control and elimination of dust, pollens and grit has been learned by AIR-MAZE engineers. Their design and development of a unique type of filter element construction, embodying distinctive advantages, has been considered a worthy contribution to the air filtering science and has resulted in wide acceptance of AIR-MAZE air filters in all fields of application.



4 in Thick Panel *2 in Thick Panel*

Views showing open end edge of AIR-MAZE Permanent Cleanable Panel Filters

Note Advantages Made Possible by Air-Maze Scientific Construction:

Costs Little to Clean--The separating layers and exact spacing of baffles permit free washing action between and around all baffles. Thus, cleaning and oil charging operations may be easily and economically performed.

Great Dust Capacity—On the face are coarse baffles exactly spaced, which evenly impinge bulky particles. Progressively inward are finer meshes. Instead of one or two progressions in density there are many; consequently, great dust capacity is possible enabling long operating periods before servicing is required.

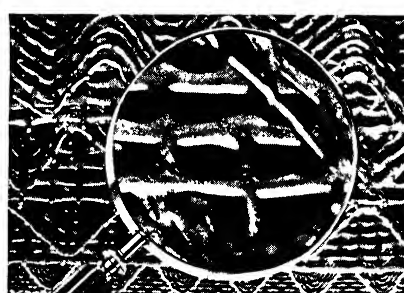
Vibration Proof—Vibrations in service cannot shake filter media out of position—the uniform density remains *permanently* perfect; **no replacements are necessary!**

No Special Oil Required In AIR-MAZE each square inch of the panel handles the same amount of air. Properly drained, as directed, *no oil will be carried through*, since the air velocity is held evenly low and uniform in every square inch. For that reason no especially adhesive oil is needed.

Efficiency—Tests under varying conditions, both in laboratories and field operations, show air filtering efficiency of from 98.50 to 99.83 per cent.

No Clogging--Because AIR-MAZE panel filters are easy to *completely* clean and since the exact density enables uniform deposit of dust, no clogging can occur.

Adaptability In addition to air conditioning and power equipment installations AIR-MAZE panel filters are effectively used in humidifiers, water eliminator units, paint spray-booths, oil separators, and other applications where specific problems and unusual requirements are easily handled by adaptations of the panels. AIR-MAZE panels will be made to fit frames of existing installations and can be furnished with locking handles and latches, or with flanged edges and lift handles.



Magnified Section of "Loaded" AIR-MAZE Air Filter Element. Note that dust has been quite evenly impinged on the wires. No obstructed spaces can be seen. This feature accounts for the Low Pressure Drop and Non-clogging characteristics of AIR-MAZE.

TECHNICAL INFORMATION

Sizes—All sizes and thicknesses are available, two and four inch thick panels are the accepted standard. Installations using large sizes of these permanent panels are surprisingly low in cost.

Capacity—Recommended air capacity is 1½ to 2½ cfm per square inch. Thus, the capacity of a 20 x 20 in. panel is 600 to 1000 cfm. Normally, 2 cfm per square inch should be used.

The Air-Maze Corporation

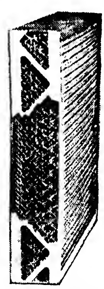
5202 Harvard Avenue, **Cleveland, Ohio**

Resistance—For 2 in. thick panels the resistance varies from 0.089 in. to 0.10 in. H_2O when handling 2 cfm per square inch of filter area (288 fpm velocity); and for 4 in. thick filters the resistance varies from 0.121 in. to 0.140 in. H_2O at 2 cfm per square inch (288 fpm velocity); the variation being in accordance with the different types of filter media construction available. To obtain specific restriction data write for graph RE-2A.

Construction—AIR-MAZE filters are of patented construction consisting of a maze of alternately placed and exactly spaced flat and crimped galvanized wire screens of selected meshes; these are arranged with precision so as to create graduated and progressive density, and to positively embody the baffle impingement principle. The filter element is enclosed in a heavy gauge metalescent enameled steel frame having an open end to simplify servicing.

EASY TO CLEAN AND CHARGE

Wash out filtered matter in a pan of cleaning solution. Similar pan may be used for charging.



Cut-away View

From a flat surface raise one end and let it drop sharply several times. This facilitates drainage.

After cleaning and also after charging, set panel on edge, with open end down, to drain.

Cleaning—Wash panels in a tank of any suitable solvent which will cut oil and accumulated matter, yet will not affect the galvanizing. Kerosene may be used or a non-inflammable solution prepared with AIR-MAZE cleaning powder and water. *Important! Be sure that panels are thoroughly drained before charging!*

Charging—Immerse in oil; or spray with oil. Any inexpensive oil of *S.A.E.* 20 viscosity or heavier is suitable.

If using the immersion method, panel should be removed from charging tank, placed on open end edge, and drained thoroughly. Filter is then ready for service, *having the performance and characteristics of a new unit.*

AIR-MAZE INSTALLATION FRAMES

Air-Maze Panel holding frames assure efficient, attractive installations.

AIR-MAZE panel holding frames are constructed of metalescent enameled heavy gage channel formed steel having rolled front edge and $\frac{3}{4}$ in. flanged back edge. A thick felt lining on inside of flange insures against air leakage when panels are in place. One frame may be used alone in single panel installations, or a group of many frames may be supplied, fixed together with felt seals between edges; thus a large bank of filter panels may be provided. Every unit is fitted with latches for the locking handles supplied on panels.

In determining frame sizes, $\frac{5}{8}$ in. is allowed over both width and length dimensions of the panel filters. This $\frac{5}{8}$ in. includes frame edge, clearance and felt seals between edges.

Specify AIR-MAZE—for all air filter installations and you will be assured of *efficient, economical* **performance.**

Engineering Service Available—The Air-Maze Engineering Department will gladly offer installation suggestions for special air filter applications.

Other AIR-MAZE Products—In addition to the panel types, Air-Maze Corporation also manufactures a complete line of circular shaped air filters for use in various Railroad, Industrial and Automotive applications.

Literature Available—Bulletins GPB-97, PPS-38, and PI-107R: covering complete panel filter details. Graphs G-11 and G-12A: showing results of tests made with A.S.H.V.E. Code Test Apparatus. Folder RR-18: describing railroad air filters; and complete catalogs covering the entire AIR-MAZE line.

AMERICAN AIR FILTER COMPANY INC.

INCORPORATED

1st Street and Central Avenue, **Louisville, Ky.**

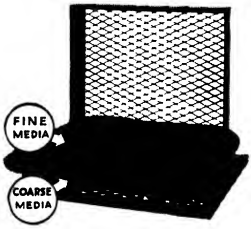
Representatives in Principal Cities

Dust Engineering—Dust Engineering is that branch of applied science which deals with the origin, nature and characteristics of the small solid air-borne particles called "dust," and the development of methods, processes and apparatus for its control or elimination.

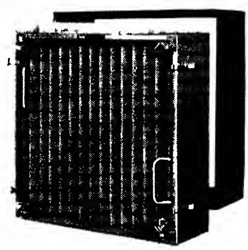
The American Air Filter Company, Inc., has had an important part in advancing the science of Dust Engineering. The efforts of its Research and Engineering Staff for the past twelve years have been devoted exclusively to the study of dust problems and the development of a complete line of air cleaning equipment for modern air conditioning, building ventilation and the control of process dust in industry.

American Air Filter products, therefore, not only embody the knowledge accumulated from years of constant research and the experience gained from designing, building and applying thousands of air filters, but are backed by ample technical and financial resources to insure their outstanding position in the Dust Engineering field.

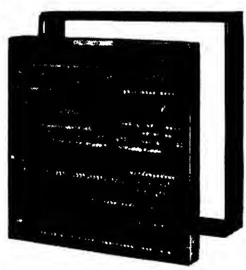
Products—American Air Filters are available for every condition, with operating characteristics and efficiencies to suit specific problems. In general, there are two distinct types based upon the "viscous film" and "dry mat" principles. Each type is made in several styles which differ in method of operation, servicing, space required and initial cost to meet the various conditions encountered in air cleaning problems. A discussion of various filter types will be found in the Technical Data Section under "Air Cleaners."



Renu-Vent Filter



Airmat Type PL-24 Filter



M/W 2 Filter

Throway Air Filter

Air filters are generally used for the removal of dust, dirt, bacteria and other foreign matter from the air and are applied to general ventilation, modern air conditioning, process dust control; for air compressors and Diesel Engines; mill motors, turbo-generators and other electrical applications; and for air or gas under pressure to remove entrained oil, moisture and dirt.

Air Filters In Air Conditioning—Filtered air is today recognized as essential in modern air conditioning. There are other important factors which contribute to our comfort such as temperature, air movement and humidity, but science today emphasizes the prime necessity of pure air for health and efficiency.

Air cleaners have, of course, always been considered an integral part of large central systems. These are usually of the fully automatic type such as the Multi-Panel filter, illustrated in the accompanying photograph.

There are now available to manufacturers of unit air conditioners moderate priced unit filters such as the Renu filter, the Throway filter, and other types of filters illustrated on this page.

The Renu filter is an entirely new departure in air filter construction. It consists of a permanent metal frame provided with a removable cover and renewable filter pad. The cover

Standard Viscous Unit Filter

is easily removed without the use of tools, and filter pad can be lifted out and replaced with a new one at very small expense.

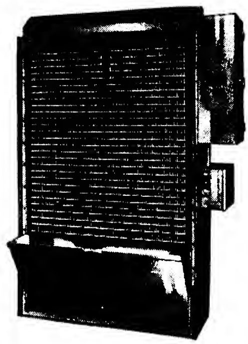
The Throway filter, as the name implies, is designed to be discarded after it has served its maximum period of usefulness and replaced with a new filter unit. The Filter pad is enclosed in a perforated cardboard container which makes it possible to readily dispose of the dirty filter by burning it.

There is probably no single item which costs as little and may mean as much in the design of an air conditioner as air filtration. These units are furnished in any dimensions or shapes desired—usually in units handling 400 cfm and from 2 in. to 4 in. thick. They are usually made in the following sizes—20 x 20 in., 16 x 25 in. and 16 x 20 in. High cleaning efficiencies can be secured, with a resistance to air flow ranging from 1/16 in. to 3/8 in. water gauge.

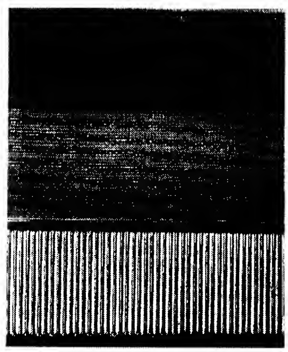
Automatic Self-Cleaning Air Filters—The American line of automatic air filters is among the most complete ever offered. Proved in principle and performance by years of actual service.

Armored Multi-Panel Filter—Introduces an entirely new and unique panel construction to further improve the already outstanding performance of the famous Multi-panel air filter. The new "armored" panel with its three stages of air cleaning maintains the present high efficiency and normal operating and resistance of the Multi-panel filter and offers the added advantage of handling excessive line concentrations or heavy dust loads without clogging.

Unimatic Self-Cleaning Filter—Developed especially for small air volumes where hand operation or push button control is practical. Fills the need for an intermediate filter between manually maintained units and the fully automatic Multi-panel. The Unimatic offers numerous advantages, one of which is the single pass curtain which makes possible the use of progressively packed media with its low resistance and great dust holding capacity.



Armored Multi-Panel Automatic



Section of Multi-Panel filter curtain showing unique construction of new Armored Panel. Dark portion of screen is bakelite-fibre coated. The bright uncoated screen lies immediately behind the Armored section of the preceeding panel and provides the second or intermediate stage of air cleaning. The Armored section is at the bottom of the panel.

Unimatic Self-Cleaning Filter

Standard Viscous Unit—The American Unit Air Filter incorporates the time tested unit principle of construction. Each unit consists of a standard steel frame and interchangeable cell equipped with automatic latches to facilitate removal for cleaning and recharging.

Airmat Filter Dry Type—The filtering media in this type is the Airmat sheet, a dry filter mat composed of thin sheets of gauzy, cellulose tissue. The Airmat sheets are supported in screen pockets mounted in a unit frame of box-like construction. These unit frames can be set up to meet any capacity requirement or space condition. Airmat sheets are renewable—their life depending on the dust condition and hours of service.

Airmat filters are used both for comfort and industrial air conditioning. In the latter field they are particularly well adapted for the recovery of valuable dusts and for abating the dust nuisance prevalent in so many industrial plants. They are available in two types, the PL-24 as illustrated and the Well Pocket type unit.

Our standard data books and catalogues are in most engineering files or libraries. We will be glad to furnish complete data to engineers or manufacturers.

Coppus Engineering Corporation

339 Park Avenue, **Worcester Mass.**

MANUFACTURERS OF AIR FILTERS, STEAM TURBINES, GAS BURNERS, FORCED DRAFT BLOWERS, COOLING FANS

"COPPUS AIR FILTERS PASS CLEAN AIR"

The Coppus Unit Air Filter (patent No 2050508 and other patents pending) is of the dry type using as filter material all-wool felt It consists of a distender frame (*C*, Fig 2), a filter "glove" (*E*, Figs. 1 and 2) and a retainer grid (*B*, Fig 1) The edges of the retainer grid form a reenforced sheet metal box (*A*, Fig 1) for protection of the filter element

The edges of the filter glove are reenforced on all four sides assuring an air tight seal against by-passing of dirty air By tightening the wing studs which hold the distender frame and the retainer grid together, the filter glove is stretched and held tautly inside of the filter box, giving the pockets a tapered shape so essential for an even air flow.

This design has the advantage of providing an effective filter area entirely unobstructed by wire or screen supports. Cut, Fig 3 shows the tapered filter pockets on the clean air side The filter glove can be readily replaced without removing the unit filter from the installation No auxiliary frames for insertion of the filter cells are required as the completely assembled unit filters can be bolted together to a filter bank of any desired size

All metallic parts are rust-proofed and Duco Painted

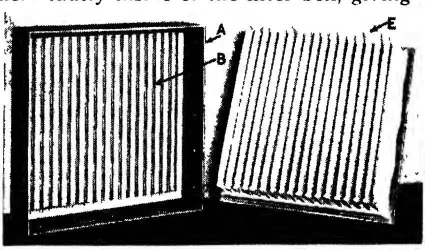


Fig 1

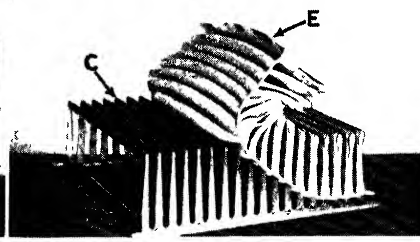


Fig 2

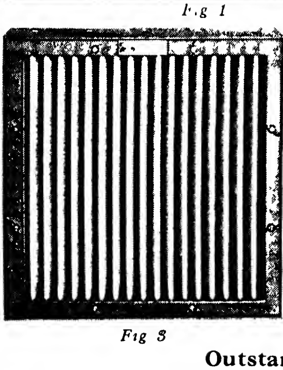


Fig 3

Specifications

Normal Rating: 800 cfm
Resistance when clean: 0 2 in W G
Dust Arrestance (cleaning efficiency): 99 61 per cent (Tested in accordance with A S H V E Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work)
Dimensions: 20 by 20 in by 5¾ in.
Weight per unit: 25 lb

Outstanding Advantages

1. It has an exceptionally high dust arrestance.
2. It maintains a high dust arrestance even under diverse conditions of neglect
3. Its operation is not impaired by atmospheric conditions
4. It is a Medium Air Resistance Type (Class C) according to the A S H.V.E. Code for Air Cleaning Devices.
5. It is easily and quickly cleaned without removing the filter element
6. Its cost of upkeep is very low because the *permanent* filter element is reconditioned periodically with a vacuum cleaner
7. It combines scientific knowledge and practical engineering methods with highest quality of material and workmanship.

Write for Complete Bulletins

Cleaning Filter Elements with Portable Vacuum Cleaner

Davies Air Filter Corp.

396 Fourth Avenue, **New York, N. Y.**

Air Conditioning, Process, Building, Industrial Filters

DAVIES TYPE M FILTER

Specially designed for ventilating and air conditioning systems where high efficiencies are required and space for air filters is small. Sturdily built frames, 14 gauge cold-rolled steel, rust-proofed under finishing coat of baked enamel. Other finishes or metals supplied on request.

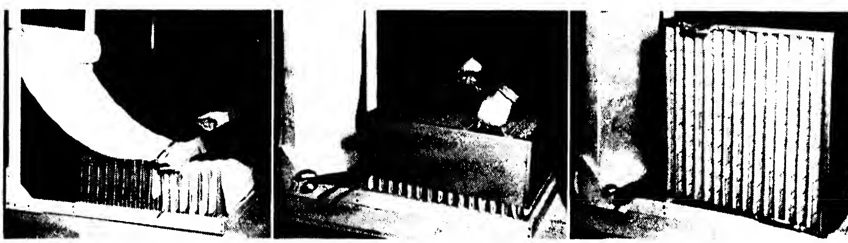
Two frames with patented interlocking members, hold the filtering medium on wire mesh and produce an air-tight joint. Large filtering area is a special feature.

The two filter frames are placed in a steel supporting frame work; heavy felt gaskets prevent leakage and insure high efficiency.

Type M filter medium is specially processed cotton with large dust holding capacity—supplied in rolls sufficient for 15 complete refills, 21½ feet each.

Specifications—Type M Filter

Filter Cell—overall	24 x 24 x 8 in.
Effective Filtering Area	40 sq ft
Capacity recommended	1200 cfm
Maximum Capacity	1500 cfm
Resistance—clean filter { 1200	.10 in. W. G.
Resistance—clean filter { 1500	.12 in. W. G.
Cleaning Efficiency	98%



Method of loading: Heavy loader rods carry the cotton material into place. After a little practice, a filter can be loaded in forty seconds—special mechanical knowledge is not required.

AIRPLEX RENEWABLE FILTER

Airplex filter medium is cotton fibre, specially processed and lightly glazed. Each filter contains 30 sq ft of filter medium and gives 500 to 1000 hours active service. Functions efficiently in temperatures below freezing and up to 200 F. Not affected by temperature or humidity—will not disintegrate. Filters can be cleaned several times before they are discarded.

Each filter is a complete cartridge—replacement can be made quickly, hence is not neglected.

Segment of Airplex Filter Showing Corrugated Spacers

Standard Sizes Airplex Filter

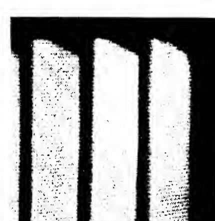
Process, Industrial, Building	20 x 20 x 4 in.
Air Conditioning Units {	20 x 20 x 2 in.
Air Conditioning Units {	16 x 25 x 2 in.

WASHABLE FILTERS—TYPE HG

Filter medium of fine spun hair glass closely packed and secured between two sheets of galvanized wire cloth; these long flexible glass fibres do not break and cannot be drawn into the air stream.

Filter element supported in a steel frame, rust proofed or galvanized as required.

Glass wool, being chemically inert, is not attacked by gases or liquids, will not rust or disintegrate, will last many years. Water, hot or cold, with or without grease solvents, used for cleaning, depending on type of air pollution.



Segment of H. G. Filter Showing Wire Mesh Holder

Stock Sizes—Type HG Filter

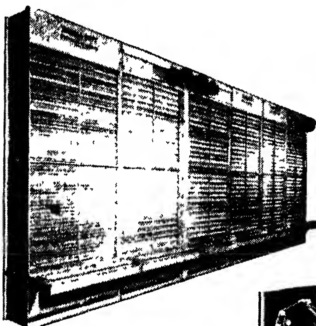
Frame Size	Depth	Filtering Surface	Capacity cfm	Pressure drop
19½ x 19½ in.	2 in.	480	500	.12
19½ x 19½ in.	3¼ in.	1053	1000	.12
16 x 25 in.	3½ in.	1053	1000	.12

Independent Air Filter Company

228 North LaSalle Street — **Chicago, Illinois**

Representatives in Principal Cities

Manufacturers of AIR CLEANING EQUIPMENT for General and Commercial Ventilating and Air Conditioning systems. Advisory Service without obligation.



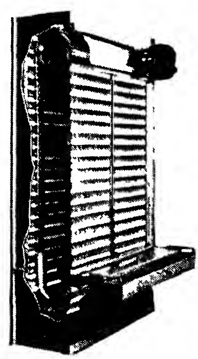
"DOUBLE-DUTY"

Self-cleaning ● Non-clogging ● Automatic

AIR FILTERS

Designed for industrial plant and large building air filtration Operates on the True Impingement principle.

Dust particles are freely and completely released when curtain plates immerse in oil bath.



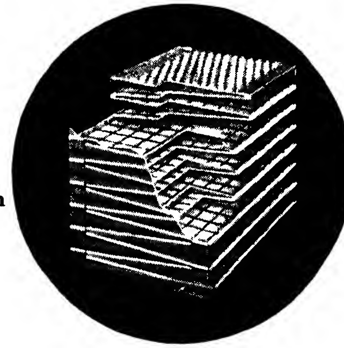
Cutaway view illustrates how front side of curtain impinges dust on up travel and rear side on down travel—an exclusive "Double-Duty" feature, giving eight acute deflections of air through both curtains

Cannot clog Constant air flow regardless of dust content of air

Oil entrainment impossible, due to angular design of steel louvre plates Oil needs never be changed—merely replenished

Standardized sectional construction meets all capacity requirements

Write for Bulletin D-110-AS



"KOMPAK"

Low velocity ● Dry fabric ● Renewable medium

AIR FILTERS

For general ventilating and air conditioning systems "KOMPAK" affords maximum area of filter medium in minimum space, large dust holding capacity, high cleaning efficiency, long useful life, low structural resistance to air flow and low refill cost. Each unit contains 28 sq ft of filtering medium Frames are of rust-proofed steel construction Changing of filter medium requires less than 5 minutes per unit and is only necessary every 3 to 6 months, depending on dust content of air.

Write for Bulletin K-120-AS

Owens-Corning Fiberglas Corporation

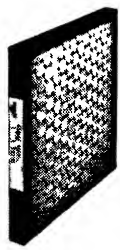
Toledo, Ohio

AIR FILTERS FOR APPLICATION TO RESIDENTIAL, COMMERCIAL *and* INDUSTRIAL HEATING, VENTILATING *and* AIR-CONDITIONING SYSTEMS

DUSTOP

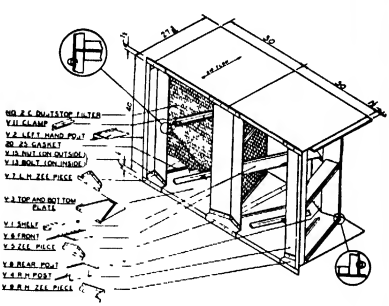
"FIBERGLAS" MEDIUM - ADHESIVE-COATED - REPLACEMENT TYPE

The Dust-Stop Air Filter consists of a series of non-combustible mats of Fiberglas, progressively packed—coarse glass fibers of lesser density at the intake and fine glass fibers of greater density at the discharge face

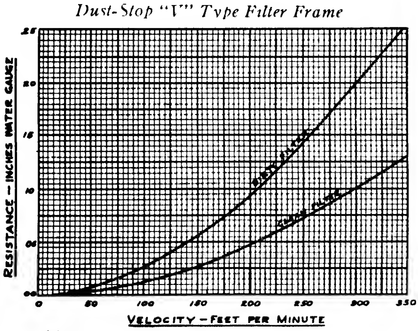


Mats are coated with non-evaporating fireproof adhesive having extraordinary wetting power, will retain viscosity under operating temperatures ranging from 15 F below to 300 F above zero, will not flow off or charge the air with adhesive . . Dust-Stop Air Filters are engineered to provide high efficiency at low cost of installation and maintenance.

Efficiency—97 per cent (Tested according to A S H V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work)



Dust-Stop "V" Type Filter Frame

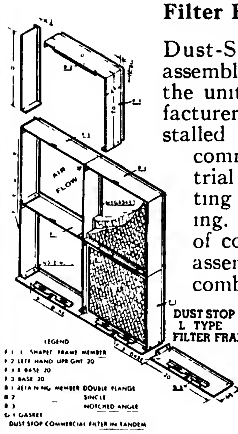


VELOCITY-RESISTANCE CURVE

Manufacturers' Acceptance

Dust-Stop Air Filters are widely accepted by manufacturers of blowers, gravity and forced warm air heating systems, and air conditioning equipment of all kinds

Filter Frames



Dust-Stop Filter frame assemblies are standard in the units of many manufacturers and are also installed by engineers of commercial and industrial heating, ventilating and air conditioning. Frame members of cold-rolled steel are assembled vertically in combinations to accommodate for any cfm and space requirement

STANDARD SIZES FOR EQUIPMENT*

Standard Sizes (Nominal)	Ratings		Resistance Inches Water Gauge
	Cfm	Fpm	
20" x 25" x 2"	1000	300	0.095–0 10
20" x 20" x 2"	800	300	0 095–0 10
16" x 25" x 2"	800	300	0 095–0 10
16" x 20" x 2"	640	300	0 095–0 10

*Other standard sizes available.

Write Owens-Corning Fiberglas Corporation for information and data.

Research Products Corporation

1011 E. Washington Ave., **Madison, Wisconsin**

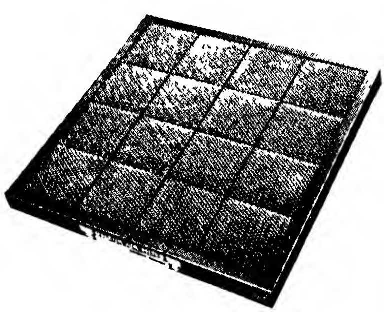
WALTON AIR FILTERS

High Dust Removal Efficiency—Low Air Resistance—Long Life

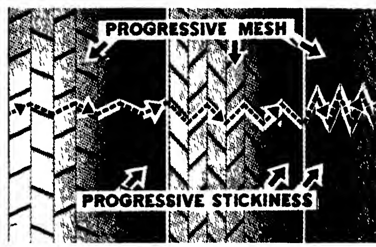
Walton air filters operate on the impingement principle, using an entirely new filter medium—expanded fibre. Large, uniformly distributed area holds large quantities of dust yet cleans effectively, with low air restriction. Air flowing through the filter impinges against more than 30,000 tiny baffles per sq ft, depositing dust, and changing direction 21 times.

To prevent clogging and insure uniform deposit of dust, the filter is made in three sections—at the inlet, coarse open mesh, middle and outlet sections progressively finer mesh, in each section a sticky compound is applied in progressive amount Progressive mesh and progressive stickiness reduce clogging and insure high holding capacity. Exposed surfaces are free from excess stickiness—clean to handle

The sticky compound will not drip or volatilize at high temperatures, nor gum at low temperatures, and is odorless



Walton air filters have been used successfully on warm air furnaces, ventilating and air conditioning systems, automobile air conditioning, industrial dust collectors, and wherever a filter is required for removing dust and pollen from the air



Diagrammatic cross section of Walton filter Three sections with progressively finer mesh and progressive stickiness Large dust particles are trapped by the coarse section, finer particles by the finer mesh—deposited uniformly throughout the filter

HIGH DUST REMOVAL EFFICIENCY

The Walton filter maintains high cleaning efficiency for all common dust sizes Because of its controlled mechanical construction, blow holes do not form and pass dirty air.

Efficiency of the filter actually increases as it becomes laden with dust. Dust particles deposited on the baffle surfaces absorb the sticky compound, and, in turn, retain additional dust particles. Vibration cannot settle or pack the filter and thus reduce its efficiency. Particles cannot break off and blow into the air stream.

WALTON AIR FILTER STANDARDS

Walton air filters are made in five standard sizes to fit most commonly used frames, special sizes for special requirements, and to standard or special specifications.

Standard Specifications

Recommended air capacity	250 to 350 cfm per sq ft of face area
Average efficiency (depending on type of dust)	86 5 to 100%
Holding capacity (standard dust)	132 grams per sq ft
Resistance when replacement is necessary at 250 cfm per sq ft of face area	0 18 in water
Resistance when replacement is necessary at 350 cfm per sq ft of face area	0 55 in water

Standard Sizes

20 x 25 x 2 in
20 x 20 x 2 in
16 x 20 x 2 in
16 x 25 x 2 in
15 x 20 x 2 in

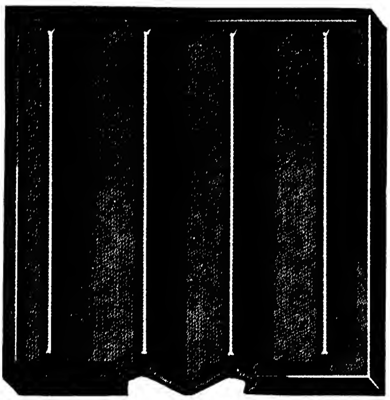
Special sizes for any purpose

Research Products Corporation engineers will gladly submit recommendations for any special applications that may be required.

H. J. Somers, Incorporated

Factory and General Office

6063 Wabash Avenue **Detroit, Mich.**



All Welded Vee Type

Somers Washable Air Filter

Somers Hair Glass Filters provide everything required in an efficient air-cleaning system. Consider these features: High rating for dust, soot and bacteria separation. Require no adhesive, coating or impregnation. Indestructible in normal service. Minimum Low Pressure Drop. Odorless and non-absorptive. Fireproof; Washable; Do not rot nor disintegrate; Permanent.

Somers Hair Glass Filters consist of a hot galvanized frame holding galvanized wire cloth packed with hair-spun glass strands. The glass strands are flexible, do not break up and cannot be drawn into air stream.

Hair-Glass, being chemically inert, has no facility of absorption; it cannot rust and lasts indefinitely in service. Water either hot or cold may be used to clean it, without impairing its efficiency.

These filters eliminate the necessity, the expense and the inconvenience of periodic replacement.

Somers Washable Air Filter—All Welded Vee Type—Stock Sizes (Partial List)

Frame Size Height and Length In.	Frame Depth In.	Filter Surface Sq In.	For Average Dry Filter Installations CFM	Wet Application where water sprays are applied against filter for humidifying CFM
15 1/2 x 24 1/2	3 5/16	1023	1023	511
15 5/8 x 24 5/8	3 1/4	1110	1110	555
16 x 21 1/2	3 1/16	816	816	408
16 x 25	3 1/8	1056	1056	528
16 x 25	3 1/4	1632	1632	816
16 x 25	3 1/4	1344	1344	672
16 x 25	3 1/4	1440	1440	720
16 x 25	3 5/16	864	864	432
16 1/2 x 24 1/2	3 1/8	800	800	400
18 x 18	3 5/16	864	864	432
19 x 20	3 5/16	1482	1482	741
19 1/4 x 19 5/8	3 1/4	1039	1039	519
19 1/4 x 20	3	1039	1039	519
19 3/8 x 19 1/2	3	936	936	468
19 1/2 x 19 1/2	3 1/4	1053	1053	526
19 1/2 x 19 1/2	2	480	480	240
19 1/2 x 19 1/2	3	936	936	468
19 1/2 x 19 1/2	3 1/4	1170	1170	585
20 x 25	3 5/16	1800	1800	900
20 x 30	3 5/16	1800	1800	900
20 x 20	3 1/8	1040	1040	520
20 x 30	3	1560	1560	780
20 x 20	3 1/4	1200	1200	600
20 x 20	2	480	480	240
20 x 20	3	840	840	420
20 x 20	3	960	960	480
20 x 20	3 1/4	1320	1320	660
20 x 25	3 1/4	1560	1560	780
20 3/8 x 20 1/4	3	550	550	275

Other sizes from 9 1/2 x 30 to and inclusive of 31 in. x 23 1/4 also available. Send for complete stock size list.
Frames zinc plated for 100 hour salt water spray test. Refill may be inserted if necessary.
Quotations and further engineering data, including master holding frame drawings will be sent on request.

Staynew Filter Corporation

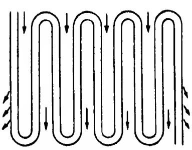
Air Filters for Every Purpose

6 Leighton Ave. **Rochester, N. Y.**

PROTECTOMOTOR AIR FILTERS

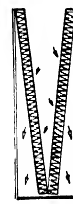
PROTECTOMOTOR DRY TYPE FILTERS

(For removing foreign matter from the air at atmospheric or other pressures, with various types for building ventilation, dust recovery, oxygen chamber and all air cleaning purposes.)



Cross Section Showing Panel Unit Construction

The fin or V-type construction is used in all Protectomotor dry filters. This basic principle permits (1) a large area of filtering medium to occupy the smallest possible space, and (2) the intake currents to move parallel to the filtering surface at low velocity. Protectomotor Dry Filters require no adhesive material to catch dust—*odorless* air is assured. Authorities agree that the positive dry filter is most efficient in stopping the smaller air-borne particles Protectomotor dry filters actually prevent the passage of bacteria.



Cross Section of 2 Multi-V-Type Cells in V formation

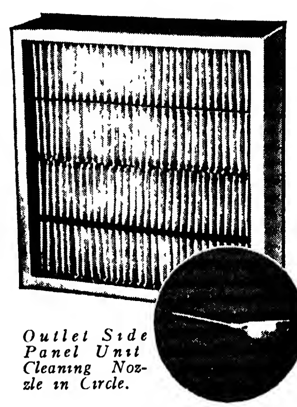
EASY TO CLEAN — LONG LASTING

Cleaning is easily effected by use of any vacuum cleaner with special nozzle. See illustration below.

Protectomotors operate from 3 months to a year without cleaning Panel units average slightly longer wear than Multi-V-Type units—several years at least without replacement.

Panel Units: Consist of Panel Insert and Frame The Insert is composed of two rows of 60 hollow loops or fins 6 in. deep, formed of rust-resisting embossed wire mesh, supported by a retaining grate of steel or aluminum and similar spacing grate Each row of fins is covered with a single piece of Feltex Filtering Medium, a felt-like material specially made for the application. Specifications below:

Overall Dimensions (Depth less locking keys) .... 20 x 20 x 6⅝ in
Size of Insert .... 19½ x 19½ x 6 in.
Capacity (average conditions) .... 800 cfm
Area of Filtering Medium .... 42 sq ft
Linear velocity of air .... 19 fpm
Resistance of clean filter to air flow 0 185 in water gauge
Total Weight .... 28 lb



Outlet Side Panel Unit Cleaning Nozzle in Circle.

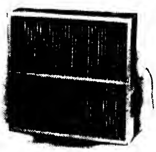
Multi-V-Type Filter Showing Ease of Cleaning

Multi-V-Type Units: Filtering medium (closely pressed cotton fibres between two sheets of cotton gauze) is arranged in patented V-shaped pockets in a fibre-board and pressed metal frame. These patented cells can be quickly and inexpensively replaced when worn out. Their arrangement makes possible an active filtering surface of 27 times face area. In certain installations the Multi-V-Type is more desirable than the Panel Unit because its construction fits the space better, or because it is lighter in weight per square foot of filtering area, or for reasons of economy. (Protectovent Window Ventilator, which supplies clean, fresh air to home or office, employs Multi-V-Type inserts). *Complete specifications mailed promptly on request.*

Staynew Filter Corporation

Air Filters for Every Purpose

6 Leighton Ave. Rochester, N. Y.



Wire-Klad Filter

Wire-Klad Units: Unique method of construction permits a high efficiency filter at low cost. Fins are reinforced on both sides with screen cloth, producing a rigid, long-wearing, *flame-resisting* filter that may be repeatedly cleaned with vacuum or compressed air, or flushed with water or liquid solvents. Made in 2 in. and 4 in. deep units.

Specifications 2 in. and 4 in. Units

Sizes	Capacity—Wool Felt	Capacity—No. 6460 Cotton	Filtering Area sq ft	
20 in. x 20 in.	800 cfm. @ 0.13 in. wg	800 cfm. @ 0.08 in. wg	2 in. 18.5	4 in. 38.5
16 in. x 25 in.	800 cfm. @ 0.12 in. wg	800 cfm. @ 0.075 in. wg	2 in. 18.5	4 in. 38.5
16 in. x 20 in.	600 cfm. @ 0.11 in. wg	600 cfm. @ 0.07 in. wg	2 in. 14.8	4 in. 30.7
20 in. x 25 in.	1000 cfm. @ 0.12 in. wg	1000 cfm. @ 0.08 in. wg	2 in. 23.0	4 in. 48.0

PROTECTOMOTOR AUTOMATIC FILTER

(For efficiently and economically filtering large volumes of air for all ventilating purposes)

Automatic Filter

Dust is removed from the air stream by impingement on *two* endless revolving curtains. The first curtain passes through an oil bath; the second, which does not pass through the oil bath, prevents any possible oil entrainment in the air stream. This second curtain, being slightly moist from entrained oil, serves as additional protection against dust passage. No other filter has the two-curtain feature.

Compressed Air Cleaners

Two additional exclusive features are to be noted. One is the compressed air cleaner system (see diagram), of two copper tubes drilled with air jets which blow off excess oil with foreign matter from the bottom of each curtain.

The other is the direction of curtain travel. The intake sides of both curtains move downward. Thus foreign matter is removed by the oil bath and compressed air before each curtain presents its outlet side, which moves upward, to the air stream.

Curtain travel is intermittent, moving 4¾ in. during the operation of the mechanism which is approximately 20 seconds each half hour. The mechanism is actuated by a ⅙ hp motor with a reliable timing device.

Sizes and Capacities

Two standard widths are made, 2 ft 9 in. and 4 ft 3 in., ranging in height from 4 ft to 13 ft by 3 in. steps. Capacities are from 2,025 cfm to 20,200 cfm for single units. Almost unlimited capacities may be secured by bolting together units.

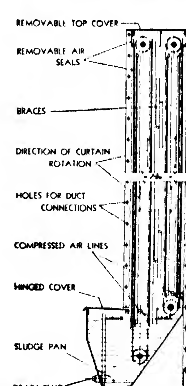


Diagram of Automatic Filter

Write for Catalog Mentioning Special Interests

PROTECTOMOTORS ALSO MADE FOR INTERNAL COMBUSTION ENGINES, COMPRESSORS, TURBO-GENERATORS, AIR TRANSMISSION LINES, ETC.

Young Regulator Company

Department G

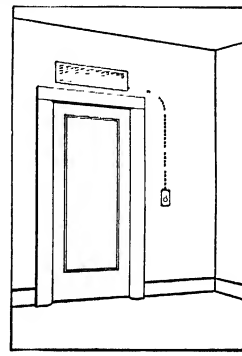
4500 Euclid Avenue, **Cleveland, Ohio**

Representatives in Principal Cities

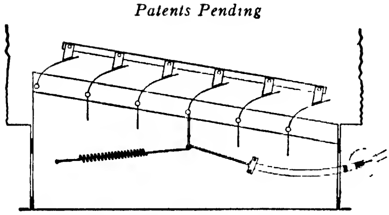
THE "YOUNG" REMOTE CONTROL SYSTEM

To Individually Control Volume of Air

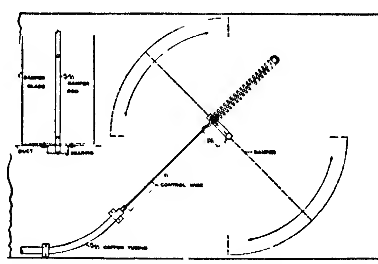
This practical device is particularly adaptable for hotels, office buildings, residences, public buildings and other buildings to manually control a damper or register at a remote distance up to 100 ft or more The damper or register is connected to the regulator by a stainless steel cable encased in control casing We recommend that it be placed in a 5/16 in copper tubing Knob sets damper or register at any desired position Knob is on a 2¾ in by 5 in escutcheon The indicator shows position of damper or register. The damper or register operates by pulling on the cable The stainless steel spring closes the damper or register and returns the cable when the indicator passes the "on" position



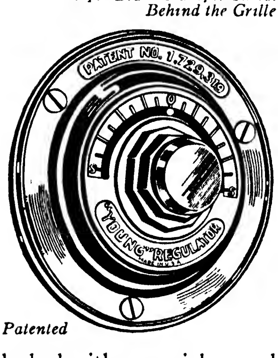
Patents Pending



Multiple Blade Damper Installed in a Duct Behind the Grille



Fastening Remote Control Regulator to Butterfly Damper



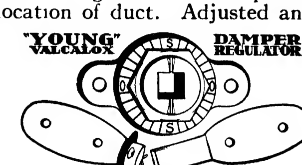
Patented

THE "YOUNG" REGULATOR

The "YOUNG" Damper Regulator is made in ⅜ in. size only and has five important features· (1) It locks securely. (2) It is tamper-proof (3) It gives positive action. (4) It is ornamental,—can be placed on partition walls. (5) It is adjusted and locked with a polygonal wrench or key.

THE "YOUNG" VALCALOX REGULATOR AND THE "YOUNG" FLUSH CUP REGULATOR

The "YOUNG" VALCALOX Regulator can be placed on partition wall or any location of duct. Adjusted and locked with a special wrench. It is much cheaper than the "YOUNG" Regulator



The "YOUNG" Flush Cup Regulator is used where regulator must be flush with the wall and is operated with a special wrench. We can also supply a concealed type regulator on which the plate must be removed in order to operate.

International Heating & Ventilating Exposition

THE AIR CONDITIONING EXPOSITION

Permanent Address—Grand Central Palace, **New York, N. Y.**

EXPOSITIONS HELD

The first in Philadelphia, 1930.
The second in Cleveland, 1932.
The third in New York, 1934.
The fourth in Chicago, 1936.
The fifth in New York, 1938

Subsequent Expositions will be held on alternate, even numbered years The next one will be Cleveland, January 22 to 26, 1940

These are held coincident with the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and are directed by the International Exposition Company, under the auspices of the A.S H V.E.

EXHIBITORS

Comprise leading firms in each phase of the industry number has varied from 150 to 327 exhibitors

EXHIBITS

These range from and comprise all the types of articles discussed or advertised in this copy of THE A S H.V E. GUIDE

1 *The* COMBUSTION *Group*
Furnaces, burners (coal, oil and gas), grates, stokers, boilers, radiators (various types), refractories and auxiliaries
2 *The* OIL BURNER *Group.*
3. *The* HYDRAULIC *Group.*
Water feeders, water heaters, pumps, traps, valves, piping, fittings, expansion joints, pipe hangers, etc
4. *The* STEAM HEATING *Group*·
Vapor heating and steam specialties.
5. *The* HOT WATER HEATING *Group.*
6. *The* AIR *Group.*
Warm Air furnaces and stoves, registers and grilles, cooling towers, air filters, motors, fans, blowers, conditioning equipment, ventilators (room and industrial types), unit heaters, etc.
7. *The* AIR CONDITIONING *Group*
Equipment which circulates and filters the air, in summer dehumidifies and cools, in winter heats and humidifies, and does all these in proper season for complete, all year round air conditioning.
8. *The* CONTROL *Group*·
Instruments of precision for indicating, controlling or recording temperature, pressure, volume, time, flow, draft or any other function to be measured
9. *The* REFRIGERATING *Group:*
Compressors, condensers, cooling apparatus, contingent apparatus and refrigerants.
10 *The* CENTRAL HEATING *Group:*
Apparatus and materials especially designed or adapted to the uses of central heating and central heating station supplies
11. *The* INSULATING *Group*
Structural insulators (refractory and cellulose materials), asbestos, magnesia clays and combinations thereof, pipe and conduit covering, etc., weather-stripping, etc
12 *The* MISCELLANEOUS *Group*·
Electric Heaters, boiler and pipe repair alloys, liquids and compounds, tools of all kinds, and equipment not specifically included in the above groups, but related thereto
13. *The* MACHINERY AND GENERAL EQUIPMENT *Group*
14 BOOKS AND PUBLICATIONS

VISITOR ATTENDANCE

Comprises a registered attendance invited to the exposition and includes:
(Figures are 1936 analysis)

INDUSTRIES

Governmental	404
Distributional Channels Contractors, Dealers, Jobbers, Supply Houses, 32 classifications	7701
Home Owners	474
Industrial Users, 49 classifications	10,791
Professional and Service Organizations, 23 classifications	2095
Public Utilities	1459
Real Estate Management and Operation, 10 classifications	1418
Educational Institutions	1935
Miscellaneous	1466
TOTAL	27,743

OCCUPATIONS

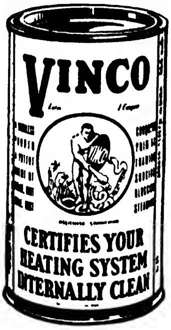
Executive (44 titles)	12,437
Construction (16 titles and trades)	2606
Operation (44 titles and trades)	4316
Technical (64 titles)	4477
Not Classified including Educators, Publishers, Home Owners, etc	3907
TOTAL	27,743

The registered attendance at the 1938 Exposition in New York was nearly 40,000, but the analysis of industries and occupations has not been completed at the time this goes to press

Industrial Expositions in America lead the expositions of the world in style, business effectiveness, industrial influence and educational value. This Exposition stands among the leaders in Industrial Expositions in America. It is an educational institution which biennially brings together the research developments and improvements in equipment and materials for use in heating, ventilating and air conditioning all types of buildings.

The Vinco Company, Inc.

305 East 45th Street **New York, N. Y.**



Boiler Cleanser
3, 5, and 10 lb cans

VINCO BOILER CLEANSER

A positively harmless insoluble powder cleaner for new, remodeled and old heating systems. A unique, scientifically processed compound on a special formula not to be confused with other powder boiler cleaners.

What Vinco Does

VINCO permanently removes oil, grease, scale, rust and dirt from the internal surfaces and from the boiler water *without the labor of blowing boilers over the top.*

By this thorough cleansing Vinco stops foaming, priming, surging, and slow steaming.

How Vinco Works

Each minute grain of VINCO powder absorbs several times its own weight of oil, rust and dirt. These larger grains of absorbed impurities then settle and are drained through the bottom, according to directions on each can.

Our Three-Fold Guarantee

1. VINCO contains no potash, lye, soda of any kind, oil, acid, or other harmful ingredients
2. Purchase price is refunded if results are not as claimed when VINCO has been used according to directions.
3. Your time, money and comfort are further safeguarded by

Our Free Laboratory Service

It saves thousands of dollars by analyzing the boiler water *before* making needless mechanical changes. If desired, the boiler water is again examined *after* the VINCO treatment is completed, and "certified chemically correct" for boiler operation

VINCO SUPERFINE LIQUID BOILER SEAL

A different liquid seal. Unique in that it does not induce priming and foaming It has no unpleasant smell. Makes speedy and permanent repairs of boiler and heating system leaks. Fine to tighten up new jobs. Directions simple

Liquid Boiler Seal
1 qt cans only

Quantities

Steam and Vapor Systems—Use 1 quart VINCO Liquid Boiler Seal to each 6 sq ft *grate* area.

Hot Water Systems—Use 2 quarts VINCO Liquid Boiler Seal to each 6 sq ft *grate* area.

VINCO SOOT-OFF

Non-explosive, thoroughly safe. No black smoke—no fire hazards Destroys soot from coal, oil or gas burning heating equipment. Easier, cleaner and quicker than brushing Makes short work of jobs too hard for brush or scraper. Cleans fire pot, flues and chimney in one simple operation.

Soot-Off
1 lb cans only

Ask for your copy of the VINCO Manual on the routine care of heating systems.

VINCO SPECIFICATIONS

Insist that Vinco be Used—Correctly—in Every Heating System You Specify or Operate.

These Tags Used by the Following Boiler Manufacturers on All Their Boilers, Give Proper Specifications for Cleaning All New, Remodeled and Old Steam, Vapor, and Hot Water Heating Systems.

AMERICAN RADIATOR COMPANY — Burnham Boiler Corporation — GILBERT & BARKER MANUFACTURING CO.

INTERNATIONAL HEATER COMPANY — NATIONAL RADIATOR CORPORATION — RICHMOND RADIATOR CO.

THE TITUSVILLE IRON WORKS COMPANY — WATERFILM BOILERS

(Face of Tag)

CAUTION

Do not use as a cleansing agent soda or any alkali, vinegar or any acid in this boiler. USE VINCO

DIRECTIONS FOR CLEANING

After the system is tested and tight, use the proper quantity of VINCO listed on the reverse side of this tag and follow directions given on each can of VINCO Satisfactory results will be obtained in record time with minimum expense

(Reverse of Tag)

QUANTITIES OF VINCO (In Pounds) REQUIRED FOR HEATING SYSTEMS

Note that quantities are based on actual installed radiation, not on boiler capacity.

For systems having	*For Steam or Vapor Systems To Prevent or Cure Priming and Foaming *And/or Hot Water Heating Systems Maintained at Approximately 200 Deg or Above	Annually, To Remove Rust Scale Dirt and for (Gravity) Hot Water Systems
Up to 350 sq ft of radiation	3	1½
351 " 600 " " " "	5	2½
601 " 1100 " " " "	8	4
1101 " 1400 " " " "	10	5
1401 " 1800 " " " "	13	6½
1801 " 2100 " " " "	15	7½
2101 " 2700 " " " "	18	9
2701 " 3100 " " " "	20	10
3101 " 3700 " " " "	23	11½
3701 " 4200 " " " "	26	13
4201 " 4600 " " " "	28	14
4601 " 5000 " " " "	30	15
5001 " 5300 " " " "	31	15½
5301 " 5600 " " " "	32	16
5601 " 5900 " " " "	33	16½
5901 " 6200 " " " "	34	17
6201 " 6500 " " " "	35	17½
6501 " 6800 " " " "	36	18
6801 " 7100 " " " "	37	18½
7101 " 7400 " " " "	38	19
7401 " 7700 " " " "	39	19½
7701 " 8000 " " " "	40	20
8001 " 8300 " " " "	41	20½
8301 " 8600 " " " "	42	21
8601 " 8900 " " " "	43	21½
8901 " 9200 " " " "	44	22
9201 " 9500 " " " "	45	22½
9501 " 9800 " " " "	46	23
9801 "10100*" " " "	47	23½

*Above 10100 sq ft use an additional pound Vinco for each additional 300 sq ft of actual installed radiation

(In the case of steam heating systems, never blow a boiler under pressure)

VINCO FOR OLD SYSTEMS

Annual cleaning of the old heating system adds years of life to the boiler, prevents rust deterioration and saves much fuel and fire attendance.

McDONNELL & MILLER

Manufacturers of McDONNELL *Boiler Water Level* CONTROLS

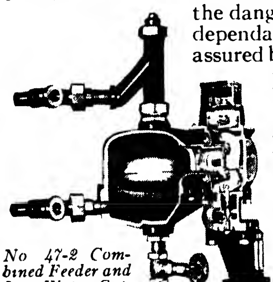
General Offices: Wrigley Building, **Chicago, Ill.**

"Doing one thing well"

PRODUCTS:

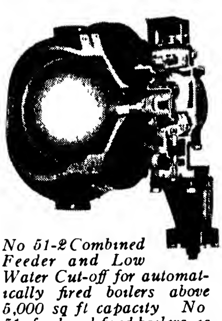
Boiler Water Feeders; Combined Boiler Water Feeders and Low Water Cut-offs; Low Water Cut-offs; Combined Low Water Cut-offs and Pressure Controls; Electric Water Valves; Pump Controls; Low Water Alarms; Constant Level Valves; Humidifier Water Valves; and related equipment.

Boiler Water Feeders— McDonnell Boiler Water Feeders (Nos 47, 147, 51, 53 and 101) supply water to boilers whenever the boiler water level tends to drop to the danger zone Their dependable operation is assured by the McDonnell "Cool Feed Valve" (Patent No 1,934,486) which prevents lime and scale formation —by packless (bellows) construction; stainless steel valves, the McDonnell "Quick-Hook-Up" (Patent No 1,997,-785), built-in strainer and many other refinements fully described in the McDonnell literature.

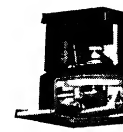


No 47-2 Combined Feeder and Low Water Cut-off with "Quick-Hook-Up" for automatically fired boilers below 5,000 sq ft capacity No 47 for hand fired boilers in same range is identical except for the No 2 cut-off switch (See below)

Feeders and Cut-offs Combined—Nos 47-2, 147-2, 51-2 and 53-2 for automatically fired boilers, are the Nos 47, 51 and 53 with the McDonnell No 2 Switch as illustrated In these combinations the feeder takes care of all normal operation, and the switch stands by to cut off the burner, if an emergency should arise such as extreme priming or foaming or failure of water supply Another type of feeder cut-off combination meeting special conditions is found in the No 67 with No 101, as explained under the illustrations at the right

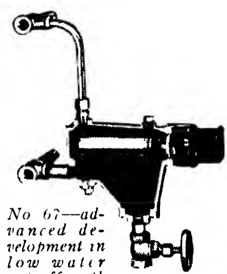


No 51-2 Combined Feeder and Low Water Cut-off for automatically fired boilers above 5,000 sq ft capacity No 51, for hand fired boilers, is the same except for No 2 switch No 53 and No 53-2 are similar in appearance but of heavier construction for higher pressures (See tables 1 and 2)



No 2 Low Water Cut-off Switch used on Nos 47, 147, 51 and 53 Safety Feeders Has high voltage cut-off contacts and low voltage alarm contacts Can be furnished on feeders or installed easily after feeders are in place on boilers

Low Water Cut-offs—The No 67 takes care of all cut-off requirements for boilers with steam pressures up to 25 lb Its operation is restricted to stopping the burner when low water threatens the boiler. However it is furnished with an extra switch, as explained under the illustration, for operating the No 101 Electric Feeder so that the feeder may be added at any time without price penalty



No 67—advanced development in low water cut-off with McDonnell Quick-Hook-Up Has extra switch which closes on small drop of water line, without cutting off burner, to complete circuit to No 101 Electric Water Feeder (See No 101) Note deep sediment chamber, removable clean out plate, and tapping which provides for No 13 Wireless Pressure Limit Control

For boilers with pressures from 25 lb up to 150 lb, the McDonnell No. 150 is the cut-off to use It can be used as either a low water cut-off, a pump control, a low water alarm, or for any combination of these three functions

Service recommendations covering McDonnell Boiler Feeders, Low Water Cut-offs and Feeder Cut-off Combinations are given in the table at the top of the opposite page which is followed by typical specifications. If there is any question as to the proper equipment or hook-up for handling a given condition, our engineering department will be glad to work with you in helping you to arrive at the best solution to your particular problem. Complete installation instructions are packed with each McDonnell product.

SERVICE RECOMMENDATIONS

Boiler Size in Square Feet	Steam Pressure	McDonnell Product to Use
Table 1—For Hand Fired Jobs		
Up to 5,000	Under 15 lb	No. 47 Water Feeder
Up to 5,000	15 to 35 lb	No. 147 Water Feeder
Above 5,000	Under 35 lb	No. 51 Water Feeder
Any size	35 to 75 lb	No. 53 Water Feeder
Any size	35 to 150 lb	No. 150 as a pump control
Table 2—For Automatically Fired Jobs Water Feeder—Low Water Cut-off Combinations		
Up to 5,000	Under 15 lb	No. 47-2 Feeder-Cut-off Combination
Up to 5,000	15 to 35 lb	No. 147-2 Feeder-Cut-off Combination
Up to 5,000	Under 25 lb	No. 67-101 Electric Feeder-Cut-off Combination
Above 5,000	Under 35 lb	No. 51-2 Feeder-Cut-off Combination
Any size	35 to 75 lb	No. 53-2 Feeder-Cut-off Combination
Any size	35 to 150 lb	No. 150 as pump control and Cut-off
Table 3—Low Water Cut-offs		
Any size	Under 25 lb	No. 67 Low Water Cut-off
Any size	25 to 150 lb	No. 150 Low Water Cut-off
Low Water Cut-off—Pressure Control Combination		
Any size	Off at 3 lb On at 1 lb	No. 67-13 Combined Cut-off and Pressure Control

Typical Specifications for Boiler Water Feeder or Combined Feeder and Cut-off

Furnish and install complete in every essential detail for each boiler unit. Automatic Boiler Water Feeder—and Low Water Cut-off (if automatically fired)—equipment as manufactured by McDonnell & Miller, Wrigley Bldg., Chicago.

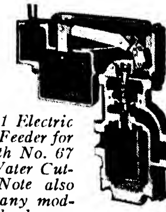
Boiler Water Feeder to be (insert*) of the manufacturer's most improved design, with all working parts isolated from the steam and hot water zone by packless sylphon construction. Float power to be multiplied through a leverage mechanism which contains a self-centering roller directly above valve stem. All bearing points, pivots, and levers to be outside of the steam and water zone. Valve cone and seat to be stainless steel, removable and renewable. A strainer to be incorporated in the design and to have a solderless basket mounted on a flange for easy removal. Feeder to be installed complete with all piping, valves, fittings, and specialties, as indicated by descriptive diagram in the manufacturer's construction bulletin.

Upon completion of the installation, the contractor to place this automatic equipment in successful operation, subject to acceptance and approval of the architect's .... engineer.

*If job is hand fired, insert: "No. 47," "No. 147," "No. 51," or "No. 53," as indicated by service conditions in Table 1.

*If job is automatically fired, insert: "No. 47-2," "No. 67-101," "No. 147-2," "No. 51-2" or "No. 53-2," as indicated by service conditions in Table 2.

When combined units No. 47-2, No. 51-2, or No. 53-2 are specified, add the following paragraph to specifications: "Make electrical connections for low water switch to control fuel supply equipment, so that the fuel is automatically shut off by the low boiler water control. Control wiring to be of flexible armored cable. All wiring to meet the requirements of the City Electrical Inspection Department and the *National Board of Fire Underwriters*."



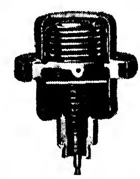
No. 101 Electric Water Feeder for use with No. 67 Low Water Cut-off. Note also that many modern boiler-burner units have "built-in" McDonnell Low Water Cut-offs which may be used to operate No. 101. The No. 101 has stainless steel valve, positive powerful closure against water pressure up to 150 lb.

Typical Specifications for Low Water Cut-off

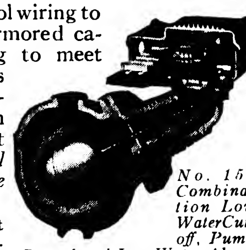
Furnish and install on each boiler in accordance with the manufacturer's instructions, a McDonnell & Miller No. (insert†) Low Water Cut-off, to be float operated and to have packless construction. Control wiring to be of flexible armored cable. All wiring to meet the requirements of the City Electrical Inspection Department and the *National Board of Fire Underwriters*.

†Insert at this point "No. 67" or "No. 150" as indicated by service conditions in Table 3.

Literature contains complete descriptions, capacity chart, installation instructions, wiring diagrams, dimensions, etc.



No. 13 Pressure Limit Control for quick and easy mounting on No. 67 Low Water Cut-off. Its plunger forces down the float of the low water cut-off when excess pressure develops, stopping the burner until proper operating pressure is restored. No wiring. May be used on "built-in" cut-offs which provide tapping for it.



No. 150 Combination Low Water Cut-off, Pump Control and Low Water Alarm built to stand up in high pressure service. Float operates two switches. One closes pump circuit when water level falls; the other cuts burner circuit and completes alarm circuit when water level falls to danger zone. Maximum steam pressure, 150 lb.

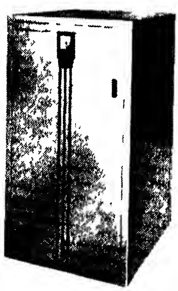
AMERICAN RADIATOR COMPANY

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

40 West 40th Street, New York, N. Y.

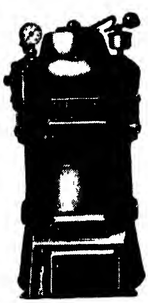
PRODUCTS FOR EVERY HEATING REQUIREMENT

IDEAL OIL BURNING BOILERS



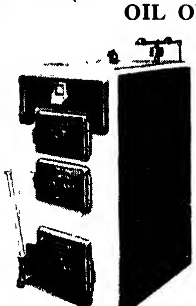
Automatic oil-burning boilers, available in three models, with standard jacket or extended jacket for concealing burner. All equipped with low-water control, air-cell insulation and provision for year-round hot water supply **Ratings:** Steam 355-2,460 sq ft; water 570-3,940 sq ft EDR.

IDEAL ARCO ROUND BOILER



An unjacketed, low priced boiler Uses hard or soft coal or coke May be easily converted to automatic firing—coal, oil or gas fuel Machine ground surfaces of doors and sections prevent waste of heat **Ratings:** Steam 200-800 sq ft; water 320-1280 sq ft E D R

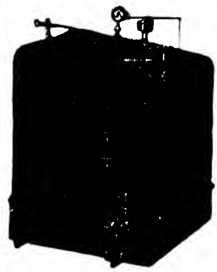
No. 7 IDEAL BOILER FOR COAL (STOKER OR HAND FIRED) OIL OR GAS



Hand Fired

A low-cost universal boiler with many high priced features. Gives quick pick up—low cost heat. Improved flue design utilizes available heat. Precision ground iron-to-iron sections. Provision for domestic hot water supply by screwed in Taco Heaters. **Ratings:** Steam 225-750 sq ft; water 360-1,200 sq ft E.D.R.

IDEAL WATER TUBE BOILERS



Ideal Water Tube Sectional Boilers for homes as well as large buildings and commercial installations Sectional construction permits easy admittance through doorways and inexpensive installation. Low water line eliminates digging of pits For all fuels **Ratings:** Steam 450-15,000 sq ft, water 720-24,000 sq ft E D.R

IDEAL REDFLASH BOILERS

A popular boiler for any kind or size of building from cottage to skyscraper Red enamel jacket. Multi-ply Asbestocel insulation Automatic damper regulator on all steam boilers and on smallest size hot water boiler **Ratings:** Steam 220-10,370 sq ft; water 350-16,600 sq ft E D R

AUTOMATIC COAL-FIRED BOILERS

Designed especially for use with mechanical stokers, these boilers are recommended by leading stoker manufacturers. Efficient and economical operation. External or built-in hot water supply. A complete range of sizes and models. **Ratings:** Steam 375-8,445 sq ft; water 600-13,512 sq ft E.D.R.

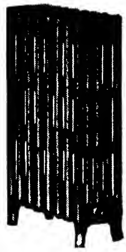
Write us for detailed information

PRODUCTS FOR EVERY HEATING REQUIREMENT

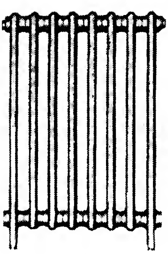
CAST IRON RADIATORS

American Radiator manufactures a complete line of cast iron radiators suitable for all types of heating. Complete ratings, dimensions and data will be sent on request.

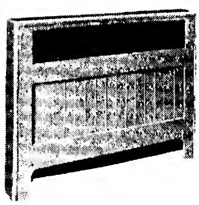
Arco Convector



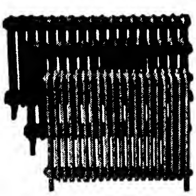
Corto Radiator



Corto Hospital Type

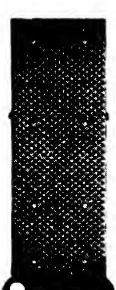


Arco Radiant Convector

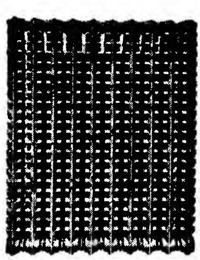


Arco Radiators

Peerless Wall Radiator



Perfection Pin



Vento Cast Iron Radiator

WATER HEATERS

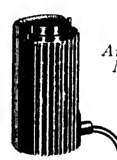
Water heaters of all types are quickly available from branch warehouse stocks and wholesalers. Capacities range from 3000 gallons with the Arco High Test Tank Heater down to 65 gallons with the Dome Type Heater shown on the right. A partial list of these heaters is given and complete information on them will be sent on request.

Arco High Test Tank Heater
Kolflash Water Heater
Ideal Oil Burning Water Heater
Old Line Water Heater
Ideal Dome Type Water Heater
Scuttle-a-Day Water Heater
Excelso Water Heater

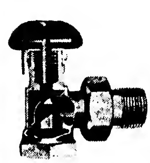
ARCO ACCESSORIES

Arco Accessories permit the installation of completely integrated heating systems, backed by a single responsibility. Illustrated below are a few of the Complete Line of valves, vents, controls for steam, vapor or water. With new products being constantly made available and with changes and improvements being made to existing products, we suggest you write us for further information on all our equipment. A complete list with ratings will be sent on request.

No. 2001 Equatrol



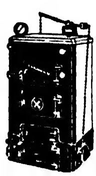
No. 300 Arco Detroit Multiport



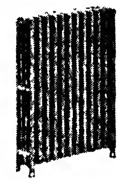
Arco Packless Valve

No. 861 Arco Detroit Hurivent

(See also American Radiator Co. pages 898-899 and Subsidiaries)



Burnham Boiler Corporation



Irvington-on-Hudson, New York

Zanesville, Ohio

There's a Burnham for Every Purpose

Catalogs Sent on Request

1—Welded Steel Boilers—Also Three Purpose Welded Steel Boilers.
For heating, hot water supply and incineration. Coal or oil. Completely welded for 15 lb working pressure. Multiple shaking grates. Sizes for commercial or domestic uses. Special folder sent on request.

2—Water Tube Boilers for Steam and Hot Water Heating.
17, 21, 27, and 36 in. Double shaking grates, long fire travel. Steam rating to 9,050 sq ft; for water, 14,480 sq ft.

3—Water Tube Boilers Jacketed in Color.
17, 21, and 27 in. Steel Jacket and 4 ply air cell asbestos insulation. Enameled rich red and black. Jacket goes on after all other set up work. Rating to 5,000 sq ft for steam and 8,000 sq ft for water.

4—Burnham Oil-Burning Boilers.
A specific-sized boiler for each specific heat job for use with any standard oil burner. Round Sectional Burnhams in 6 series and 24 sizes. Square Burnhams in 5 series and 39 sizes. For steam, vapor, or water.

5—Big Twin Sectional Boilers.
50 in. Grate divided for easy shaking. Twin sections, divided down the middle. Ratings to 29,200 sq ft for steam, 46,720 sq ft for water.

6—Tube Type Smokeless Boilers.
Burn soft coal efficiently, without smoke. Meet smoke ordinances everywhere. Similar to (2) above with addition of smokeless feature.

7—Round Sectional Boilers.
This boiler made the long fire travel famous. Handled easily. Very large steam dome. Ratings up to 2,080 sq ft for steam 3,440 for water.

8—High Pressure Hot Water Supply Boilers.
Sectional construction Guaranteed to 120 lb working pressure. Supplies up to 14,000 gal.

9—Dome Top Hot Water Supply Boilers.
Will keep 50 to 1500 gal tank always full of hot water. Guaranteed to 120 lb working pressure.

10—Burnham-Taco Tanks.
Combining water heater and storage tank in one unit for summer-winter use. Removable copper heating element. Tanks may be galvanized, Everdur or copper.

11—Burnham Slenderized Radiators.
Cast iron radiators that occupy 40 per cent less space than ordinary type of same rating Shorter. Lower. Narrower. 3-tube type $3\frac{1}{4}$ in. wide. 4-tube type $4\frac{7}{16}$ in. wide 5-tube type $5\frac{11}{16}$ in. wide. 6-tube type $6\frac{15}{16}$ in. wide. Can be recessed.

12—Fero Tube Radiators.
All heights—3, 4, 5, 6, and 7 tubes.

13—Burnham Air Conditioning Units.
Do double duty of both heating and winter air conditioning. Units placed in the room. Have no basement equipment. Take up no more room space than usual grille-enclosed radiator. Entirely automatically controlled.

14—Burnham Air and Vacuum Valves.
Full line for radiators, risers and mains.

15—Burnham Radiator Valves.
Complete line of heating accessories. Including steel tanks of all kinds.

16—Burnham Flexible Headers.

17—Burnham Unit Heaters.
Complete line in modern designs.

18—Burnham Fan Cooler.
Portable unit for offices, hotels, restaurants and home rooms. Also complete equipment package for attic cooling of residences.

Burnham Boiler Corporation

Manufacturers of Cast Iron and Welded Steel Boilers, Cast Iron Radiators and Heating Accessories

Irvington-on-Hudson, N. Y.

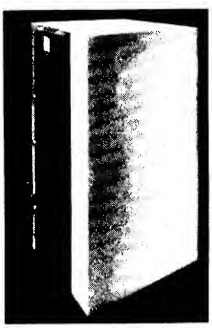
Zanesville, Ohio

Branch Offices

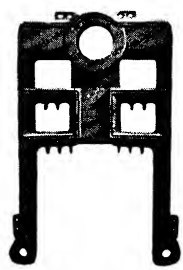
BOSTON; PHILADELPHIA; CHICAGO; QUEENS VILLAGE, L. I.; LONG ISLAND CITY, N. Y.; BALTIMORE; SPRINGFIELD; LANCASTER; PITTSBURGH; ZANESVILLE; ELIZABETH; GENEVA, N. Y.

Plants

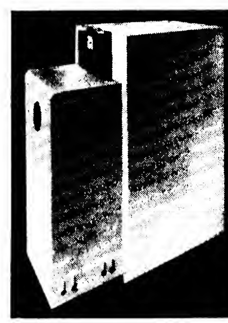
ELIZABETH, N. J.; LANCASTER, PA. ZANESVILLE, OHIO; GENEVA, N. Y.



As jacketed for burning hand fired coal or gas.



Section, showning central extension, of combustion chamber to top of boiler.



As jacketed for oil burning. Jacket extension easily removable. Its use is optional.

BURNHAM ALL-FUEL YELLO-JACKET BOILER

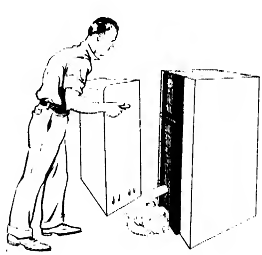
That Stings The Fuel Bill

Its outstanding economy point is the expanded combustion chamber, that extends from the bottom to the top of the boiler, while still having the Burnham basic, back and forth fire travel. The fire shine, or direct surfaces, are greatly increased. Further efficiency is secured by the adding of fins.

To prevent a too rapid rush of fire travel gases, there is a balanced down draft, that overcomes the cooling off losses, or heat lag, incident to the intermittance with automatic firing of coal, oil or gas.

The main combustion chamber has the advantage of additional height, an essential, especially with oil and stoker fired coal.

It is equipped with a fully bilt-in Taco hot water supply heater. You have the option of the Tankless or Storage Tank type. The jacket is a light dandelion yellow, combined with black and a touch of chromium. It has an optional removable extension for completely enclosing an oil burner. Can be easily converted from oil to coal, or the reverse.



Showing how jacket extension can be easily lifted off. Handles provided for the purpose.

Made in 7 sizes.

For steam: from 325 to 775 square feet.
For water: from 520 to 1240 square feet.

Crane Co.

BOILERS, RADIATORS, VALVES, FITTINGS, PIPE, STEAM SPECIALTIES, PLUMBING AND HEATING MATERIALS

General Offices: 836 South Michigan Avenue, **Chicago, Illinois**

Nation-wide Service Through Branches, Wholesalers, Plumbing and Heating Contractors

CRANE HEATING EQUIPMENT FOR RESIDENTIAL AND COMMERCIAL BUILDINGS

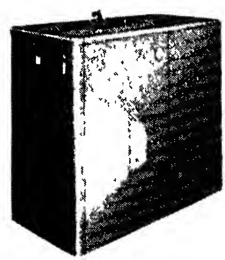
Crane offers a single source of supply for a full line of heating equipment, automatic or manually operated, for both residential and commercial buildings.

The advantages of a single responsibility for the whole system are obvious, and the reputation of the name Crane is your guarantee of quality and satisfaction.

CRANE OIL AND GAS-FIRED BOILERS

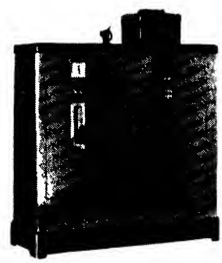
CRANE SUSTAINED HEAT OIL-BURNING BOILERS

Combination boiler and burner unit with patented sustained heat principle which extracts more heat from the fuel; and down-draft construction which prevents escape of gases into flue before their heat has been absorbed.



CRANE FIN-TYPE OIL-BURNING BOILERS

Specially designed for oil-burning with any gun-type burner. Many integrally cast fins increase absorption of heat. Made with plain insulated jacket, or De Luxe jacket completely enclosing boiler and burner



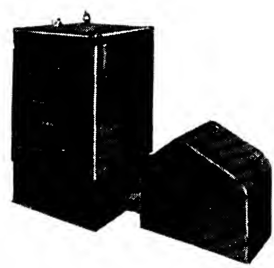
BASMOR GAS-FIRED BOILERS

A wide variety of sizes available for every home. Its specially designed "Butterfly" Bunsen-type gas-saving burners assure proper combustion Immune to backfiring In Standard and De Luxe Models with insulated jackets.

CRANE COAL-FIRED BOILERS

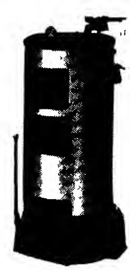
CRANE SECTIONAL COAL BOILERS

Crane Sectional Boilers are made in sizes to meet a wide range of load demands. Patented control of water circulation contributes to faster heating and higher efficiency. Available with or without insulated jacket.



CRANE STOKER-FIRED BOILERS

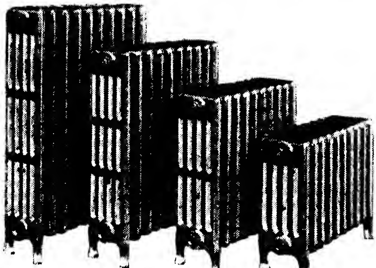
Identical to the Crane Sectional (manually fired) Boiler with grates and shaking mechanism eliminated. Available with special high base to obviate need of placing stoker in pit or raising boiler.



CRANE ROUND COAL BOILERS

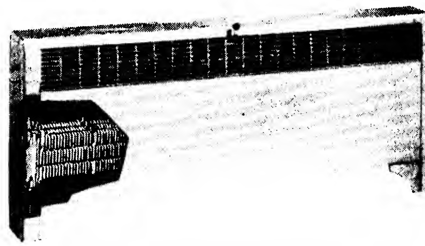
A round boiler with exceptionally good flue travel and generous depth in fire pot Available in range of sizes sufficient for small to medium size home. Furnished with or without insulated jacket.

CRANE RADIATORS AND CONVECTORS



CRANE CAST-IRON RADIATORS

Crane Radiators are available in a complete range of sizes to meet every structural need. Screw nipple construction enhances appearance and dependability. Optional equipment is Crane patented invisible shields which direct heat to "living zone."



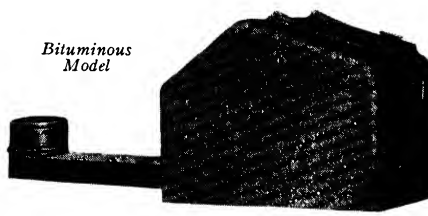
CRANE CONVECTOR RADIATORS

Enclosures are of heavy steel, with removable grille sections of plastic or pressed steel. For completely or partially recessed free-standing, wall-hung, or plaster front installation. Crane advanced design and assembly method assure lasting efficiency.

CRANE OIL BURNERS AND STOKERS

CRANE CONSERVOIL BURNERS

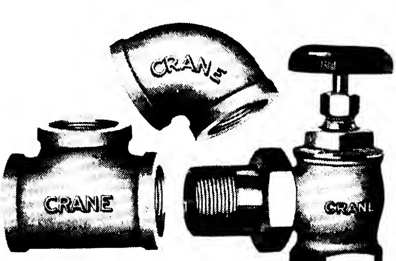
A high efficiency burner with controlled air delivery, assuring proper mixture of air and oil—reducing combustion noises. Silent operation achieved through a simple yet efficient fan and two-stage fuel pumps operating on a single shaft—making only one moving part.



CRANE AUTOCOAL STOKERS

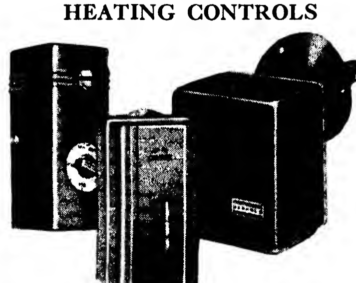
A worm-feed stoker, with five speeds. Every friction point is in transmission case and runs in oil bath. Burner is full coking type with sectional tuyeres of chromium alloy iron. Has thermal-overload protector switch. Worm tube is seamless, smooth. Also available for anthracite coal.

VALVES AND FITTINGS



Crane makes every type of valve, fitting, and steam specialty for heating installations. Backed by Crane experience of more than 80 years in engineering, Crane piping equipment in any installation assures greater efficiency—tighter and more workmanlike assembly.

HEATING CONTROLS



The Crane line of automatic controls includes room thermostats, primary oil burner and stoker controls, fan or circulator relays, water temperature and steam pressure limit controls and regulators, furnace temperature controls of simplified design —precision-built throughout.

Spencer Heater Division

Lycoming Manufacturing Company

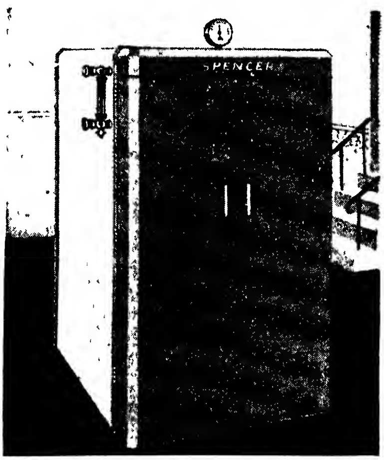
Williamsport, Pa.

Sales Representatives in Principal Cities

Spencer Automatic Magazine Feed Heaters are furnished in cast iron sectional types —and steel tubular types for larger buildings—for steam, vapor and hot water heating. There is a size and capacity for every type of building, to provide economical and convenient heat—safe, dependable, sure

The Spencer Automatic Magazine Feed Warm Air Furnace makes available the advantages of the Spencer principle of magazine feed, sloping grates, automatic coal action, and low operating cost - for warm air systems of either gravity or forced circulation type

COMFORTABLE HEAT AT LOW COST



Spencer Jacketed Heater L-1 Series

Why Spencer Heaters and Furnaces perform so satisfactorily can best be explained by an inspection of their design and construction. The Spencer principle, illustrated in the cross-sectional view, is simple

Once a day fuel (No. 1 Buckwheat Anthracite or small size by-product coke) is put into the magazine It fills the sloping grate to the level of the magazine mouth The fire bed always stays at the proper level, for as fast as fuel burns to ash, it shrinks and settles on the sloping grate; and more fuel rolls down automatically over the top of the fire bed. Fuel feed is by gravity alone, in just the right amount to keep the fire always burning at its most efficient combustion point

This explains why a Spencer Automatic Magazine Feed Heater or Furnace always gives the same uniform, satisfying heat, and burns less fuel. These exclusive Spencer advantages are available in all types of the magazine feed heaters and furnaces.

Coal — Coke — Gas — Oil — Spencer J and L series heaters, M series boilers and W series furnaces are primarily designed to burn low cost No 1 Buckwheat Anthracite or small size coke, the CN series Nut or Pea Anthracite, or Nut size coke

If at any time a property owner desires to burn more expensive fuels—oil or gas—his Spencer Heater will show a higher efficiency than can be secured with any ordinary boiler.

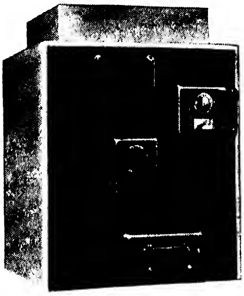
Thermostats—Thermostats and electric damper motors are furnished as optional equipment.

Jacketed Covering—Attractive metallic jackets, as illustrated, are available for Spencer Cast Iron Heaters, either with or without the enclosing jacket doors

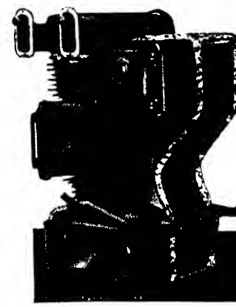
Spencer Heavy Duty Tank Heaters—Of the magazine feed type, they are available in the capacities indicated With the automatic magazine feed construction, they provide ample domestic hot water at lowest cost, and with a minimum of tank heater attention

Cutaway sectional view Spencer Cast Iron Heater

SPENCER AUTOMATIC MAGAZINE FEED WARM AIR FURNACES



DeLuxe Jacket



Unjacketed

Spencer Furnaces operate on the automatic magazine feed principle. Once fuel (No. 1 Buckwheat Anthracite or small size coke) is in the storage magazine, it feeds over the fire on the sloping grate in just the right amount for the heat required. Attention is necessary only once or twice a day.

Unique in furnace design, the Spencer burns low cost No. 1 Buckwheat Anthracite or small size coke, and eliminates the disadvantages of ordinary furnaces —no "hot spots," no furnace cement, no brick lined fire pot, no leaky joints. Assures an air-tight, gastight, leakproof, long life installation.

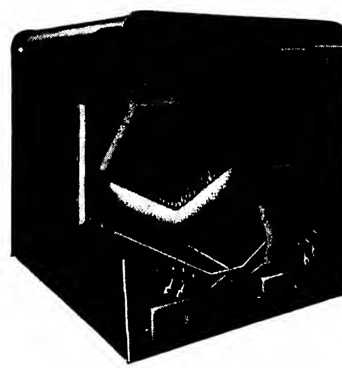
Spencer Furnaces are compact, streamlined, and so designed that other air-conditioning features can be readily added.

Illustrated here is the attractive enameled metallic jacket. Galvanized casing also available, if desired.

SPENCER STEEL TUBULAR MAGAZINE FEED BOILERS

For large buildings we recommend Spencer Steel Tubular Magazine Feed Boilers, burning low cost No. 1 Buckwheat Anthracite or coke.

In the cross-section diagram, part of the fire bed is cut away to show the sloping grates and the two magazines filled with fresh coal, ready to feed down automatically of its own weight to the fire. These boilers are built in two vertical sections for ease in handling and installation—a great advantage on replacement jobs, eliminating the necessity of costly tearing out of wall or partitions. Combination water and fire tube construction; built to *A.S.M.E.* standards.



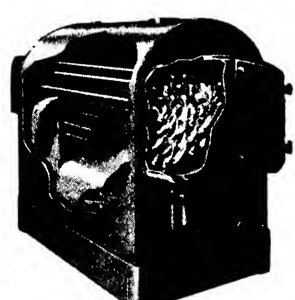
Steel Tubular Magazine Feed Boiler

SPENCER STEEL BOILERS For Oil, Stoker, Gas or Hand-Firing

For more than 40 years, Spencer has been building, in the opinion of experts, one of the most efficient, economical and dependable coal burning boilers on the market. With this background of experience, Spencer Engineers developed the Spencer Steel Tubular Boiler for oil, gas, stoker and hand-firing—the "C" series for residential use, and the Type "A" for larger buildings. It is a better boiler both for the property owner and for the architect or engineer who specifies it.

The high sustained efficiency of these boilers means adequate heat for a lower fuel cost. Design is of the three pass type. Combustion chamber is amply large. Built of best quality open hearth steel boiler plate, and tubes. Can be furnished with domestic hot water heating coils, either of the storage tank or instantaneous type.

"C" Series Steel Boiler

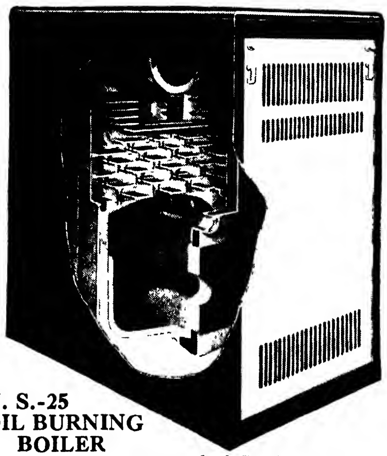


Type "A" Steel Boiler

UNITED STATES RADIATOR CORPORATION

General Offices: Detroit, Michigan • Branches and Sales Offices in Principal Cities

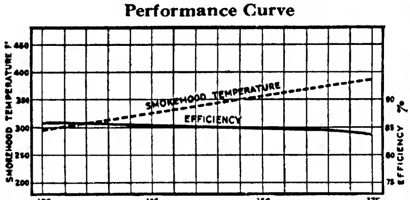
Detroit, Michigan



U. S.-25 OIL BURNING BOILER

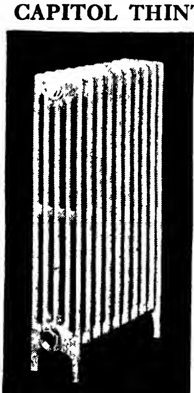
Front Extended Jacket

Performance Curve



Output—% of Direct Standing Radiator Load Flue Gas Analysis CO^2—12 5%, O^2—4 1%; CO—0 0%

CAPITOL THINTUBE RADIATORS



3-Tube

Heights	Per Section Heating Surface
19″	1 1 Sq Ft
22″	1 3 Sq Ft
25″	1 5 Sq Ft

4-Tube

19″	1 4 Sq Ft
22″	1 6 Sq Ft
25″	1 8 Sq Ft

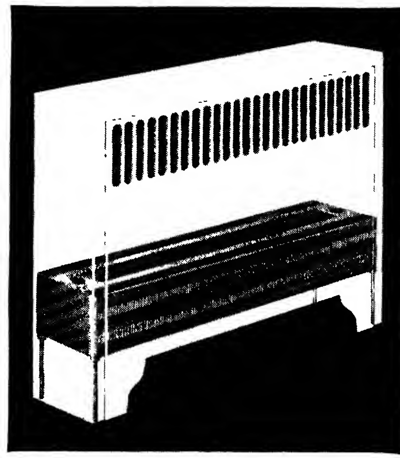
5-Tube

20″	1 8 Sq Ft
23″	2 1 Sq Ft
26″	2 4 Sq Ft

6-Tube

19″	2 2 Sq Ft
32″	3 85 Sq Ft

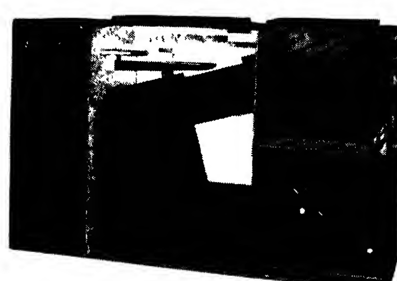
40 per cent less space needed for these graceful, efficient Capitol ThinTube Radiators.



CAPITOL FINCAST ENCLOSURES AND CONVECTORS

Made entirely of cast iron.
Without joints—Cast in one piece.
Many lengths and widths.
Tappings, top, bottom and ends.
Complete choice of enclosures.
Finished to harmonize with modern interiors.

CAPITOLAIRE DIRECT FIRED



CONDITIONING UNIT

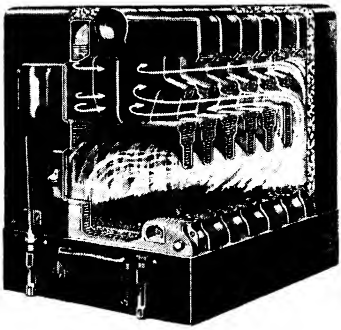
An air conditioning unit especially designed for oil firing Streamline flue construction in preheating and prime-heating sections insures high efficiencies. Deluxe jacket completely encloses all controls, heating unit, automatic humidifier, filter and fan.

Other gravity and forced air furnaces available for gas, stoker or hand coal firing.

UNITED STATES RADIATOR CORPORATION

General Offices: Detroit, Michigan • Branches and Sales Offices in Principal Cities

Detroit, Michigan



CAPITOL SQUARE BOILERS FOR ALL FUELS

"A" Series

Ratings in Sq Ft	Direct Cast Iron Radiator Loads—Sq Ft
Steam— 575-1450 Water— 975-2475	280- 655 460-1085

"B" Series

Steam—1200-3600 Water—1980-5940	550-2030 910-3350

"C" Series

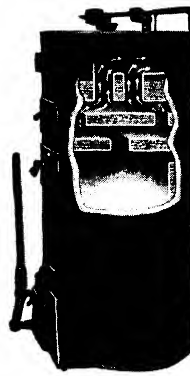
Steam—4700-10,500 Water—7760-17,325	1865-5805 3080-9580

Illustrated above is a Capitol Red Top Series "C" Boiler. These sectional, cast iron, all fuel boilers can be furnished either jacketed or unjacketed. Sections are connected with precision-made slip nipples in accurately machined ports and the individual sections are ground to permit an iron-to-iron gas-tight assembly. The large, smooth flueways are designed without sharp turns reducing friction, permitting unrestricted gas travel. For easy cleaning, all flueways are fully accessible through the large front clean-out doors. Controlled internal water circulation and large nipple ports insure complete separation of water and steam. Capitol Red Top Boilers are heavily insulated against heat loss by a thick blanket of rock wool, reinforced with wire mesh to prevent slipping or sagging. Capitol Red Top Boilers can be furnished with extra high steel bases to provide extra setting height or desired additional furnace volume for stoker firing. These bases eliminate the necessity for pitting or building a brick or concrete base.

CAPITOL RED CAP BOILERS

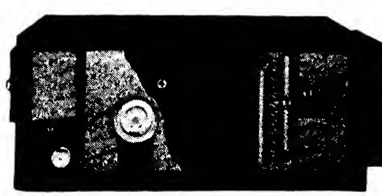
For All Fuels

Boiler No.	Direct Cast Iron Radiator Loads Sq Ft	
	Steam	Water
19-4	300	495
19-5	350	580
20-4	400	660
20-5	450	745
22-4	500	825
22-5	550	910
25-4	625	1030
25-5	675	1115



Capitol Red Cap Boiler design brings the advantages of (1) long fire travel, (2) flue passages that force the hot gases to circulate through every section, extracting the maximum heat from the fuel consumed. (3) Deep firepot that provides the extra space needed for better combustion, and smooth, tapered firepot walls to assure a clean surface for better heat absorption.

CAPITOLAIRE CONDITIONING UNIT FOR SPLIT SYSTEMS



A suspended unit, quiet in operation, using ducts to convey heated, humidified and filtered air to various rooms of the home. Heating supplied to the Unit heat exchange coil by a steam or hot water boiler. Floor model conditioning units with a wide range of heating capacities also available.

SPECIFICATIONS

Unit No.	Capacity—BTU/HR				Air Capacity cfm
	Heating*		Cooling**		
	200° H. W.	2# Steam	50° Water	40° Refrig.	
S-1	34,000	28,000	14,000	15,000	500
S-2	48,000	39,000	19,000	21,000	750
S-3	65,000	54,000	26,000	28,000	1000
S-4	99,000	128,000	37,000	39,000	1500

Weil-McLain Company

Manufacturing Division: Michigan City, Ind. and Erie, Pa.

General Offices: **641 W. Lake Street, Chicago**

NEW YORK OFFICES **501 Fifth Avenue**

Prompt Weil-McLain Boiler and Radiator service is made conveniently available through local stocks carried by Weil-McLain Distributors in most of the important distributing centers

New All-Fuel Boilers No. 67 and No. 77

Conversion type boilers with insulated enameled jacket. For hand or automatic firing. Connected Load Ratings: Steam 300 to 950 sq ft, Water 480 to 1,520 sq ft.

No. 78 Boiler for Automatic Firing

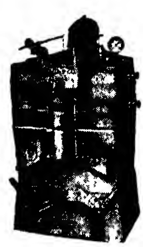
Boiler has insulated enameled de luxe jacket. Front or rear jacket extension available. Connected Load Ratings: Steam 400 to 1,030 sq ft, Water 640 to 1,650 sq ft.

"RO Series" Boiler for Automatic Firing

Jacketed and insulated round boiler for small homes. Connected Load Ratings: Steam 420 and 520 sq ft, Water 630 and 790 sq ft.

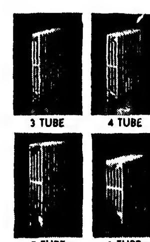
Square-Type Boilers

Sectional boilers for larger installations. Complete range of sizes. Connected Load Ratings: Steam 650 to 9,300 sq ft, Water 1,050 to 14,900 sq ft.



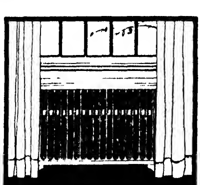
Self-Feed Boiler

Magazine type boiler for small inexpensive sizes hard coal or coke. Connected Load Ratings: Steam 240 to 785 sq ft, Water 385 to 1,260 sq ft.

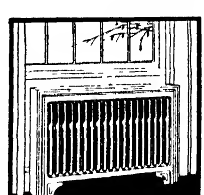


Junior Radiators

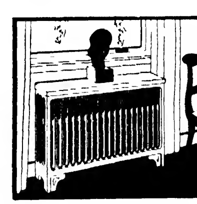
Occupy 40 per cent less space than conventional radiators of same rating. 3-tube, $3\frac{3}{8}$ in. wide; 4-tube, $4\frac{9}{16}$ in. wide; 5-tube, $5\frac{3}{4}$ in. wide; 6-tube, $6\frac{15}{16}$ in. wide.



Concealed Raydiant



Partially Recessed



Cabinet Raydiant

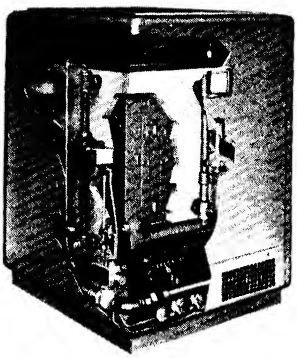
Raydiant is a convector type all cast iron radiator made in Concealed, Partially Exposed and Cabinet Types. Raydiant Radiators, however, differ from conventional convectors in that they supply not only convected heat but also sun-like radiant warmth from their heat radiating "live" panel front. A second important advantage is their ability to hold heat longer. This helps increase the comfort of (on and off) automatic heating and makes mixed installation of Raydiant and standard radiation practicable.

American Gas Products

Division of American Radiator Company

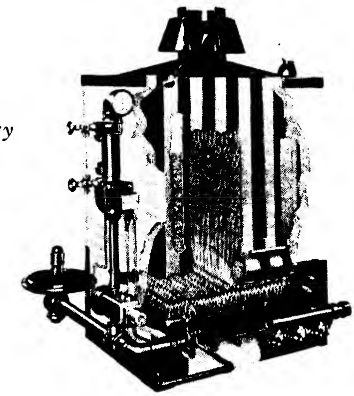
40 West 40th Street, **New York, N. Y.**

Gas Boilers for Hot Water . . . Steam and Vapor Heating . . . Automatic Hot Water Storage Heaters . . . Air Conditioners . . . Gas Fired Steam Radiators



Empire Ideal

Approved by A. G. A. Laboratory for Steam, Vapor or Hot Water Heating Systems. Ratings Below.



Standard Ideal

Specifications for Empire and Standard Ideal Gas Boilers

STEAM BOILERS					WATER BOILERS					
Steam Boiler Number	A.G.A. Steam Rating Sq. Ft.	Supplies Sq. Ft. Direct C.I. Radiation	No. and Size of Tappings		Water Boiler Number	A.G.A. Water Rating Sq. Ft.	Supplies Installed		No. and Size of Tappings	
			Supply In.	Return In.			Gravity	Accelerated	Supply In.	Return In.
0-GS-4	270	171	2-2½	1-1½	1-GA-4	210	126	131	2-1½	2-1½
0-GS-5	360	231	2-2½	1-1½	1-GA-5	280	167	175	2-1½	2-1½
0-GS-6	450	291	2-2½	1-1½	1-GA-6	350	210	220	2-1½	2-1½
0-GS-7	540	357	2-2½	1-1½	1-GA-7	420	253	265	2-1½	2-1½
0-GS-4-E	255	161	1-3	1-1½	2-GA-4	420	253	265	1-2½	1-2½
0-GS-5-E	340	217	1-3	1-1½	2-GA-5	560	342	359	1-2½	1-2½
0-GS-6-E	425	275	1-3	1-1½	2-GA-6	700	434	455	1-2½	1-2½
0-GS-7-E	510	336	2-3	1-1½	2-GA-7	840	527	554	1-2½	1-2½
0-GS-9-E	680	460	2-3	1-1½	4-GA-9	1120	724	761	2-2½	2-2½
0-GS-11-E	850	592	2-3	1-1½	4-GA-11	1400	931	980	2-2½	2-2½
1-GS-4	610	408	2-4	2-4	4-GA-13	1680	1149	1215	2-2½	2-2½
1-GS-5	775	533	2-4	2-4	1-GW-4	980	624	656	2-4	2-4
1-GS-6	940	666	2-4	2-4	1-GW-5	1240	810	853	2-4	2-4
1-GS-7	1105	792	2-4	2-4	1-GW-6	1500	1007	1063	2-4	2-4
1-GS-8	1270	934	2-4	2-4	1-GW-7	1770	1222	1294	2-4	2-4
1-GS-9	1435	1055	2-4	2-4	1-GW-8	2030	1410	1493	2-4	2-4
1-GS-10	1600	1176	2-4	2-4	1-GW-9	2300	1597	1691	2-4	2-4
1-GS-11	1765	1298	2-4	2-4	1-GW-10	2560	1778	1882	2-4	2-4
4-GS-6	2000	1471	2-6	2-5	1-GW-11	2820	1958	2074	2-4	2-4
4-GS-7	2400	1765	2-6	2-5	4-GW-6	3200	2222	2353	2-6	2-5
4-GS-8	2800	2059	2-6	2-5	4-GW-7	3840	2667	2824	2-6	2-5
4-GS-9	3200	2353	2-6	2-5	4-GW-8	4480	3111	3294	2-6	2-5
4-GS-10	3600	2647	2-6	2-5	4-GW-9	5120	3555	3765	2-6	2-5
4-GS-11	4000	2941	2-6	2-5	4-GW-10	5760	4000	4235	2-6	2-5
4-GS-13	4800	3529	4-6	3-5	4-GW-11	6400	4444	4706	2-6	2-5
4-GS-15	5600	4118	4-6	3-5	4-GW-13	7680	5333	5647	4-6	3-5
4-GS-17	6400	4706	4-6	3-5	4-GW-15	8960	6222	6588	4-6	3-5
4-GS-19	7200	5294	4-6	3-5	4-GW-17	10240	7111	7529	4-6	3-5
4-GS-21	8000	5882	4-6	3-5	4-GW-19	11520	8000	8471	4-6	3-5
4-GS-22	8400	6176	6-6	4-5	4-GW-21	12800	8889	9412	4-6	3-5
4-GS-25	9600	7059	6-6	4-5	4-GW-22	13440	9333	9882	6-6	4-5
4-GS-28	10800	7941	6-6	4-5	4-GW-25	15360	10666	11294	6-6	4-5
4-GS-31	12000	8824	6-6	4-5	4-GW-28	17280	12000	12706	6-6	4-5
4-GS-33	12800	9412	8-6	5-5	4-GW-31	19200	13333	14118	6-6	4-5
4-GS-37	14400	10588	8-6	5-5	4-GW-33	20480	14222	15059	8-6	5-5
4-GS-41	16000	11765	8-6	5-5	4-GW-37	23040	16000	16941	8-6	5-5
					4-GW-41	25600	17778	18824	8-6	5-5

All Boilers except Type 4-G and Nos. 1-GS or W-11 are available in either Standard or Empire Ideal models.

(*See also Page 900*)

THE BABCOCK & WILCOX COMPANY

85 Liberty Street **New York, N. Y.**

Manufacturers of

Water-Tube Boilers **Chain-Grate Stokers**

Oil Burners **Seamless Steel Tubing and Pipe**

Branch Offices and Representatives in all Principal Cities

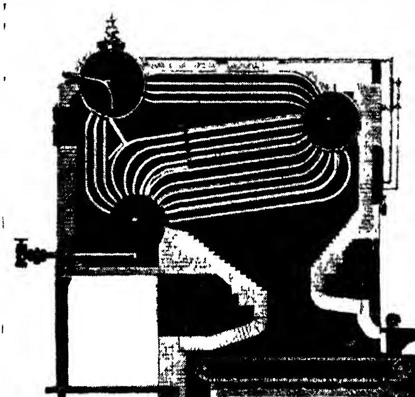
Type H Stirling Boiler

The Babcock & Wilcox Type H Stirling Boiler is a highly efficient unit built for moderate pressures at moderate prices.... and is designed to occupy minimum floor space and head room for the heating surface required.

This boiler is built in four classes and 36 sizes ranging from 691 to 6225 sq ft of heating surface, and can be designed for operation with any fuel and every method of firing.

The moderate price is due only to the simplicity of design, efficient production methods and superior shop equipment.

Class	Heating Surface	Depth of Setting, Ft, In.	Width of Setting: Single Boiler, Ft, In	Width of Setting: Two Boilers in Battery Ft, In	Floor to Center of Mud Drum, Ft, In.	Floor to Face of Steam Outlet, Ft, In	Floor to Top of Boiler, Ft, In	Size of Steam Outlet, In
	691	15-2	6-0	11-0	5-2	14-5½	13-3⅛	5
	921	"	7-0	13-0	"	"	"	"
	1152	"	8-0	15-0	"	"	"	"
H-1	1382	"	9-0	17-0	"	"	"	"
	1612	"	10-0	19-0	"	"	"	"
	1843	"	11-0	21-0	"	"	"	"
	2073	"	12-0	23-0	"	"	"	"
	2304	"	13-0	25-0	"	"	"	"
	877	17-8	6-0	11-0	4-9¾	14-5½	13-3⅛	5
	1169	"	7-0	13-0	"	"	"	"
	1462	"	8-0	15-0	"	"	"	"
H-2	1754	"	9-0	17-0	"	"	"	"
	2046	"	10-0	19-0	"	"	"	"
	2339	"	11-0	21-0	"	"	"	"
	2631	"	12-0	23-0	"	"	"	"
	2924	"	13-0	25-0	"	"	"	"
	1063	20-2	6-0	11-0	4-5½	14-5½	13-3⅛	5
	1417	"	7-0	13-0	"	"	"	"
	1772	"	8-0	15-0	"	"	"	"
	2126	"	9-0	17-0	"	"	"	"
H-3	2480	"	10-0	19-0	"	"	"	"
	2835	"	11-0	21-0	"	"	"	"
	3189	"	12-0	23-0	"	"	"	"
	3544	"	13-0	25-0	"	"	"	"
	3898	"	14-0	27-0	"	"	"	6
	4252	"	15-0	29-0	"	"	"	6
	1245	22-8	6-0	11-0	4-1¼	14-11⅜	13-3⅛	5
	1660	"	7-0	13-0	"	"	"	"
	2075	"	8-0	15-0	"	"	"	"
	2490	"	9-0	17-0	"	"	"	"
H-4	2905	"	10-0	19-0	"	"	"	"
	3320	"	11-0	21-0	"	"	"	"
	3735	"	12-0	23-0	"	"	"	"
	4150	"	13-0	25-0	"	"	"	"
	4565	"	14-0	27-0	"	"	"	6
	4980	"	15-0	29-0	"	"	"	6



Type H Stirling Boiler with Babcock & Wilcox Chain-Grate Stoker

The advantages of the Babcock & Wilcox Type H Stirling Boiler may be summarized as follows:

Unusual steaming capacity for the floor space and head-room required.

Boilers may be set singly or in battery.

Setting heights can be varied to suit any condition of firing.

The choice of three locations for gas exit reduces cost of flues and breeching.

Distribution baffles make effective all of the heating surface.

Tube renewal is facilitated by correct tube spacing, and a tube removal door.

Soot blowers can be readily installed to simplify thorough cleaning of all tubes

A superheater can be furnished without any change in the standard design or construction.

The boiler is supported by a structural-steel framework entirely independent of the brickwork.

Ample provision is made for free movement of parts due to expansion and contraction.

A complete table of sizes and dimensions, together with pertinent installation data, is contained in a new bulletin which will be sent upon request. Simply ask for Bulletin G-8-C.

Farrar & Trefts

Incorporated

Buffalo, N. Y.

HEATING AND POWER BOILERS
Bison Compact Boilers
Bisonette Compact Boilers
Firebox Return Tubular Boilers
Firebox Locomotive Type Boilers
Scotch Marine Type Boilers
Vertical Boilers
Horizontal Return Tubular Boilers
Bison Two-Pass Return Tubular Boilers

Established 1864

STEEL PLATE CONSTRUCTION
Storage and Pressure Tanks
Receivers, Welded or Riveted
Steel Pipe, Welded or Riveted
Buoys, Welded or Riveted
Condensers and Kettles
Smokestacks and Breechings
Special Work in Stainless Steel, Everdur, Nickel, Aluminum or Monel Metal

The Bison Compact

The **F&T Bison Compact Welded Heating Boiler** is more than just another boiler. It has been designed carefully so as to have a large furnace volume, the proper volume of water, just the right amount of steam liberating surface, the correct volume for steam storage and a balanced circulation. The result is a remarkably steady water line—**A Balanced Boiler.**

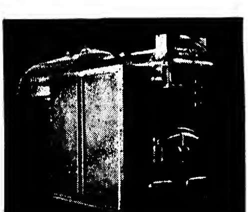
This boiler requires a minimum amount of floor space and is easy and inexpensive to install. It is reasonable as to first cost and economical in operation. Construction is in accordance with the *A.S.M.E.* Code for 15 lb working pressure and boilers are designed for hand firing with anthracite or bituminous coal or for mechanical firing with oil, gas or stoker. There are various sizes available from 1800 to 35,000 sq ft of steam radiation, all ratings as required by the *Steel Heating Boiler Institute.*

The **Bisonette Compact Boiler** has the same characteristics as the larger Bison Compact Boiler. It has been designed for installation in large residences and small business establishments where the advantages inherent in a **Steel** boiler are desired.

Firebox Return Tubular Heating Boilers are **Quality Boilers.** They are constructed to measure up to the high standards set by Heating Engineers and will give unfailing service under all conditions. Being economical to install and operate, they are highly favored by Architects and Engineers for heating Schools, Hospitals, etc.

Firebox Return Tubular Boiler

There are two types of Firebox Boilers, the Up-Draft Type and the Down-Draft Type. Both types are made of welded or riveted construction for heating purposes at 15 lb working pressure and of riveted construction for power purposes at 100, 125 and 150 lb. working pressure in accordance with the *A.S.M.E.* Code. Sizes from 1800 to 35,000 sq ft of steam radiation, as rated by the *Steel Heating Boiler Institute,* are designed for hand firing with coal or for mechanical firing with oil, gas or stoker.



The Bison Two-Pass

The **Bison Two-Pass Return Tubular Refractory Lined Boiler** is favored by many Engineers because it **Eliminates Water Legs, Flat Sides and Staybolts.** This boiler is dependable for long years of continuous and economical operation. It stands up under heavy loads and provides that surplus of power so often needed.

This type of boiler is made of welded construction for 15 lb working pressure and of riveted construction for 100, 125 and 150 lb working pressure in accordance with the *A.S.M.E.* Code. Boilers are designed for ratings from 25 to 250 hp and are inspected, tested, and stamped by a representative of a reliable insurance company before shipment. They are furnished for hand firing with coal or for mechanical firing with oil, gas, or stoker.

Fitzgibbons Boiler Company, Inc.

Established 1886

General Offices: Architects Bldg., 101 Park Avenue

New York, N. Y.

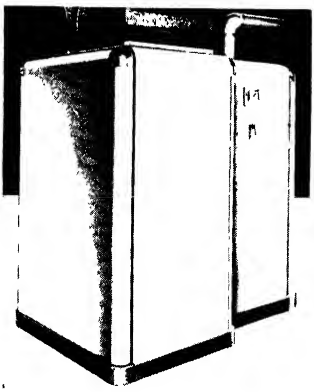
Works· OSWEGO, N Y.

Branches and Representatives in Principal Cities

PRODUCTS—STEEL HEATING and POWER BOILERS for all fuels and all heating systems. Capacities to meet requirements of any building. Built and rated according to *S. H. B. I.* Code. —AIR CONDITIONERS for "Split-Systems" and for Direct-Fired installations in residences of all sizes.

"Split-System" Air Conditioners

The FITZGIBBONSAIRE



combines with Fitzgibbons Steel Boilers for automatic firing with oil, gas or stoker, to provide. (1) CONDITIONED AIR (cleaned, humidified, tempered, circulated) to all rooms where desired; (2) RADIATOR HEAT to kitchen, baths, garage and other parts through which recirculation is undesirable; (3) YEAR-'ROUND DOMESTIC HOT WATER, without a storage tank

The FITZGIBBONSAIRE JR.

Ceiling type unit for combination with oil, gas or stoker fired Fitzgibbons Steel Heating Boiler. Provides moderate cost "Split-System" air conditioning to lower floor, and radiator heat to rest of house, with year-'round tankless domestic hot water

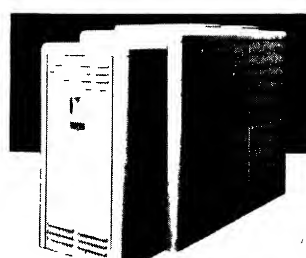
The BOILER-AIRCONDITIONER

The same three services as provided by the FITZGIBBONSAIRE, concentrated in a single compact unit for oil, gas or stoker firing.

Direct-Fired Air Conditioners

The DIRECTAIRE

A new departure in direct-fired conditioners, far ahead of contemporary units in Efficiency, Ruggedness, Durability, Quietness, Cleanability and Appearance. Two models—The ENCLOSING Model (illustrated) which conceals the oil or gas burner; the STANDARD Model for stokers and installations where an enclosing jacket is not required. Each model made in 4 sizes—100,000—135,000—200,000 and 300,000 Btu at the bonnet



Steel Heating Boilers

The **OIL-EIGHTY AUTOMATIC**—the outstanding residential steel boiler for oil firing. Teams up with any good rotary or gun type burner to form a highly efficient unit. Provides room for burner inside the jacket. Year-'round tankless domestic hot water optional. Ratings, Steam—12 Sizes—425 to 2680 sq ft.

The **GAS-EIGHTY**—for gas. Jacketed Ratings, Steam—12 Sizes—425 to 2680 sq ft.

The **STOKER-EIGHTY**—For bituminous firing. Jacketed. Stoker may be installed at either side if desired, to allow free access for inspection through door in front. Ratings, Steam—6 Sizes—485 to 1100 sq ft.

The **COAL-EIGHTY AUTOMATIC**—For anthracite stoker firing Jacketed. Provision for installing stoker at either side as well as front. Supplies year-'round hot water with or without tank as desired. Approved by *Anthracite Institute.* Ratings, Steam—12 Sizes—425 to 2680 sq ft.

FITZGIBBONS R-Z-U JUNIOR

Multi-Service Steel Boiler

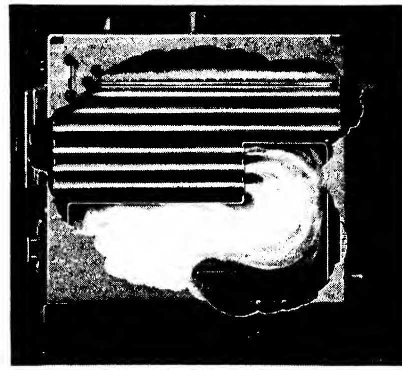
RATINGS, STEAM

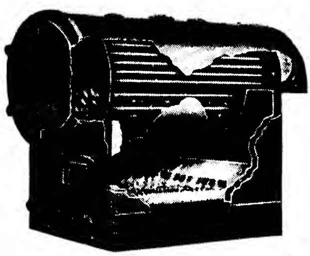
Coal Burning Type........ 900 to 3200 sq ft
Oil Firing Type................1100 to 3900 sq ft
Stoker Firing Type.........1100 to 3900 sq ft

Outstanding Features

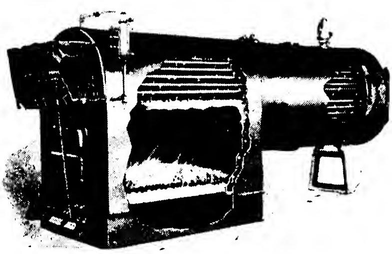
Tanksaver (optional) supplies year-'round hot water without a separate storage tank. **Tankheater** (optional) a more efficient indirect water heater. **Auxiliary Grate** (optional), for refuse disposal and stand-by heating duty in oil fired installations. **Compact,** largest size will pass thru a 31 in. doorway. **Low Water Line,** eliminates need for a pit. **Jacket** (optional), on all types.

Descriptive Bulletin on Request

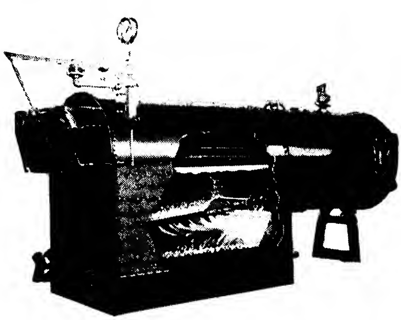




R-Z-U



500 Series



600 and 800 Series

FITZGIBBONS Z-U

Steel Firebox Boilers

Built for 15 lb w.s.p.—*A.S.M.E.* Code.
Up-Draft Type.......1800 to 35,000 sq ft steam

FITZGIBBONS R-Z-U

Steel Firebox Boilers

The Z-U arranged for rear smoke outlet.
Built for 15 lb w.s.p.—*A.S.M.E.* Code.
Up-Draft Type.......1800 to 35,000 sq ft steam
Smokeless Type.......1800 to 35,000 sq ft steam
Oil, Gas, Stoker.....2190 to 42,500 sq ft steam

FITZGIBBONS "F" SERIES

Portable Riveted Firebox Boilers

Built for 100 lb w.s.p.—*A.S.M.E.* Code.
Ratings, steam—1800 to 15,000 sq ft

FITZGIBBONS 500 SERIES

Portable Welded Firebox Boilers—Return Tubular

Built for 15 lb w.s.p.—*A.S.M.E.* Code
Ratings, steam—3500 to 35,000 sq ft

FITZGIBBONS 700 AND "P" SERIES

Portable Riveted Firebox Boilers

700 Series for 15 lb w.s.p.—*A.S.M.E.* Code.
Ratings, steam—3500 to 35,000 sq ft
"P" Series for 100 lb w.s.p.—*A.S.M.E.* Code.
Ratings, horsepower—25 to 250.

FITZGIBBONS 600 AND 800 SERIES

Smokeless Down-Draft Riveted Firebox Boilers

Built for 15 to 100 lb w.s.p.—*A.S.M.E.* Code.
Ratings, steam—3500 to 35,000 sq ft

Descriptive Bulletins on any or all of above boilers will be mailed on request.

E. Keeler Company

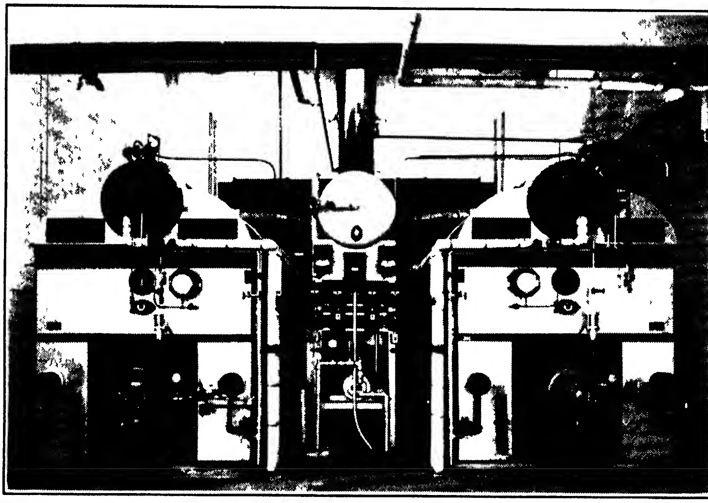
Williamsport, Pa.

75TH ANNIVERSARY

BUILDERS OF BETTER BOILERS SINCE 1864

Steel Boilers For Heating and Power

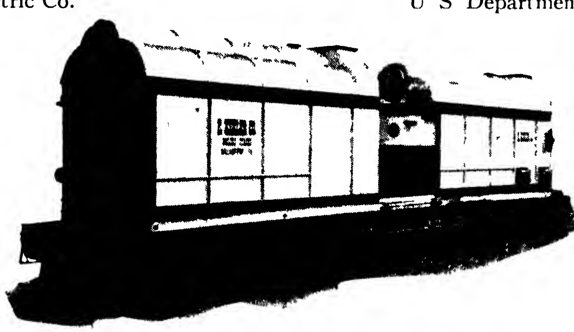
Steel Stacks, Breechings and Plate Fabrications



Two 100 hp Keeler "CP" Water Tube Boilers in Jersey City Plant

A Few Prominent Users of Keeler "CP" Boilers

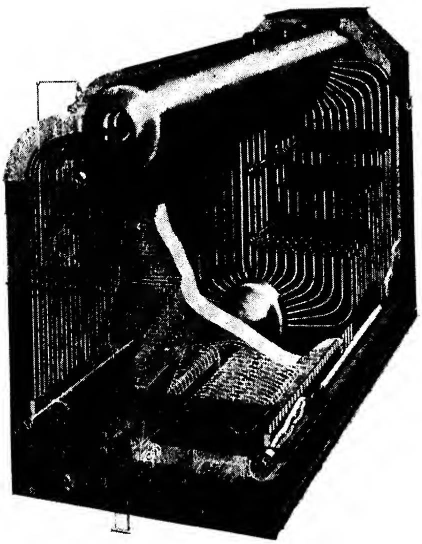
Abbots Dairies
Aluminum Co of America
American Oak Leather Co.
Anaconda Copper Co.
Armstrong Cork Co
Bethlehem Steel Corp.
Dairymen's League
General Electric Co.
General Motors Co
General Chemical Co.
Hahneman Hospital
Hershey Creamery Co.
National Dyeing & Printing Co
Haverford Laundry
Pittsburgh Plate Glass Co.
U S Department of Justice



Two 100 hp Keeler Type "CP" Boilers on One Car.

The Keeler Type "CP" Boiler, while built in standard sizes, can be modified so as to meet all practical requirements—to conform with special conditions of installation or space limitations

Also manufactured in Canada by Canadian Vickers, Ltd , Montreal.



KEELER Type "CP"
STEAM GENERATOR

(Patent No. 2097268)

Keeler Water Tube Boilers are designed to have unrestricted circulation which permits them to respond quickly to sudden steam demands, and to efficiently develop high overload capacity. In the Keeler design no group of tubes or drums is limited to storage of steam, or to the functions of a feed water heater. The entire boiler is a steam generating unit.

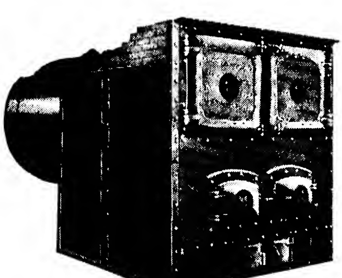
Write for Bulletin F-9

Copies of tests showing efficiencies of 80 per cent and better will be furnished on request.

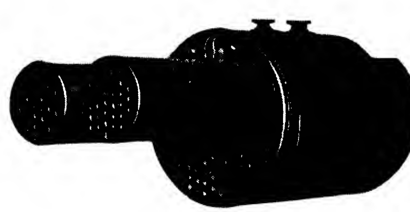
The Keeler Type "CP" Water Tube Boiler Embodies These Important Features

1. It is completely steel encased and insulated.
2. No brickwork required except for front wall and bridge wall.
3. The sides of the furnace are water cooled.
4. Clinkers cannot adhere to sides of furnace.
5. Furnace maintenance costs are reduced to a minimum.
6. Provides very large capacity in a given space.
7. Can be operated at high overloads without disturbing water level.
8. Has built-in soot blower. Easy to clean interior or exterior.
9. Made in large range of sizes in pressures up to 450 lb.
10. Units as large as 250 hp can be shipped completely assembled.
11. Units larger than 250 hp are shipped knocked down.
12. Highly efficient with any method of firing.

The E. Keeler Company manufactures Straight Tube Water Tube Boilers, both long drum and cross drum types, Curved Tube Boilers, both three drum and four drum types, Return Tubular and Double Duty Fire Tube Boilers for every power or heating requirement.



Keeler Double Duty Boiler



Keeler Double Duty Boiler
Before Encased

Bulletins of Any Type Sent on Request

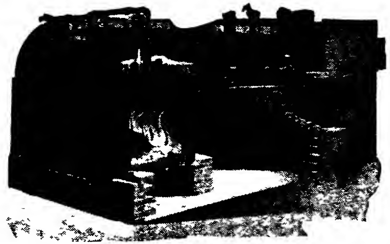
Offices in principal cities

Kewanee Boiler Corporation

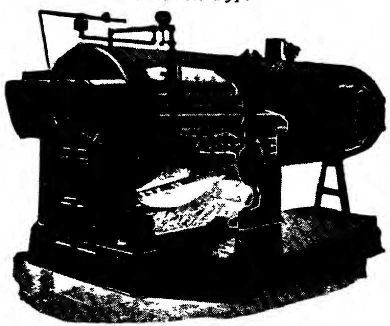
Kewanee, Illinois

BRANCHES IN 64 PRINCIPAL CITIES

Steel Heating and Power Boilers, Water Heating Garbage Burners, Tabasco Heaters and Tanks.



Kewanee Firebox Boilers Brick-set Type



Kewanee Smokeless Boilers Portable Type

KEWANEE STEEL HEATING BOILERS

Kewanee offers a dependable line of Steel Boilers built for heating every size building, with high efficiency, burning any kind of fuel There are 350 standard sizes and 33 types of Kewanee Boilers most of which are kept in stock, ready for immediate delivery.

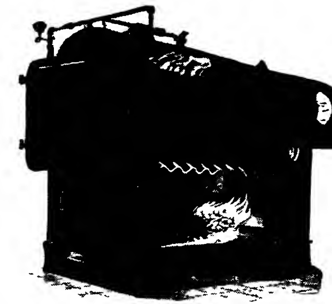
Seventy years of intensive study and effort are back of Kewanee Boiler designs. They are all constructed in our extensively equipped factory at Kewanee, Illinois, in conformity with these Codes: *American Society of Mechanical Engineers* for construction, and for rating with the *Steel Heating Boiler Institute* Simplified Practice.

The Kewanee series include:

HEAVY DUTY RIVETED FIREBOX TYPES: *1,240 ft to 42,500 ft Brickset and portable settings, Updraft and Downdraft Smokeless Furnace, Single-pass tubes for rear smoke outlet, Two-pass tubes for front smoke outlet.*

WELDED BOILERS: *2,200 ft to 42,500 ft. Direct Draft or Smokeless Arch with Corrugated Crown Sheet. Rear Smoke outlet and Weld+Rivet for front Smoke outlet.*

RESIDENCE STEEL BOILERS *320 ft to 2,924 ft Square and Round Type "R" with and without Jackets and Hot Water Heating Coils for Storage Tank or Instantaneous flow*



Kewanee Welded Type C Boilers

Keuanee Residence Square Type R Boilers

SPECIFICATIONS—BRICK-SET AND TYPE "K" PORTABLE UP-DRAFT BOILERS

Table for two series of Boilers lists maximum dimensions only

Boiler No.		3 3K	4 4K	5 5K	6 6K	8K	8 9K	9 10K	10 11K	11 12K	12 13K	13 14K	14 15K	15 16K	17 18K	18 20K	19	20
Rated Steam Capacity.																		
Coal	Sq Ft	1240	1380	1800	2200	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000
Oil, Gas or Stoker	Sq Ft	1770	2020	2190	2680	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290
Width and Length	In. x Ft In	30x9-10	36x8-10	36x10-4	36x11-10	42x11-4	42x12-10	48x11-10	48x13-4	48x14-10	54x15-3	54x17-9	60x17-0	60x19-6	66x19-6	72x19-6	72x16-0	72x18-0
Overall Height Shell	In.	59	65	65	65	71	71	77	77	77	83	83	93	93	99	105	107	107
Height of Water Line	In	52	55	55	55	58½	58½	61	61	61	66	66	75	75	80	85½	85½	85½
Approximate Weight Coal	Lb	3500	3900	4200	4700	5700	6400	7000	7600	8200	9500	10700	12600	14500	17600	20700	19400	21900
Oil	Lb	3130	3550	3800	4200	5100	5800	6400	6900	7400	8600	9700	11500	13300	16100	19100	17700	20000

Boiler No.	376	377	378	379	380	381	382	383	384	385	386	387	388	389	390
Rated Steam Capacity:															
Coal ... Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker ... Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length ... In. x Ft In.	42x8-3	42x9-3	48x8-7½	48x9-5	54x11-2½	54x12-11	60x12-8½	60x14-9	66x14-11	66x17-4	72x16-5	78x17-0½	78x20-7½	84x19-11	84x22-8
Overall Height Shell ... In.	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line ... In.	70	70	73	73	78	78	85	85	88½	88½	94½	95	95	107½	107½
Approximate Weight: Coal ... Lb	6800	7500	8100	8800	10000	11100	13600	15300	18000	20500	22900	25100	29500	33600	37500
Oil ... Lb	6200	6900	7400	8000	9100	10100	12500	14100	16500	18900	21200	23200	27500	31500	35200

SPECIFICATIONS—PORTABLE UP-DRAFT BOILER

Boiler No.	576	577	578	579	580 480	581 481	582 482	583 483	584 484	585 485	586 486	587 487	588 488	589 489	590 490
Rated Steam Capacity:															
Coal ... Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker ... Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length In. x Ft In.	42x8-7	42x9-6½	48x8-10	48x9-6½	54x11-5	54x13-1½	60x13-4½	60x15-5½	66x15-6½	66x18-0½	72x17-0	78x17-7½	78x21-3½	84x20-7	84x23-4
Overall Height Shell ... In.	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line ... In.	70	70	73	73	79½	79½	84½	84½	89½	89½	94	96½	96½	105	105
Approximate Weight: Coal . Lb	6100	6700	7300	7900	10200	11400	13300	14800	17300	19700	22000	24200	28400	32300	36100
Oil . . Lb	5500	6100	6600	7100	9300	10400	12200	13600	15800	18100	20300	22300	26400	30200	33800

SPECIFICATIONS—TYPE "C" WELDED BOILER

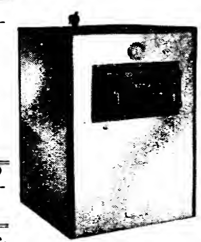
*Boiler No.	2773	774 2774	775 2775	776 2776	777 2777	778 2778	779 2779	780 2780	781 2781	782 2782	783 2783	784 2784	785 2785	786 2786	787 2787	788 2788	789 2789	790 2790
Rated Steam Capacity:																		
Coal ... Sq Ft	2200	2600	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker . Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width & Length. In.xFt in.	36x5-10	36x6-4	36x6-10	36x7-9	42x7-10½	42x8-6½	42x9-2	48x9-4½	48x10-7	54x9-11	54x11-2½	60x11-6½	66x12-3½	72x12-1½	72x13-4½	78x14-9½	84x14-2½	84x15-11½
Overall Height Shell . . . In.	77½	77½	77½	77½	83½	83½	83½	86½	86½	99	99	108½	112	118	118	122	135	135
Height of Water Line . . In.	69	69	69	69	72	72	72	73	73	85	85	94	95	101	101	103	114	114
Approximate Weight:																		
700 Series, Coal . . . . Lb	....	3900	4400	5000	5500	6000	6500	7500	8400	9700	11000	12900	14900	16600	18400	22000	25200	28400
2700 Series, Coal . . . . Lb	3400	3800	4300	4800	5300	5800	6300	7200	8100	9400	10600	12500	14400	16100	17900	21200	24400	27500
Oil . . . . Lb	2900	3300	3700	4100	4600	5000	5400	6100	6900	8000	9100	10700	12300	13800	15400	18100	20900	23400

*Boiler Series 1773-1790 for Oil, Gas, Stoker; Series 2773-2790 for Anthracite; Series 7L73-7L90 Hi-Firebox for Stoker, 2680 Ft to 42500 Ft.

SPECIFICATIONS—RESIDENCE SQUARE TYPE "R" BOILER ROUND "R"

*Boiler No.	742	743	745	746	747	748	734	735	736	1737
Rated Steam Capacity: Coal . Sq Ft	790	1000	1350	1600	1780	1960	320	410	550	...
Oil, Gas or Stoker ... Sq Ft	840	1120	1470	1900	2160	2380	400	540	680	900
Width and Length ... In. x In.	32x39½	32x45½	32x45½	32x51½	32x57½	32x63½	...	...	...	...
Diameter Firebox Inside ... In.	...	...	...	...	...	...	20	23	26	26
Overall Height Shell Top ... In.	59½	59½	70¼	70¼	70¼	70¼	54¾	55¼	58½	61
Height of Water Line ... In.	48	48	58½	58½	58½	58½	44	44	46	49
Approximate Weight: Coal ... Lb	2150	2360	2800	3050	3300	3550	880	1040	1280	...
Oil ... Lb	1900	2060	2500	2730	2920	3125	800	920	1130	1400
Standard Jacket, Crated ... Lb	205	225	175	190	200	225	...	190	215	...
Boiler No., Square "R" Oil or Gas		83R1	83R2	83R3	83R4	83R6	83R7	83R8	83R9	
Rated Steam Capacity ... Sq Ft		901	1105	1326	1513	2091	2363	2652	2924	
Approximate Weight with Jacket ... Lb		1600	1800	2000	2200	2800	2900	3100	3300	

*Boiler Series 1742-1748, 1734-1737 for Oil, Gas or Stoker; 2742-2748, 2734-2736 for Anthracite. Kewanee Indirect Hot Water Heating Coils for Type C, Square and Round "R" Boilers; 170

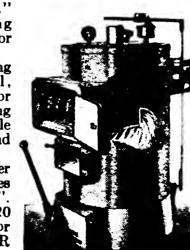


Rex Square Jacket

Kewanee Round "R" and Square Type "R" Residence Heating Boilers are made for all fuels.

Standard snug fitting jackets or Regal, Regal-Square styles for completely enclosing burners are available for Round "R" and 83R.

Hot Water Copper Coil. 65-75 gal comes with Round "R". Capacities up to 720 may be ordered for Square and 83R Boilers.



Round Jacket

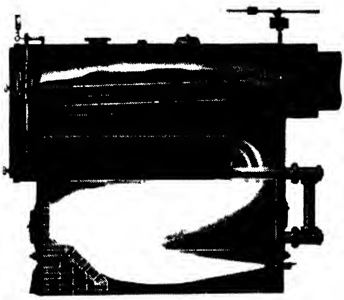
Pacific Steel Boiler Division

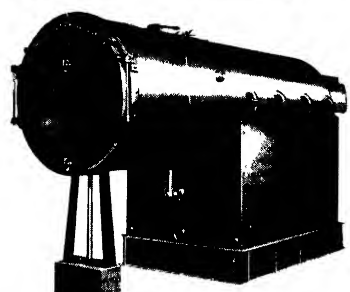
United States Radiator Corporation

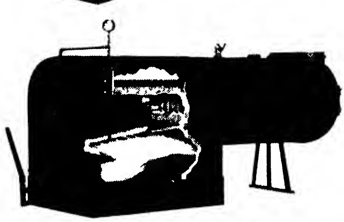
General Offices: **Detroit, Michigan**

Sales Offices in Principal Cities

A Complete Line of Low Pressure Steel Heating Boilers







All Pacific Boilers are built using the *A.S M.E.* Boiler Code Standards as minimums

LOW WATER LINE SERIES

Built in the following capacities for steam:

Coal Burning Sizes—1800 to 35,000 sq ft

Mechanically Fired Sizes—2190 to 42,500 sq ft.

High Fire Box for Stoker Firing—Sizes—2190 to 42,500 sq ft.

All Pacific Boilers are built, inspected, and tested under the supervision of the Hartford Steam Boiler Inspection and Insurance Company

TWO-PASS FRONT SMOKE OUTLET

Built in the following capacities for steam:

Coal Burning Sizes—4000 to 30,000 sq ft.

Mechanically Fired Sizes—4860 to 42,500 sq ft.

All Pacific Boilers are made of steel with each joint and seam electrically arc-welded—built to last a life-time

SINGLE-PASS REAR SMOKE OUTLET

Built in the following capacities for steam.

Coal Burning Sizes—1800 to 6000 sq ft

Mechanically Fired Sizes—2190 to 7290 sq ft.

PACIFIC THREE-PIECE CONSTRUCTION

Made up of three parts, shell, firebox and base, Pacific Boilers are particularly adaptable to replacement work. Where necessary Pacific fireboxes can be split (as illustrated) allowing the boiler to be taken into the building in four pieces and erected without welding on the job

Descriptive Bulletins on Pacific Steel Boilers will be mailed on request.

Smith Twin Tubular Boiler Co., Inc.

State Road and Cottman Street

Philadelphia, Pa.

MANUFACTURERS STEEL HEATING BOILERS

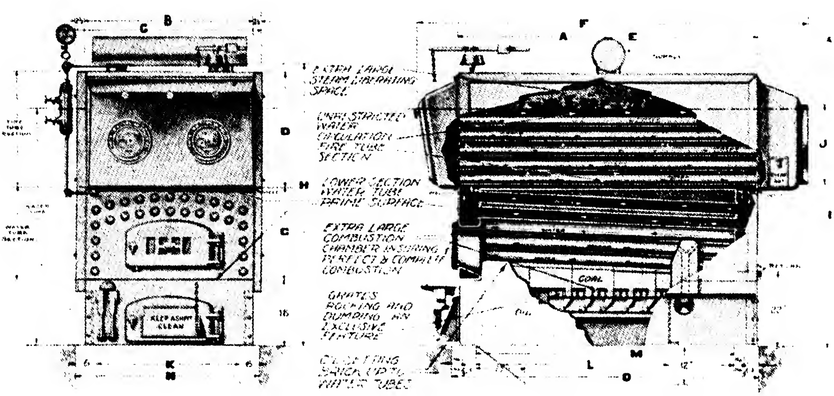
Smith Sectional Steel Boilers have gained favor with Architects and Engineers due to their compact construction and adaptability for installing in boiler rooms of building through existing doors or openings. Built on foundations or f.o.b. shop for coal, stoker or oil firing.

All the outstanding advantages of both the water tube and fire tube boilers without complicated baffle construction.

Faster steaming and higher efficiency obtained by rapid water circulation due to water tube construction that comprises 50 per cent of the boiler heating surface.

Write for circular for Domestic Boilers especially designed for Oil or Stoker firing.

SPECIFICATIONS—SMITH SECTIONAL-STEEL HEATING BOILER



Number of Boiler	2-60	2-72	2-84	2-96	2-108	3-84	3-96	3-108	3-120	4-108	4-120	4-132	4-144	5-132	5-144	5-156
Steam R. (Oil-Stoker)	4800	5800	6800	7800	8800	10000	11400	12800	14200	16000	17800	19600	21400	24800	27100	29300
Heating Surface	285	343	401	458	525	589	672	753	835	944	1050	1154	1262	1462	1593	1724
Water Line In.	65	65	65	65	65	71	71	71	71	78	78	78	78	85	85	85
Floor to Water Tubes In.	42	42	42	42	42	42	42	42	42	46	46	46	46	50	50	50
Furnace Vol Cu Ft	51	63	75	88	100	93	108	124	139	161	181	201	221	253	279	304
Length of Shell In. A	60	72	84	96	108	84	96	108	120	108	120	132	144	132	144	156
Width of Shell In. B	51	51	51	51	51	61	61	61	61	71	71	71	71	81	81	81
Height Bottom Sec. C	31	31	31	31	31	42½	42½	42½	42½	48½	48½	48½	48½	53½	53½	53½
Height Top Sec. D	28	28	28	28	28	26½	26½	26½	26½	30½	30½	30½	30½	34½	34½	34½
Length Top Sec. E	60	72	84	96	108	84	96	108	120	108	120	132	144	132	144	156
Length of Boiler In. F	84	96	108	120	132	110	122	134	146	136	148	160	172	160	172	184
Width of Boiler In. G	56	56	56	56	56	66	66	66	66	76	76	76	76	86	86	86
Height of Boiler In. H	72½	72½	72½	72½	72½	78½	78½	78½	78½	84½	84½	84½	84½	92	92	92
Height Top Header I	84	84	84	84	84	90	90	90	90	98	98	98	98	107	107	107
No. & Size Outlets In.	1-8	1-8	1-8	1-8	1-8	1-8	1-8	1-8	1-8	1-10	1-10	1-10	1-10	1-10	1-10	1-10
No. & Size of Returns In.	1-4	1-4	1-4	1-4	1-4	1-4	1-4	1-4	1-4	1-6	1-6	1-6	1-6	1-6	1-6	1-6
Dia. Smoke Collar J	20	20	20	20	20	24	24	24	24	30	30	30	30	36	36	36
Height Stack Ft	50	55	60	65	70	65	70	75	80	75	80	85	90	85	90	95
Length of Pit In. M	50	62	74	86	98	68	80	92	104	92	104	116	128	104	128	140
Width Foundation N	56	56	56	56	56	66	66	66	66	76	76	76	76	86	86	86
Length Foundation O	62	74	86	98	110	86	98	110	122	110	122	134	146	134	146	158
Width Ash Pit K	44	44	44	44	44	54	54	54	54	64	64	64	64	74	74	74
Length Ash Pit L	50	50	56	56	62	52	58	64	64	58	64	70	70	64	64	70
Steam R. (Hand-Fired)	4000	4800	5600	6400	7200	8250	9400	10500	11700	13200	14700	16200	17700	20500	22400	24000
Grate Size In.	46x48	46x48	46x54	46x54	46x60	56x54	56x60	56x66	56x66	66x60	66x66	66x72	66x72	76x66	76x66	76x72
Grate Size Sq Ft	15.3	15.3	17.3	17.3	19.1	21.0	23.3	25.6	25.6	27.5	30.0	33.0	33.0	34.8	34.8	37.3

Standard Ratings conforming with the Industries "Simplified Practice Recommendations."

Special High-furnace Boilers furnished for Oil Burners or Stokers.

S. T. Johnson Co.

940-950 Arlington Avenue, **Oakland, California**

Branches:—San Carlos, Cal ; San Francisco, Cal ; Sacramento, Cal ; 401 N. Broad St , Philadelphia, Pennsylvania.

Founded in 1904, S. T. Johnson Co. is one of the oldest and largest manufacturers of oil burning equipment and offers a complete line from small domestic to large industrial units, with manual or full automatic ignition and control, burning from the lighter fuel oils to the heaviest grades . . . it includes combination oil and natural gas burners, burners with Low Fire Starting and Modulated Firing accessories, the Laddi Du-all Boiler-Burner unit and the air-conditioning, ventilating and heating unit, "Selectair."

The company provides complete engineering and technical service for assistance in the solution of heating and air-conditioning problems. Representatives in principal cities throughout the world.

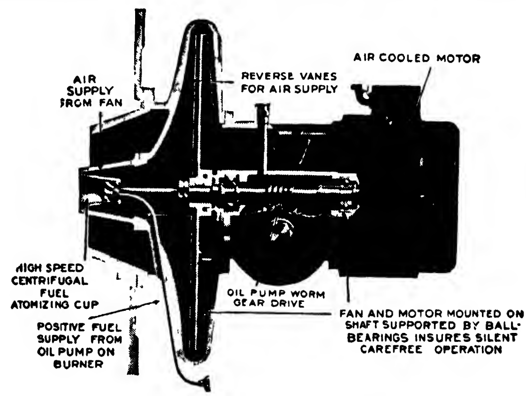
JOHNSON COMMERCIAL TYPE ROTARY OIL BURNERS

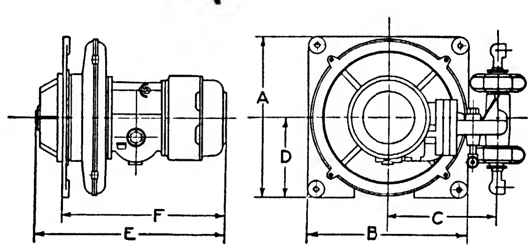
(Capacities up to 55,600 sq ft steam radiation or its equivalent).

The three commercial types of Johnson Rotary Oil Burners are of the same basic construction, consisting of motor, fan, housing and firehole plate, concentrically mounted in a compact unit . . . easily installed on any make of boiler. Exact amount of air necessary for perfect atomization is supplied by motor-driven fan and adjusted by butterfly valve. Complete combustion of cheaper grades of fuel oil is accomplished by the Johnson method of centrifugal atomization An oil-air and water separating tank prevents foreign matter entering the fuel.

TYPE 30-AV AND 30-AVH FULLY AUTOMATIC

The 30-AV uses up to No. 5

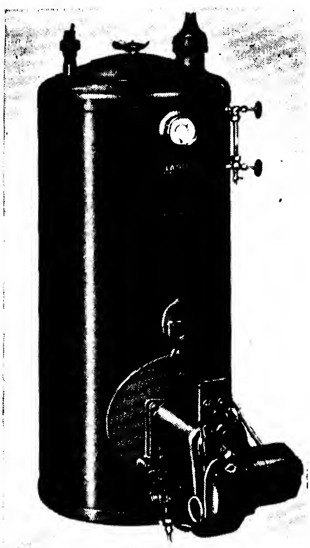




Dimensions and Capacities of Types 28, 30AV and 30AVH Burners

Burner Size No.	Capacity Steam *Radiation Sq Ft		Boiler Hp Rating	Gallons Oil per Hour		Size Motor Hp	Lb Approximate Shipping Weight			Dimensions in inches					
	Min	Max		Min	Max.		28H	30AV	30AVH	A	B	C	D	E	F
2½	825	2775	20	2	7	⅓	220	275	285	14	13	9½	6 15/16	17	15
3½	1400	6950	50	3	15	½	250	300	320	16 8/16	15⅞	10¼	8¼	17½	15½
4½	2775	13900	100	6	30	¾	400	470	505	17	19½	13	9 5/16	21	19¼
150½	4150	20900	150	9	45	1½	460	525	560	21½	21½	13	11	24	20
5½	5550	27800	200	12	60	2	475	540	575	21½	21½	13	11	24	20
6½	8325	41700	300	24	90	3	525	575	625	21½	21½	13	11	25	20½
7½	18500	55600	400	40	120	3	725			21½	24	14	11	25	20½

*Boiler output Type 28 Burner available without built-in pump in same capacities.



'Laddi' Du-All

fuel without pre-heating. Johnson-built viscosity compensating control valve furnishes constant flow of fuel to burner, regardless of fuel temperature or viscosities and without excessive pressure on lines or undue strain on pump or motor. Built in six sizes, up to 41,700 sq ft steam radiation.

The 30-AVH is designed for efficient use of heavy No. 6 fuel oil. Fundamentally, this is the standard 30-AV burner with equipment for pre-heating the fuel. Built in six sizes, up to 41,700 sq ft steam radiation.

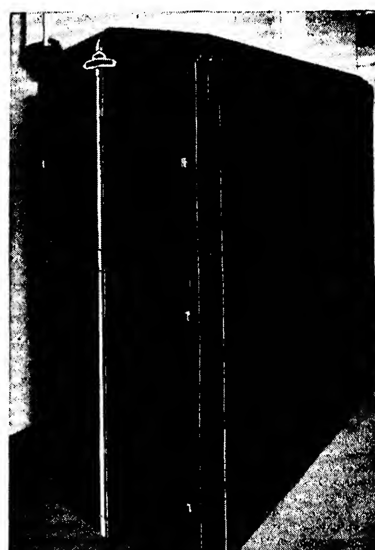
Type 28 Semi-Automatic Burner, manually ignited, is made for heating boilers up to 55,600 sq ft of steam radiation or 400 boiler horsepower. Built in seven sizes, with built-in oil pump, and seven sizes for use with separate oil-pumping equipment. Fuel oils (pre-heated when necessary) as heavy as No. 6 may be used.

'Laddi' Du-All combination boiler-burner unit for small and medium-sized homes, is fully automatic, odorless, quiet and trouble-free. It furnishes heat or domestic hot water or both in combination at low cost. Made in three sizes: 100,000, 120,000 and 140,000 Btu per hour output. These units rest on the floor. Diameter at base is 24 in. on all three sizes while heights are 4½ ft, 5 ft, and 5½ ft respectively.

SPECIFICATIONS: Vertical 3/16 in. copper-bearing steel boiler shell electrically welded, tested to 150 lb hydrostatic pressure. Horizontal firebox with pre-cast refractory combustion chamber; seamless extra heavy copper-bearing steel fire tubes; reversible chimney connector for either top or back connection. Completely insulated with mineral wool. Heavy steel enameled jacket. Johnson Type B or BH burner. Electric ignition or combination electric-gas ignition. Each unit complete with necessary controls. Burns either Diesel fuel oil or No. 3 standard. Each unit fully assembled and tested at factory before shipment.

The Pressure Atomizing Burner used with the "Laddi" is available as a conversion burner for all types of furnaces and boilers. It is built in four sizes, covering a range from ¾ to 7½ gal per hour, equivalent to 300 to 3100 sq ft of steam radiation (E.D.R.).

Air-Conditioning Unit "Selectair" combines in one compact unit every feature desired by home-owners, architects and engineers for economical heating, air-conditioning and ventilating the modern home. Supplies year-'round hot water for domestic use. Forced circulation of air for summer needs. Equipped with Johnson "Bankheat" Pressure Type Burner.



"Selectair"

Capacities and Dimensions of SELECTAIR Units

Size No.	Max. Rating* Btu per hour	Cu Ft Air per min. ¼ S. P.	Heat Exchanger Btu per hour	Size Stack Connection	Oil Rate Gal per hour	Cabinet Width	Dimensions	
							Length	Height
1	125,000	1,485	105,000	7″	1 to 1¼	27″	60″	54″
2	200,000	2,320	166,000	8″	1¼ to 2	31″	69″	63″
3	300,000	3,300	235,000	9″	2 to 2¾	33″	86″	69″

*Total output includes steam to heat exchanger, heat for domestic hot water, and heat available for installed radiation if used.

The Cooling Tower Company, Inc.

15 John Street, **New York, N. Y.**

A DIVISION OF THE FLUOR CORPORATION, LTD.

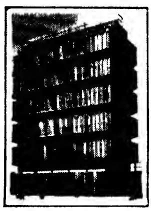
909 E. 59th St., Los Angeles, Calif.—Mail Address: P. O. Box 128, Station K., Los Angeles, Calif.

Representatives in Principal Cities

Manufacturers of

Aerator Type Cooling Towers, Spracoolers, Forced Draft and Induced Draft Cooling Towers, Spray Nozzle Cooling Systems, Spirodome and Impact Type Spray Nozzles, Louvered Spray Fences, and Indoor Type Cooling Towers.

Fluor Aerator Type Cooling Towers



Our successful Aerator type design is such that no straight line passage of air or water through the cooling tower is possible. This results in (1) undisturbed water distribution through the tower, (2) elimination of windblown spray, and (3) uniform cooling performance Structurally, these towers are sturdy California Redwood, the structural material, is selected because of its characteristic strength, durability and long life, as well as its low maintenance cost The tower is rigidly cross-braced transversely and longitudinally with full cross section beams, providing equal strength in tension or compression Architecturally, the completed tower is an outstanding example of modern industrial design. The distribution system of the tower consists of a longitudinal pipe header with openings in the top connecting to two large transverse distributing arms in each bay. Water distribution is effected by double staggered bowed panel type deck surfaces, effecting perfect water filming with fastest dissipation of heat. Wind pressure drop through the tower is negligible with wind velocities under six miles per hour Successful operation at any outside temperature, in any location, with any wind velocity, is assured Aerators are standard on all Fluor Towers, from the smallest to the largest. Towers are built in any sizes from five gallons per minute upward.

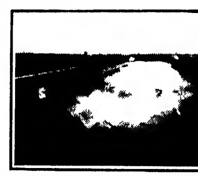
Type A Spracoolers

For use where cost will not permit a standard deck type cooling tower and space will not allow a spray pond to be installed. Wood or steel construction, light in weight, equipped with an efficient nozzle distributing system.

Mechanical Draft Cooling Towers

We fabricate, erect, and guarantee forced and induced draft cooling towers of any capacity from one ton up, for indoor or outdoor use Special Balke type filling provides for thorough mixing of air and water, resulting in efficient cooling

Spray Nozzle Systems



We make two types of spray nozzles—Impact spray nozzle which throws a flat, fan-shaped spray cloud Spirodome spray nozzle producing a high conical spray cloud suitable for general installations

Either type may fit requirements and our Engineering Department will submit proposals covering the best type for a particular purpose

Outstanding features of our spray pond systems are:

(1) Non-clogging nozzles.

(2) Freedom from interruption of operation.

(3) Automatic drain and flushout valve.

(4) Well designed spray fence for protection against spray dirt.

Indoor Cooling Towers

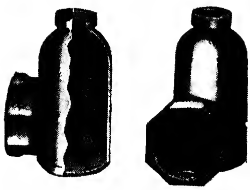
For indoor installations of one to forty tons capacity Available in commercial type or quiet rating. Steel shell, Balke type wood filling, drift eliminators, float valve, access door Easy to erect, economical to operate. Low pumping head Shipped complete or knocked down.

The Marley Company

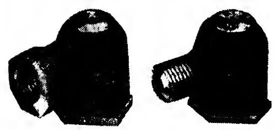
(Fairfax and Marley Roads,) **Kansas City, Kansas**

Branches or Agents in Principal Cities

Spray Nozzles and a Complete Line of Water Cooling Equipment



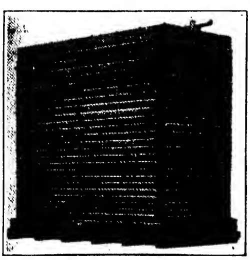
MARLEY Standard Water Cooling Nozzles for Spray Pond Use, and Cooling Towers.



MARLEY Small 2-Piece Nozzles for Brine Spraying, Air Washing and Similar uses.

MARLEY Ice-Melting Nozzle for cooling systems using ice.

MARLEY Humidifying Nozzle adds moisture to air in open rooms or duct system.



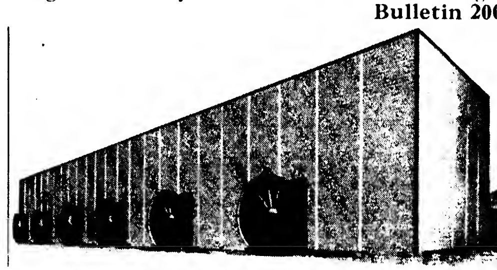
One of the small sizes of MARLEY Atmospheric Spray Towers.

MARLEY PATENTED NON-CLOG SPRAY NOZZLES

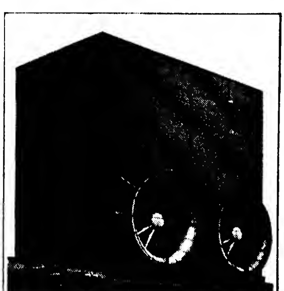
Made in scores of types and sizes. Practically any metal or alloy the purpose may demand. **Bulletin 100.**

MARLEY ATMOSPHERIC SPRAY TOWERS

For all industrial water cooling services, refrigeration, Diesels, etc. Virtually unlimited range of sizes. Entirely shop fabricated. Rugged, efficient. Low initial maintenance and operating costs. Many exclusive MARLEY advantages. **Bulletin 200.**



Large MARLEY Double-Cased Redwood Forced-Draft Tower.



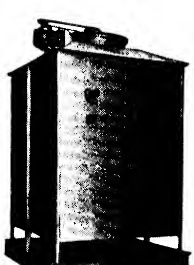
MARLEY Steel Forced-Draft Tower typical of roof installations for air conditioning.

MARLEY STANDARD FORCED DRAFT TOWERS

For heavy-duty heat-dissipating services of all kinds. Any capacity. MARLEY patents cover a variety of important features for extreme operating flexibility, high efficiency and economy. **Bulletin 85.**

MARLEY STANDARD INDUCED DRAFT TOWERS

For same types of service as above but where the induced draft principle is more desirable or efficient. Same MARLEY patented features and engineering.

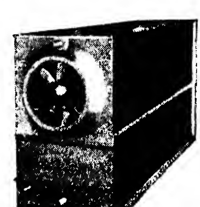


MARLEY "SMALL SERIES" STEEL INDUCED DRAFT TOWERS

For jacket water cooling of engines, compressors and other small equipment. Popular in refrigerating and air conditioning, 3 tons and up. 28 standard models for any space or capacity requirements.

Vertical style for outdoor service, **Bulletin 500.**

Horizontal style for indoor service, **Bulletin 502.**



Also MARLEY Spray Coil Towers, Large Induced Draft Coil Towers, Deck-Type Atmospheric Coil Towers, Spray Ponds and Related Equipment.

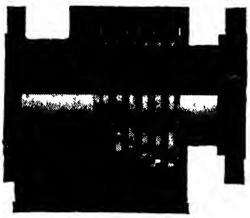
AMERICAN DISTRICT STEAM COMPANY

NORTH TONAWANDA, N.Y.

IN BUSINESS OVER SIXTY YEARS

Branches and Agents in Principal Cities

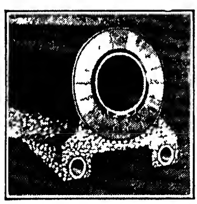
PRODUCTS for STEAM SERVICE



Packless, U-Ring Type

PACKLESS, U-RING TYPE JOINT

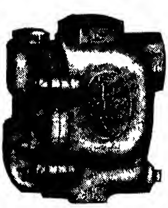
A compact, fully guided, packless joint with steel body and fully enclosed expansion element for high pressures and high temperatures. Requires no servicing The element is a series of welded U-rings of stainless alloy steel Particularly adaptable where space is limited Pressures to 300 lb and temperatures to 750 F. Write for Bulletin No. 35-50G.



Red Diamond Brand Casing

RED DIAMOND BRAND CASING

Combined conduit and insulation for underground steam or hot water lines, made of kiln-dried, wire-bound, strong wood staves with a waterproof covering It is good for 25 or more years of efficient service as proved by letters from satisfied users Easily and quickly installed Write for Bulletin No 35-65G.



ADSCO Vertical Steam Trap

ADSCO VERTICAL STEAM TRAP

A float type steam trap with or without thermostatic air by-pass for vacuum service to 15 lb pressure and gravity service to 125 lb pressure The cover with all working parts can be removed without disturbing the piping connections The trap is equipped with a reversible valve and reversible seat of stainless alloy steel. Write for Bulletin No 35-86G.

ADSCO Instantaneous Water Heater

ADSCO HEAT EXCHANGERS

Made in various sizes and capacities to heat or cool water, oils, other liquids or gases according to expert engineering specifications Simple in design, sturdy in construction, dependable and economical in operation Available in U-tube or straight tube types of heaters, economizers, condensate coolers or special units Write for Bulletin No 35-75BG, 35-76G

Rotary Condensation Meter

ROTARY CONDENSATION METER

Measures steam consumption by metering condensate from heating systems or industrial equipment. Accurate within 1 per cent and factory tested to 150 per cent of rated capacity Compact, easily cleaned, tamper-proof and equipped with non-fogging counter mechanism. Counter reads directly in pounds. Suitable for vacuum or gravity service. Available in 7 sizes from 250-12,000 lb per hour capacity. Write for Bulletin No. 35-80AG.

(*See also Page 1039*)

E. B. Badger & Sons Co.

General Office: 75 Pitts Street, **Boston, Mass.**

Representatives

Atlanta, Ga.	140 Edgewood Ave.	Minneapolis, Minn.	732 Builders Exchange
Charlotte, N. C.	1408 Independence Bldg.	Montreal, Quebec	1411 Crescent St.
Chicago, Ill.	1307 So. Michigan Ave.	New Orleans, La.	916 Union St., Room 203
Cincinnati, Ohio	1801 Carew Tower	New York, N. Y.	271 Madison Ave.
Cleveland, Ohio	Guardian Bldg.	Philadelphia, Pa.	1500 Walnut St.
Denver, Colo.	725 Denver National Bldg.	Pittsburgh, Pa.	302 Benedum Trees Bldg., Fourth Ave.
Detroit, Mich.	424 Book Bldg.	Salt Lake City, Utah	Kearns Bldg.
Houston, Texas	4421 Rusk Ave.	San Francisco, Calif.	Sharon Bldg.
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Kansas City, Mo.	1332 Oak St.	St. Louis, Mo.	4060 W. Pine Blvd.
Los Angeles, Calif.	609 So. Anderson St.	Washington, D. C.	1103 Vermont Ave., N. W.

ENGINEERS AND MANUFACTURERS

Manufacturers of Copper and Stainless Steel Badger Corrugated Expansion Joints; Engineers and Manufacturers of Chemical Apparatus; Engineers on Process Work; Designers of Complete Plants.

Forty years' experience in design, manufacture and application are back of Badger Expansion Joints. Over the years, many improvements in design have been brought out, resulting in better construction and longer life. Every advantage has been taken of controlled heat treatment throughout fabrication to obtain its benefits. The Directed Flexing feature, one of the more recent developments, assures full distribution of flexing stresses by progressively controlled flexing. The all-curve corrugations and correspondingly shaped equalizing rings provide this. The adoption of Stainless Steel, another recent development, solves the problem of using a "packless" type expansion joint under high temperatures, high pressures and special corrosive conditions.

Badger Expansion Joints are of the packless type, requiring no maintenance during their long life. This feature makes them particularly useful in crowded quarters or in underground tunnels where space is at a premium. They are compact and easily installed and insulated. They do not have to be serviced after installation.

Three distinct designs are available, each described in an illustrated bulletin. Copies of bulletins will be sent on request.

Bulletin No. 100
Directed Flexing, Self-Equalizing

List prices, weights and dimensions of copper and stainless steel expansion joints together with installation data on this design which is used for traverses ranging from fractions of an inch up to 6 inches single and 12 inches double; for pressures ranging from vacuum to 200 pounds (copper) and 300 pounds (steel); for temperatures from sub-zero to 500 F (copper) and 900 F (stainless steel).

Bulletin No. 200
Non-Equalizing

List prices, dimensions, weights of copper and stainless steel expansion joints and other data on the design which is used principally to absorb vibrations or lateral displacement between piping and equipment or between connected equipment. Traverse is limited to ½ in. or less and line pressure to 25 pounds or less. Standard shapes: round, square, rectangular, or oval; special shapes to order.

Bulletin No. 300—Flexible Seal

Dimensions and other data on a modification of the standard Badger Non-Equalizing Joint for use on pipe lines passing through walls, foundations, decking, bulkheads, etc. Its function is to allow for line movements such as expansion, contraction, lateral displacement or vibration yet completely seal the opening so that water or other liquids will not work back along the line into buildings or compartments. Made standard in carbon steel; other metals to order. List prices on application.

American Coolair Corporation

COOLING AND VENTILATING FAN SYSTEMS

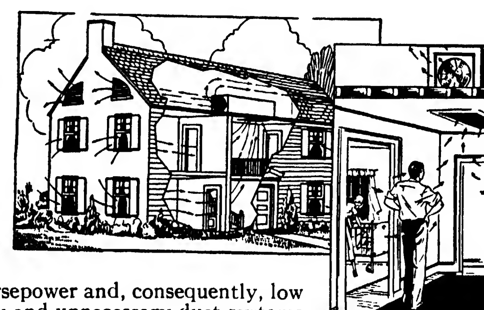
3604 Mayflower Street, **Jacksonville, Florida**

PRODUCTS

Silent Reversible Belt Drive Cooling and Ventilating Fans for homes, stores, offices, factories, etc. Quiet, high speed direct drive fans.

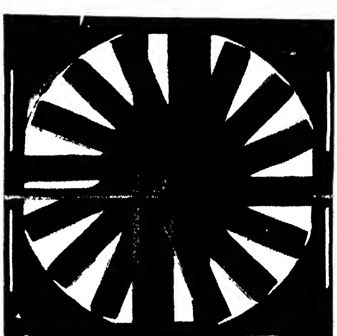
DIFFERENTIATE clearly between cooling and ventilating. Cooling requires complete air changes as rapidly as every half minute to every minute and one-half; ventilation requires air changes only every two to five minutes, depending on the building. Coolair has pioneered home cooling by air circulation.

One of the outstanding advantages of COOLAIR ventilating fans is that they will deliver large quantities of air under free delivery with small horsepower and, consequently, low operating cost. Elimination of costly and unnecessary duct systems will often more than pay for the COOLAIR equipment. It is important to provide ample exhaust and intake openings to avoid restricting the air volume. Net free air passage area should never be less than area of fan. For maximum quietness free area should be enough larger than fan to assure a grille velocity of not more than 750 fpm.



Typical Coolair Home Cooling and Ventilating Installation

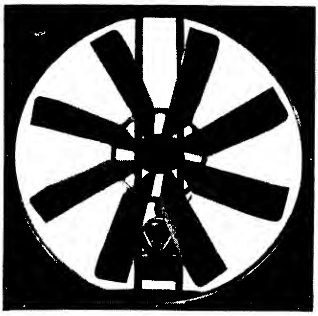
Description



Standard Type—Front View
Diameters Six to Nine Feet
Performance Data on Request

COOLAIR Belt Drive Fans meet the ventilating and cooling requirements of any size building They are just as efficient when blowing in as when exhausting. The V-belt drive permits use of standard electric motors operating at the most efficient speed, and easily replaced in event of motor trouble. Slower speed motors are costly, difficult to replace and less efficient With simple exchange of a motor the COOLAIR unit can be converted from a silent, slowly turning residence unit using a small motor, to a high speed industrial unit using a large motor.

With a reversible motor, air can be blown into or exhausted from the building at will. Air can be pulled from the cool side of the building in the morning and the fan reversed when the sun moves to the opposite side in the afternoon.



Coolair Belt Drive
Diameters 26 in. to 62 in.

Over-All Dimensions (In.)

Size	Height and Width	Depth
2	30⅝	12
2½	36⅝	12
3	42⅝	12
3½	49	16
4	55⅛	17½
4½	61⅛	18½
5	67⅝	16½
*6	75¼	28
*7	87½	34½
*8	99½	34
*9	112	34

*Standard type frames.
Note: Depth Approximate.

Performance Data—Coolair Belt Drive Fans

Fan Size	Hp	Fan RPM	Cu Ft Air per Min
2	° ¼	448	4600
	† ¼	535	5500
	†† ⅓	595	6100
2½	° ¼	395	7500
	† ⅓	432	8200
	† ½	495	9400
	†† ¾	567	10700
3	° ¼	308	10000
	† ⅓	335	10800
	† ½	394	12800
	†† ¾	440	14200
	††1	485	15700
3½	° ⅓	244	12500
	† ½	294	15000
	† ¾	325	16600
	††1	358	18300
	††1½	410	21000
4	° ⅓	211	15500
	† ½	248	18200
	† ¾	290	21300
	††1	320	23500
	††1½	366	26800
4½	° ½	210	21800
	† ¾	236	24500
	†1	260	27000
	††1½	298	30900
	††2	328	34000
5	° ½	175	24600
	† ¾	201	28200
	†1	222	31200
	†1½	254	35700
	††2	280	39300
	††3	321	45100

°Very quiet. †Quiet. ††Industrial.
Data in accordance with Standard Code Test,
American Society of Heating and Ventilating Engineers.

Advantages

(**1**) Low initial cost. Great air delivery per dollar of power.

(**2**) Economical operation. Use of a small motor at the most efficient speed.

(**3**) Quiet performance. Great volumes of gently moving air without objectionable noise or drafts.

(**4**) May be equipped with reversible motors to exhaust or blow in at will.

(**5**) Rubber insulated motor support. Adjustable to belt tension.

(**6**) Equipped with SKF ball bearings in dustproof, grease-packed housings.

(**7**) Easy to install. The steel blades and frame are durable, yet light in weight.

(**8**) Standard motors. Enclosed when specified. No vent holes are necessary to keep them cool.

(**9**) Blades are individually mounted, inexpensive to replace.

(**10**) Coolair Belt Drive units, when equipped with ball bearing motors, are suitable for operation in any position.

(**11**) Residence units are light, spring suspended for ultra-quiet operation.

Coolair Direct Drive Fans

Coolair Direct Drive—16 in. to 24 in.
Data on Request

Covered by U. S. Patent No. 1,855,660, designed for maximum air per horsepower with minimum noise.

Type "B" Direct Drive COOLAIR Fans are made in four sizes. At 1200 rpm for residence kitchens, small stores, etc. At 1800 rpm for cooling, drying; for removing dust and fumes. Motors totally enclosed, long service types. The 1/6 hp and larger are ball bearing, will operate in extremes of position, temperature and humidity. Cushion mountings.

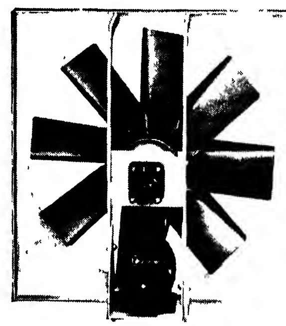
Coolair Residence Unit and Spring Suspension covered by U. S. Patent No. 1,992,112.

Autovent Fan & Blower Company

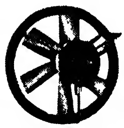
1809-23 N. Kostner Ave., Chicago, Illinois

FANS—BLOWERS **UNIT HEATERS**

Member National Association of Fan Manufacturers and Industrial Unit Heater Associations



Coolvent Fan— Belt Driven Propeller Fan for commercial and industrial applications as well as residential attic fan installations where large volume of air is required with quiet operation at low initial and operating costs Sizes from 24" to 54" diameter Bulletin No 204-A sent on request



Autovent "31" Series Propeller Fans—Will not churn air or overload the motor Especially recommended for economical ventilation Ruggedly constructed Capacities from 500 to 38,000 cfm Write for bulletin No 200

Acid-moisture and Explosion Proof Fans—The acid-Moisture proof units are equipped with fully enclosed motors and specially constructed fan wheels The vapor-explosion proof units designed to handle explosive dusts and gases have underwriters label "Class 1—group D" motors and non-ferrous fan wheels to meet the requirement Write for bulletin No 201.

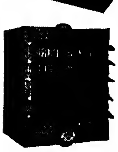
AUTOVENT "31 SERIES" PROPELLER FANS

Constant or Two Speed—Alternating Current—Multiphase—220 or 440 Volts—60 Cycle

Size Type	Motor Hp	Cfm Max.	Appr Rpm Max.	Appr. Shpg Wt Lb	Size Type	Motor Hp	Cfm Max	Appr. Rpm Max	Appr Shpg Wt Lb
16HMN	1/8	1700	1140	64	30HML	1/3	7100	850	210
16HMR	1/5	1950	1725	66	30HMN	½	8100	1140	220
18HMN	1/8	2350	1140	77	36HMT	½	10000	690	335
18HMR	1/4	2950	1725	90	36HML	¾	12500	850	390
20HMN	1/4	3450	1140	115	42HMT	1 0	13500	690	510
20HMR	1/4	3550	1725	115	48HMT	1½	18900	690	650
24HML	1/6	3850	850	145	54HME	2	21500	575	890
24HMN	1/4	4500	1140	150	60HME	2½	26500	480	1200
24HM	1/3	5250	850	175	72HME	3	38000	480	1500

Autovent Uniblade Volume Blowers—Motor driven—universal discharge for fume hoods, chemical labs, processing, drying, forced draft, etc. Handles low volumes of air at medium pressures. Wheels range from 6 in. to 11 in. dia.—same design as heavy duty blowers. Can be mounted on floor, wall or ceiling. Direct Connected Blowers for general ventilating applications, available in wheel diameters up to 25 in. Bulletin No. 300. Forward Curve Belt Driven Blowers No 301.

Autovent "V" Belt Driven Unit Blowers—Forwardly curved blade type with smooth air flow and quiet bearings Motor mounted on steel pedestal, integral with blower housing, making a compact unit. Air delivery can be decreased or increased if desired Interchangeable motors. Sturdily constructed of sheet steel, rolled lock seams Write for bulletin No 300. Backward Curve Belt Driven Blowers No. 302.



Autovent Super-Type Steam Unit Heaters—This suspended type heater forces air circulation and directs warm air to lower part of room. Heating element of flat copper fins, attached to seamless, drawn copper tubes. Tubes run vertically for perfect drainage No welded or brazed joints in coil or header. Fans have non-overloading power feature. Motor furnished to requirement. Write for bulletin No 102

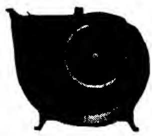
The Complete Line of AUTOVENT Propeller Fans is tested and rated in accordance with the Standard Test Code adopted jointly by the *National Association of Fan Manufacturers* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Bayley Blower Company

1817 S. Sixty-Sixth Street **Branches in Principal Cities** **Milwaukee, Wis.**

Builders of Heating, Ventilating, Cooling, Purifying, Humidifying and Air Washing Equipment; Exhaust and Drying Apparatus, Mechanical Draft and Blast, Fans and Blowers of all Types

Bayley Plexiform Fan:



Is a multi-blade fan for supplying air for heating and ventilating systems, manufacturing processes, drying systems, forced and induced draft systems. It is suitable for handling high or low temperature gases at medium or low pressure. Will deliver maximum quantities requiring minimum space with great economy.

This is a distinct Bayley product, high class material and workmanship, properly designed to avoid excessive vibration and overstressing of parts. Inlets and outlets are properly sized for maximum delivery and maximum efficiency. Fans are furnished in single or double width of any required arrangement and with sleeve or anti-friction bearings.

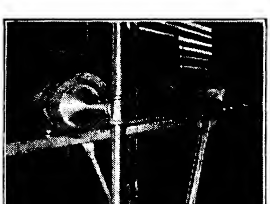
Aeroplex Fan:

Is of high speed design with self limiting power characteristics. Application parallel to the Plexiform Fan. Highly efficient and quiet in operation.

Bayley Exhausters and Pressure Blowers:

Type "B" exhaust fan is for heavy duty, handling refuse from industrial and textile plants. Type "SE" is used in handling smoke, fumes and dust-laden gases. Type "H" for high-pressure work. These units are highly efficient and of high class design and workmanship.

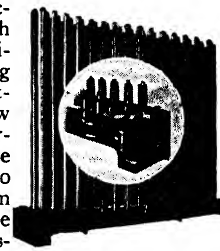
Bayley Turbo Air Washers, Humidifiers and De-Humidifiers:



The Bayley Turbo Air Washer Showing Turbo Atomizer and Eliminator

The Turbo Atomizer used in the Bayley Washer produces a steady, fine spray. Water at low pressure is delivered to the center of a rapidly revolving cone-shaped rotor provided with atomizing pins set in its periphery. This atomizer requires very little attention, and will operate successfully under low water pressure. The orifices are large and this atomizer, unlike high pressure nozzles, cannot clog.

Bayley Chinook Heating Sections:



The Chinook section is used with blast heating, ventilating and drying systems, and is suitable for high or low pressure steam circulation. The base is divided into two chambers. Steam enters (see cut) the lower chamber, rising through $\frac{3}{8}$-in. pipes located within the $1\frac{1}{4}$-in. pipes leading from the upper chamber. Condensation takes place in the larger pipes, the water falling into the upper chamber and draining away through the return outlet. The Chinook can be repaired in the middle of the bank without breaking steam connections or taking down a section.

Shipped assembled in smaller sizes, and knocked down in the larger units. May be installed in horizontal or vertical position.

Bayley Chinookfin Heating Sections:

Are the same design as the Chinook Heaters, using heavy gauge copper fin tubes. As compared with Chinook it is much lighter and occupies less space.

Bayley Plexfin Unit Heaters:

This unit incorporates Chinookfin radiation and Plexiform or Aeroplex fans. The fan assembly including top plate and motor is removable as a unit for maintenance and inspection. The heating element is a removable unit. Casing all welded extra heavy gauge. This is an exceptionally high grade unit at a moderate price.

Buffalo Forge Company

450 Broadway, Buffalo, N. Y.

Branch Offices

- ALBANY, N Y — 611 Standard Bldg.
- ATLANTA, GA — 16th Floor, 22 Marietta Bldg
- BALTIMORE, MD — 404 St Paul St
- BOSTON MASS — 486 Main St
- CHICAGO, ILL — 20 North Wacker Drive
- CINCINNATI, OHIO — 626 Broadway
- CLEVELAND, OHIO — 418 Rockefeller Bldg
- DALLAS, TEXAS — 702 Tower Petroleum Bldg
- DAVENPORT, IOWA—D C Murphy Co, 305 Security Bldg
- DENVER, COLO—Hendrie & Bolthoff Mfg & Supply Co, 1635 Seventeenth St
- DES MOINES, IOWA—D C Murphy Co, 214 Old Colony Bldg
- DETROIT, MICH—Coon DeVisser Co, 2051 W Lafayette Blvd
- GREENVILLE, S C — 312 Franklin National Life Bldg
- HOUSTON, TEXAS — 713 Bankers Mortgage Bldg
- KANSAS CITY, MO — 428 Dwight Bldg
- KITCHENER, ONT, CANADA—Canadian Blower & Forge Co, Ltd
- KNOXVILLE, TENN—C F Sexton — 702 Empire Bldg
- LOS ANGELES, CALIF — 708 Pershing Square Bldg
- MINNEAPOLIS, MINN — 2102 Foshay Tower
- NASHVILLE, TENN—Southern Sales Co, 117 Fifth Ave South
- NEW ORLEANS, LA—Devlin Bros — 1003 Maritime Bldg
- NEW YORK, N Y — 39 Cortlandt Bldg, Room 1110
- PHILADELPHIA, PA — 703 Cunard Bldg
- PITTSBURG, PA — 431 Fulton Bldg
- RICHMOND, VA—T Spencer Williamson, Jr, Inc, Mutual Bldg
- SAN FRANCISCO, CALIF—Moore Machinery Co, 1625 Van Ness St
- ST LOUIS, MO — 1598 Arcade Bldg
- SEATTLE, WASH — 500 First Ave South
- TOLEDO, OHIO — 1922 Linwood Avenue
- WASHINGTON, D C—820 Woodward Bldg, 15th and H Sts, N W
- WILKES-BARRE, PA—Power Engrg Corp, 517 Brooks Bldg

PRODUCTS: Heating and Ventilating Equipment including: Unit Heaters, Multiblade Fans, Pipe Coil Heaters, Buffalo Air Washers, Buffalo Unit Air Washers, Buffalo Unit Coolers, Drying Equipment, Mechanical Draft Fans, Air Preheaters, Exhaust Fans, Blowers, Dust Collectors, Disc Fans, Spray Nozzles.

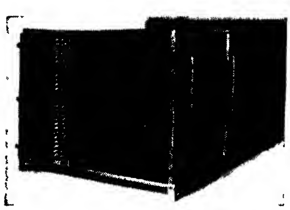
Buffalo Ventilating Fans

No matter what type of fan your work calls for, there is a Buffalo Fan of that type, of the right size, quiet, efficient, practical Write for our fan catalogs

Breezo Ventilating Fans

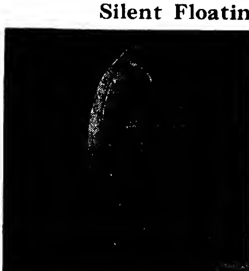
Breezo fans provide efficient, inexpensive ventilation on any job where they may be used to exhaust into the open. Made in sizes from 8 in. to 36 in. in diameter

Buffalo Comfort Conditioning Cabinets



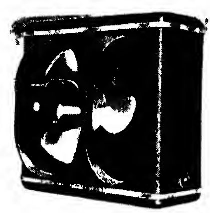
—for cooling and heating Heating coils are two-row copper fin type capable of heating from 70 F to 135 F with 2 lbs steam Cooling capacities—from 3 tons up.

"Limit-Load" Conoidal Fans with Silent Floating Base



The Buffalo "Limit-Load" high-efficiency, non-overloading ventilating fan mounted on the silent, floating fan base eliminates all motor and fan vibration.

Buffalo Unit Heaters



Breezo-Fin Steam Unit

Buffalo Unit Heaters are built in a wide range of sizes and types for the efficient and economical handling of any heating problem Operate with either low or high pressure steam We also offer a wide selection of Gas Unit Heaters that are fully automatic and equipped with every necessary safety device

Buffalo Unit Coolers

Quiet unit coolers for use with cold water, brine, methyl chloride or Freon. Compact, simple, inexpensive Available in both suspended and floor types

Buffalo Air Washers

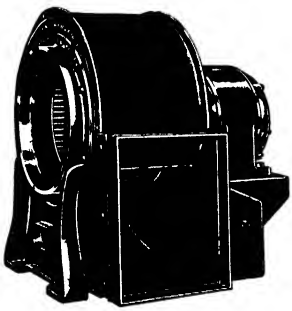
Buffalo Air Washers are in use in thousands of buildings, many for more than 30 years Bulletin 480 gives details

Champion Blower & Forge Co.

Manufacturers and Engineers

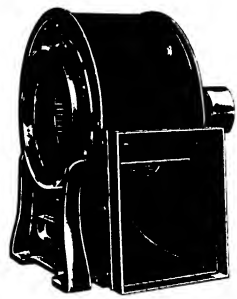
Plant and General Offices: Lancaster, Pa.

Manufacturers of Blowers, Ventilating Fans and Exhaust Fans for Air and Material; and Blast Gates



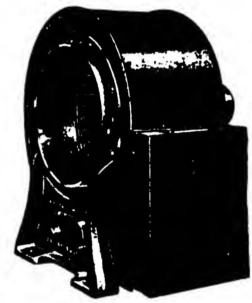
Type "SE"
Electric Driven Fans

These Fans are built for direct connection to Motor, with Adjustable Motor Base, quiet operation, large volumes, low outlet velocities.



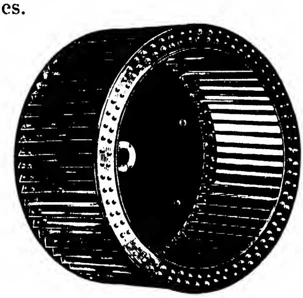
Type "S"
Belt Driven Fans

Built in all standard arrangements with Ball Bearings or Ring Oil Bearings, suitable for V Belt Drive.



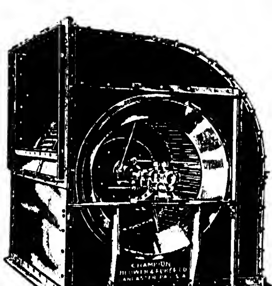
Type "BC"
Backward Curve Fans

Designed for higher speeds, furnished in any arrangement. The capacity characteristics show a flat horsepower curve.



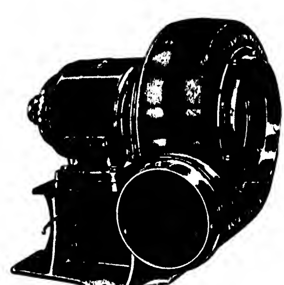
Type "C" Single and Double Width Forward Curve Blast Wheels

We are well equipped to build these Wheels in quantities for Oil Burner, and Stoker Manufacturers as well as Manufacturers of Air Conditioning and Ventilating Equipment.



Type "S" Double Width
Belt Driven Fans

Built in sizes up to 60 in. Wheel for Ventilating and Air Conditioning.



Type "CE" Electric Driven Fans

For use in Forced Draft and small ventilating work. Built in sizes up to and including 18 in. dia. Wheel.

DeBothezat Ventilating Equipment Division

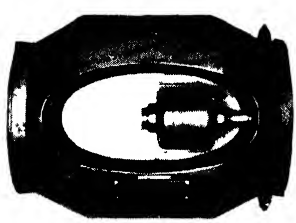
American Machine and Metals, Inc.

Executive and Sales Offices. 100 Sixth Ave., **New York, N. Y.**

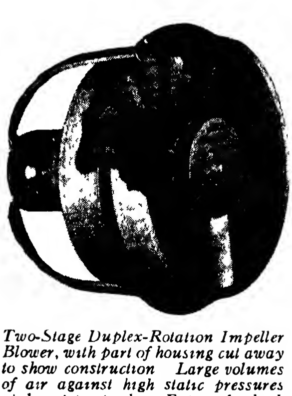
Factories: EAST MOLINE, ILL.

Branch Offices and Agents in All Principal Cities

Type H Disc Pressure Fan 24 in., ¾ hp, 1140 rpm, 5170 cfm against ½ in static pressure



Bifurcator Efficient unit for use where fumes of excessive temperature, corrosive or explosive character, are to be removed Motor outside air stream



Two-Stage Duplex-Rotation Impeller Blower, with part of housing cut away to show construction Large volumes of air against high static pressures at low tip speeds Extremely high efficiency over entire range of operation.

Fans and Blowers are guaranteed to have non-overloading power characteristics Complete operating safety under varying working conditions

Disc Pressure Fans (Direct and Belt Drive)—Three types H for high static pressures, HL, for moderate, and L for low—designed to meet all pressure-volume requirements efficiently Sizes from 8 in to 10 ft with a wide range of rpm "Selective Series" bulletin SS101 contains complete technical data and instructions for selecting from over 200 units the proper fan for your particular requirements

Giant Fans—all diameters from 5 ft to 10 ft May be supplied with directly connected motor, chain or V-Belt drive Widely used on Forced and Induced Draft Cooling Towers

Bifurcators—for use wherever gases are injurious to motors Can be placed horizontally or vertically in straight duct without introducing right angle turns in the duct No troublesome extended shafts, flexible couplings or bearings Motor located outside air-stream assuring safe, care-free operation

Duplex Rotation Impeller Blowers—Efficient units consisting of two disc pressure fans rotating in opposite senses, so designed that rotational losses are completely eliminated Direct drive by Duplex Rotation motor, ball bearings used throughout. Ideal for installation where large volumes of air are desired at low tip speeds

Vari-Speed Belt Driven Fans—25 per cent variation of speed provided by manual adjustment of motor pulley. Fan and motor are mounted as unit by mounting ring Quiet in operation, rugged in construction and economical in power consumption

Roof Ventilators—Assure positive ventilation under all atmospheric conditions Built in 14 to 48 in diameters (special sizes on specification) with capacity ranges from 925 to 52,300 cfm of free air Totally enclosed motor, ball bearing mounted Also furnished in Bifurcator Type, i e, motor isolated from air stream.

Attic Fans—DeBothezat attic fans adaptable to every type house Built in 18 in, 24 in., 30 in and 36 in diameter. Steel construction, insulated, with four-bladed fan rotating on rubber-mounted ball bearings.

Catalogs—Descriptive catalogs sent upon request

ILG Electric Ventilating Company

Propeller Fans, Blowers, Unit Heaters, Air Conditioning Equipment

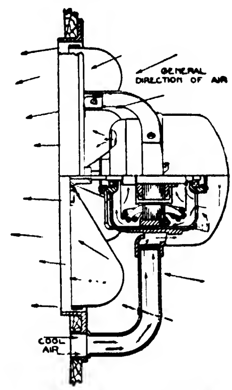
2880 North Crawford Avenue, **Chicago, Ill.**

Sales Representatives in all Principal Cities

ILG SELF-COOLED MOTOR PROPELLER FANS

Used everywhere for removal of foul air, fumes, heat, etc. Features include: the patented Ilg self-cooled motor, dynamically balanced, vibration-free, bucket-type wheel, and a strong one-nameplate responsibility. Motor, wheel and frame are all Ilg-built.

The self-cooled motor design combines the low operating cost of the open motor with the protection of the fully enclosed motor. The fan action draws clean air through vent pipe from outside; circulates it through motor (follow the arrows) and exhausts it. The Ilg motor stays clean, cool. Ilg fan sizes range from 12 to 72 in. in A.C. and D.C. Ratings are in accordance with the Standard Test Code of the A.S.H.V.E. and the National Association of Fan Manufacturers.

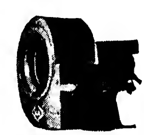


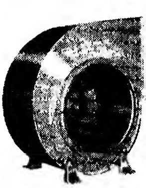
The Motor that Breathes

OTHER ILG PRODUCTS

Ilg Universal Multiblade Blowers—Type B Universal Blowers combine compactness, quietness and efficiency. Motor is recessed in side of blower, requiring no separate base. Multiblade wheel is mounted on motor shaft. There is no inlet bearing. Available also for belt drive. Capacities 1750 cfm to 70,000 cfm, single and double width.

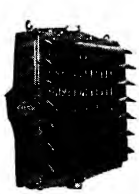
Ilg Type "B" Volume Blowers—A new design in small volume, low pressure blowers. Light weight, quiet running. Dynamically balanced multiblade wheel is mounted on motor shaft. Motor and steel housing are supported by cast-iron base. Universal discharge. Available in 11 capacities, 180 cfm to 2100 cfm.



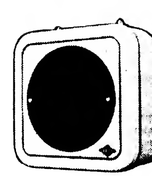


Ilg Type "BC" Universal Blower—Motor load remains constant over unusually large variation in air volume and static pressure. Type "BC" comes in direct connected drive (*left*); also belted drive with ball bearings (*right*). Blower wheel has backward curved blades riveted to side and back plates. Universal discharge. Available in 11 sizes.

Ilg Unit Heaters—Steam and Electric—Copper tube and fin construction and enclosed self-cooled motor. Ilg-built throughout. For steam or hot water. Tested with 500 lb hydrostatic pressure. Available in 67 capacities; also, Ilg electric unit heaters for all electric operation, available in 20 sizes.



Ilg Cooling and Air-Conditioning Units—Self-contained Ilg Spot Koolers in ½ ton cooling capacities with water or air-cooled compressors. Also central unit systems using floor cabinet or ceiling suspension units with remotely located compressor. For cooling, dehumidifying, and recirculating. Also, for heating and humidifying.

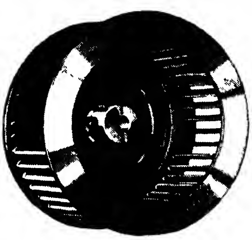


The Torrington Mfg. Co.

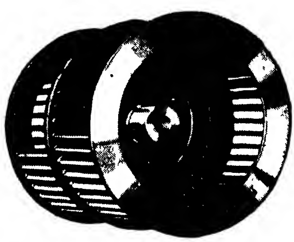
50 Franklin Street, **Torrington, Conn.**

Manufacturers of All-Aluminum Blower Wheels and Disc Propeller Type Fan Blades. Air impellers for every purpose.

AIRISTOCRAT Quiet Propeller Fan Blades | TORRINGTON BLOWER WHEELS | AUTOCRAT Fan Blades

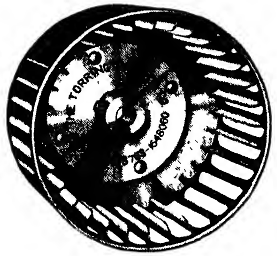


Single Inlet Blower Wheel



Double Inlet Blower Wheel
Pat 1,700,017

Special Design Blower Wheel



Cup Type Blower Wheel
Pats 1,513,763 and 1,648,060
(Produced under license from American Blower Co)

Torrington Aluminum Blower Wheels produce the smooth, quiet performance which is essential in modern heating and air conditioning units because the unique patented construction breaks up resonance and minimizes noise Made of aluminum, they resist corrosion and their light weight facilitates quick starting—saves power Every wheel is perfectly balanced by hand and given a running test

Bulletin lists 31 sizes of single inlet single width and 31 sizes of double inlet double width wheels, including guaranteed capacities for each Also gives detailed dimensions for all wheels and table of dimensions for housing scrolls. We do not manufacture housings

Sizes 3 in. to 15 in diameter in all standard widths

Special Design—for unit type air conditioners 2 sizes only Design is of double inlet type with a center spider plate to which hub is attached Blades, in addition to being riveted to inlet rings, are held rigidly in their proper position by the channel shaping of the inlet rings which closely fit the blade feet Made of aluminum or steel, cadmium plated

No ½ SD 5 in diameter x 4 ⅞ in width—36 blades

No ⅝ SD 5⅜ in. diameter x 6⅜ in. width—45 blades.

Bulletin gives performance curves, also details of wheel dimensions and table of dimensions for housing scrolls

Torrington Cup Type Wheels—The one piece cup construction is economical for production and the design is efficient and sturdy Ideal where maximum air delivery and minimum power is required Used for automobile heaters, windshield defrosters, small hair dryers, hand dryers, ice box and refrigerator circulators, window ventilators, exhausters, etc Made for either clockwise or counter clockwise rotation, of steel or aluminum, in the following sizes:

Dia.	Blade Width	Dia.	Blade Width
3 in	$1\frac{9}{16}$ in	6 in	⅞ in
3⅝ in	$1\frac{9}{16}$ in	6 in	$1\frac{11}{16}$ in
4½ in	$2\frac{7}{16}$ in	6 in	$2\frac{11}{16}$ in
4½ in	3 in	6 in	3¼ in
5 in	2¼ in	7½ in	$4\frac{1}{16}$ in
5 in	3 in.	9 in	4⅞ in

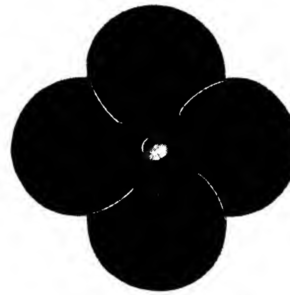
AIRISTOCRAT Quiet Propeller Fan Blades are beautiful, ornamental, ultra quiet, modern as the year 1939. Designed for free air applications (desk fans, air circulators, etc.) Perform well at moderate pressures. Used for window ventilators, unit type air conditioners, unit heaters, refrigeration devices, etc. Blades are hand set for alignment, statically balanced and packed in special containers for protection in transit. Clockwise rotation only. Aluminum alloy blades, steel center. Finishes as follows: 1. Plain. 2. Blades with no finish; spider and hub with cadmium plate or black lacquer. 3. All black lacquer, with or without center button. 4. Buff and lacquer blades, black lacquer spider and hub, with or without center button. Bulletin No. 123738 gives detailed dimensions and guaranteed performance curves recorded under **NEMA** code tests at various speeds.

De Luxe Model—The round, convex centerplate adds to the beauty of this ultra quiet blade which will improve the appearance of and help to sell your unit. Sizes 10 in., 12 in. and 16 in.

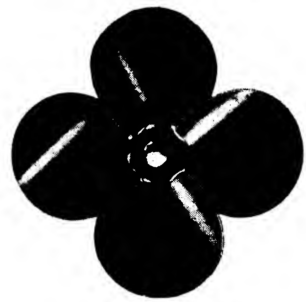
Standard Model—Same as DeLuxe Model but with blades mounted on a conventional type spider instead of round center disc. A sturdy and beautiful blade which has withstood extreme laboratory breakdown tests. Sizes 8 in., 10 in., 12 in., 14 in., 16 in. and 18 in. Priced lower than DeLuxe Models.

Air Circulator Model—The design of this blade is the result of two years of laboratory experiment to produce a better air circulator blade. At recommended speed these blades produce a high velocity air stream effecting deep penetration with unusual quietness. Sizes 20 in., 24 in. and 30 in. (Larger sizes ready soon).

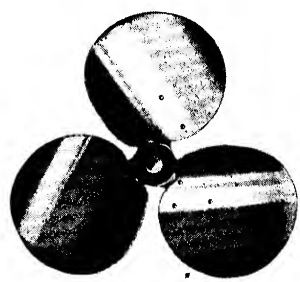
AUTOCRAT Fan Blades—For auto heaters, windshield defrosters, etc. Have been standard ever since these devices were first marketed. Made in sizes 3 in., 4 in., 4½ in., 5 in., 5¼ in., 5½ in., 6 in., 6½ in., all four blades, also 7 in. 5-blade, in one piece of cold rolled steel or aluminum with brass hubs, complete with set screw. ¼ in. bore is standard. Either clockwise or counter clockwise rotation (expressed when looking at air delivery side of fan). White nickel is standard finish for steel blades. Bulletin gives complete specifications and ratings.



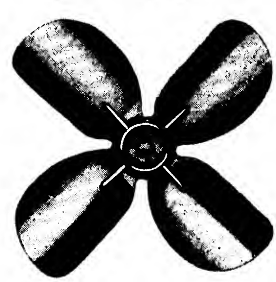
Airistocrat DeLuxe Model



Airistocrat Standard Model
Pats. 2,072,322 and 2,021,707



Airistocrat Air Circulator Model



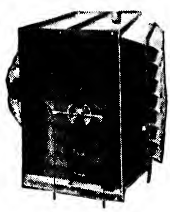
Autocrat Fan Blade

Every Torrington fan has resulted from scientific research and development under experienced engineers in a laboratory fully equipped for the study of aerodynamics. Other new Torrington models are under development. Get on our mailing list and you will be kept informed on important new fan developments.

Wing Utility Heaters

A lightweight suspended unit heater for delivering heated air in one general direction. Has the same powerful fan and rugged heating element as WING Featherweight Unit Heaters. This is the latest refinement of the original horizontal lightweight heater which was developed by WING. *Bulletin U-4.*

Wing Industrial Fog Eliminators

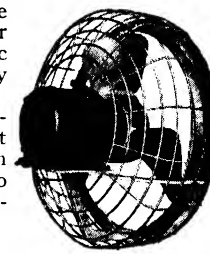


Eliminate fog, odor and fumes in dyeing, bleaching and finishing plants, creameries, pasteurizing, bottling, canning and packing plants, chemical works, paper mills, steel pickling plants, etc. No ducts are required. *Bulletin FE-12.*

Wing Featherfin Process Heating Units

For manufacturing processes such as drying, aging, etc., requiring the recirculation of the heated air. Motor or turbine located outside air current. *Bulletin P-2.*

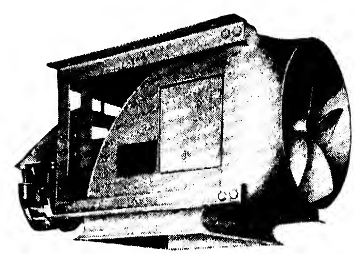
Wing-Scruplex Safety Ventilating Fans



A propeller type fan that will deliver air against static pressure, quietly and efficiently.

Moves the air forward in straight lines with minimum eddy. Capacities to 100,000 cfm. *Bulletin F-7.*

Wing-Scruplex Exhausters



For economically moving air wherever ducts are used. It combines the efficient WING-Scruplex Propeller Fan with a housing which places the motor entirely outside the air duct. Motor and drive remain cool and clean and are easily accessible.

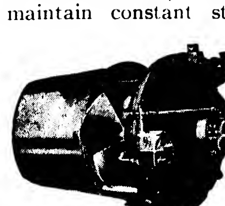
The powerful WING-Scruplex Fan delivers high air volume with low power consumption against any pressures for which duct systems should be designed. V-belt or direct drive.

Light, compact and easy to install. *Bulletin E-70.*

Wing System of Controlled Combustion

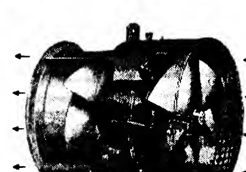
For low-pressure heating boilers. Increases capacity and permits use of lowest-cost fuel. Eliminates necessity of frequent firing, allowing intervals as great as 24 hours in some cases in zero weather. *Bulletin M-76.*

Wing Turbine-Driven Blowers



Applied to hand, stoker, oil or pulverized fuel fired boilers, increase boiler capacity, maintain constant steam pressure and permit complete combustion of low-cost fuels. The exhaust steam, free from oil, can be used for heating or processes. *Bulletin T-97A.*

Wing Type COM Blowers (High Static Pressures at Low Speeds)



Applied to high pressure boilers, produce high static pressures at low speed. Equipped with constant-speed motor and built-in damper, permitting variation of air delivery over a wide range with decreasing horsepower. Ideal for many ventilation applications. *Bulletin CO-3.*

B. F. Sturtevant Co.

Hyde Park, Boston, Mass.

Sturtevant
REG U S PAT OFF
Puts Air to Work

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DATA ON HEATING, VENTILATING, AIR CONDITIONING AND VACUUM CLEANING EQUIPMENT FOR ARCHITECTS, ENGINEERS, CONTRACTORS

The publications listed below have been prepared to aid the architect, engineer and contractor in the selection of proper equipment for industrial, public, and private buildings of all types and sizes If you do not have all of these publications in your file we will gladly send copies upon request

COOPERATION

Sturtevant Engineers, located at each of the offices listed are always ready to cooperate with architects, engineers and contractors in the selection of equipment suitable for any prospective installation

VENTILATING FANS

Catalog No	
271	Multivane Fans (Forwardly curved blade type)
414	Rexvane Fans (Radial blade type)
400	Direct-connected Fans and Blowers (Propeller Fans; Window Fans for kitchens and offices, Centrifugal Fans from 80 to 6460 c f m , Portable Gas - Engine - Driven Fans, Coal Burning Blowers, Forge Blowers, Dust Blowers)
422	Roofvane Ventilators.
435, 442	Silentvane Fans (Backwardly curved blade type)

MISCELLANEOUS HEATING AND VENTILATING EQUIPMENT

Catalog No	
291	Pneumatic Collecting and Conveying Systems
377	Unit Ventilators
395	Rexvane Speed Heaters (Floor type unit heaters)
396	Speed Heaters (Suspended Type Unit Heaters).
443	Design 8 Propeller Fan (Specially designed to operate against duct and wind resistance)

AIR CONDITIONING EQUIPMENT

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295	Air Washers
AC 101	Industrial Air Conditioning
378	Filticooler (Compact, high efficiency air washer For filtering, washing, humidifying, cooling, dehumidifying. Used principally for public buildings and factories)
398	Comfort Air Conditioning
401	Railway Air Conditioning
424	Fans and Air Washers for Theatres.
425	Air Conditioning Apparatus -- Fans and Accessories

VACUUM CLEANERS

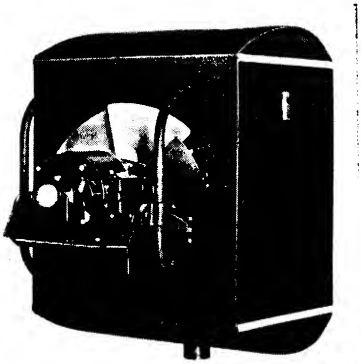
Catalog No	
368	Industrial Vacuum Cleaning Systems.
373	Vortex Furnace Cleaner.
397	Central Vacuum Cleaning Systems (Commercial Buildings).
413	Vortex Portable Vacuum Cleaners

Airtherm Manufacturing Company

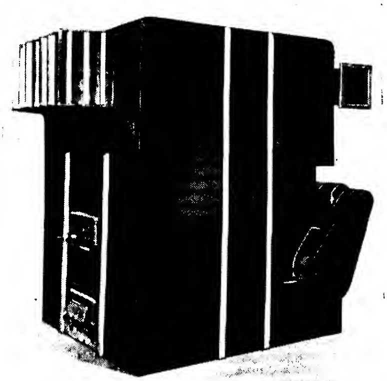
1474 South Vandeventer, St. Louis, Mo.

THE ENGINEERED LINE OF UNIT HEATERS

AIRTHERM IMPROVED UNIT HEATERS are backed by thirty years experience in unit heater construction, and are recognized for their many new, and exclusive features. Sound engineering principles, highest quality materials and workmanship, plus modern design and styling make Airtherm Units the logical choice. It is significant that most sales of Airtherm Heaters have been made to those concerns maintaining engineering staffs of their own, many of whom are engaged in heating and allied industries.

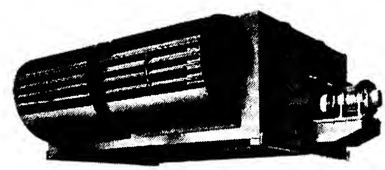


THE AIRBLANKET. Propeller Fan Type . . . with a patented method of air direction control to hold warm air in the heating zone.

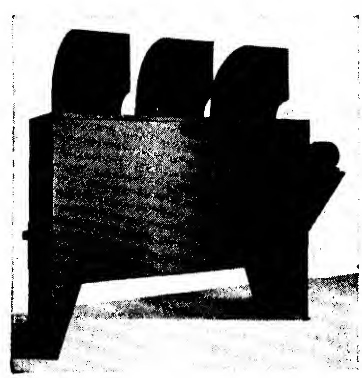


THE DIRECTHERM. Newly Designed Direct-Fired Unit for Gas, Oil or Coal (Hand or Stoker).

ENGINEERING SERVICE. The Airtherm Manufacturing Company Engineering Department and District Representatives are at all times available for consultation. At your request we will place experienced engineering aid at your disposal. Representatives in all principal cities.



THE AIRBLANKET. Centrifugal Fan Type . . . with the exclusive Airblanket Principle. Airblankets are designed so that a stream of cool air of high velocity is delivered just above a stream of warm air at less velocity. The cool air stream retains and distributes the warm air stream in the lower portions of a building, preventing unnecessary heating above the breathing level . . . it thus produces an artificial ceiling or blanket of cool air above the breathing line.



THE AIRHEATOR. The Airtherm blower fan type unit heater for floor or ceiling mounting.

THE AIRVECTOR. (Not illustrated). A propellor type fan unit for ceiling suspension or for mounting from the floor on recirculation stack.

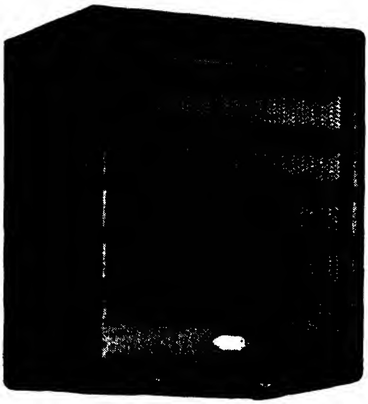
Fedders Manufacturing Co.

HEAT TRANSFER SPECIALISTS SINCE 1896

57 Tonawanda Street, **Buffalo, N. Y.**

Branches and Representatives in All Principal Cities

Unit Heaters, Air Conditioning Surface and Unit Conditioners, Unit Coolers, High Capacity Thermostatic Expansion and Constant Pressure Valves, Commercial and Household Refrigeration Equipment



Patents 1,970,105, 2,025,426

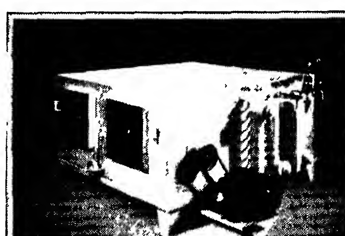
Fedders Series 4 Unit Heaters

New, exclusive Series 4 Double Headers for unusually free steam flow, improved Full-Floating Mountings, tubes with aerodynamically correct 3 to 1 streamline ratio and patented saddle fins prevent differential expansion stresses, improved mono-piece cabinet design with integral mouldings, those are some of the features of Fedders New Series 4 Unit Heaters Complete line with single, two and three speed standard and explosion proof motors Write for Bulletin 573

Model No EDR	3023 75	3141 100	3143 125	3181 150	3183 175	3211 200	3261 225	3273 250	3353 275	3311 300	3374 325	3372 350	3451 375
Model No EDR	3411 400	3421 450	3563 500	3564 550	3662 600	3653 650	3663 700	3772 800	3776 900	3862 1000	3873 1100	3972 1200	

FEDDERS ALL SEASON UNITS FOR AIR CONDITIONING

A complete line of All-Season Unit Air Conditioners complete with any combination of cooling and heating coils, humidifiers filters and blowers, ready to connect to refrigerant, cold water, steam or hot water supply. Cooling capacities from one ton up, heating capacities up to 2200 E D.R. Available for floor or suspended installations. Write for Bulletin AC-201.



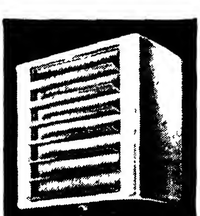
Fedders High Capacity Thermostatic Expansion and Constant Pressure Valves

Patents 1,974,631 1,987,948, 2,011,379

Model 37 Thermostatic Expansion Valves have 8, 12, 16 and 20 Tons capacity with Freon Model HCP38 Constant Pressure Valves up to 60,000 Btu/hr with 26 7 lb per sq in Pressure Drop, Freon.

FEDDERS AIR CONDITIONING SURFACE

Complete lines of all-copper cooling and heating coils, factory engineered and manifolded for correct distribution of cooling and heating medium. Engineered and cataloged on a package basis. Write for Bulletins.



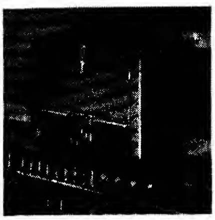
FEDDERS SERIES 73 UNIT COOLERS

Series 73 Unit Coolers are built for comfort cooling and commercial refrigeration. Handsome cabinets, non-rusting throughout, all-copper cooling element, complete with motor, fan and Fedders Thermostatic Expansion Valve Write for Bulletin.

McCord Radiator & Mfg. Co.

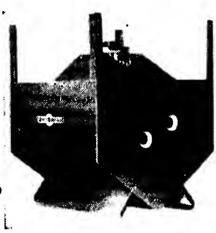
Heating and Air Conditioning Division

Detroit, Michigan



UNIT AIR CONDITIONER

A new development to heat, cool and condition stores, large offices and small factories. Cooling capacity 5 tons with 45 deg water; heat capacity 77,000 Btu per hour. Two 2 speed squirrel cage fans; silent vibrationless motor; washable spun glass filters. Smart modern design cabinet of chrome trimmed lacquer over bonderized steel.



VERTICAL TYPE UNIT HEATER

Developed to simplify the piping layout on high and wide buildings. Warm air taken from top of building is reheated and discharged downward into working area. Adjustable deflector outlets provide efficient distribution in any direction, requiring fewer units. Capacities from 115,000 to 250,000 Btu per hour. Totally enclosed ball-bearing motor.

UNIT HEATER

Advanced design permits rear entry of inlet and outlet, eliminating unsightly connections and requiring small head room. Wide range of output—Capacities from 20,000 to 500,000 Btu per hour. Quiet, totally enclosed motor. Beautifully designed welded steel case with rounded corners, stainless steel mouldings and baked crystal enamel finish.

CENTRIFUGAL BLOWER HEATER

For use wherever a large volume of air at low face velocity is required. Capacities 120,000 to 900,000 Btu per hour. Extremely quiet with large volume of air. Low speed, large diameter fan may be used with any type of duct construction or with hood as illustrated.

OTHER McCORD PRODUCTS

McCORD CENTRIFUGAL BLOWER UNIT HEATER—for use where ceiling suspension units are not desirable, particularly where ceilings are high. Floor or wall mountings. McCORDFIN BLAST HEATING AND COOLING UNITS—in convenient standard sizes. Solid copper and bronze, solder dipped, protected against corrosion and electrolysis. Greater efficiency due to perfect metal to metal contact and reduced air resistance. McCORD SUSPENSION TYPE AIR CONDITIONING CONVECTORS—modern attractive design, sturdy construction, efficient operation. McCORD CENTRAL SYSTEM AIR CONDITIONERS—complete units for installations requiring 3 to 24 tons cooling and 160,000 to 500,000 Btu per hour heating. McCord maintains one of the best equipped heating transfer laboratories in the industry. McCord's 34 years of experience and progressive engineering will help you maintain your reputation.

For engineering recommendations, catalogs, and prompt quotations, write McCord Radiator & Mfg. Co., Detroit, Michigan.

Modine Manufacturing Company

Heating and Air Conditioning Division

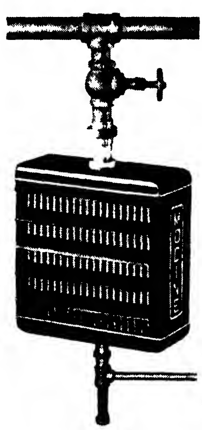
General Offices: 17th and Holburn Sts., **Racine, Wis.**

Factories at Racine, Wis. and La Porte, Ind.

Branches in all Principal Cities

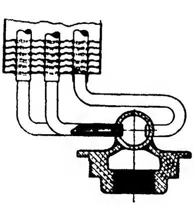
Complete information on the following products including engineering data and prices, can be secured by writing to the Modine General Offices at Racine, Wisconsin— or by communicating with nearest Modine representative.

MODINE UNIT HEATERS



Application—Not only successfully used for industrial, garage and similar space-heating applications, Modines are ideally suited also to showrooms, stores, oil stations, churches, school rooms, residential recreation rooms, etc. They have also effected important economies in the drying of paint, leather, enamels, paper, wood, etc, by speeding up and bettering the drying process.

Exclusive features that not only make the Modine a better unit structurally but assure effective, economical distribution of heat. (1) **Expansion Bend,** given each tube of the condenser before entering lower header, provides for free expansion, thus eliminating strain from being transferred to header tanks (2) **Velocity Generator** for redirecting and controlling the heated air stream, assures greatest possible throw consistent with comfortable heating. (3) **Direct Pipe Suspension** facilitates horizontal redirection of the heated air stream, permitting full 360 deg rotatability Also means easier installation at less cost—no brackets, pipe rods or straps being necessary. Write for Unit Heater **Catalog 138.**



Unit Heaters—Capacities and Dimensions

(In Inches)

Model No	Over-all Height	Width	Depth Less Motor	E D R	C F.M	Motor R P.M
76	10½	9¾	5¾	76	187	1550
126	16	13	8	126	456	1590
152	18	15	8	152	540	1590
181	18	15	8	181	770	1140
204	18	15	8	204	735	1140
238	18	15	8	238	731	1140
275	19½	18	9	275	1052	1140
352	22½	18	9	352	1425	1140
440	22½	18	9	440	1320	1140
542	23	22	9	542	1710	1125
620	26½	22	9	620	2140	1120
710	26½	22	9	710	2230	1120
903	27⅞	26	10½	903	3000	1125
1163	30⅜	26	10½	1163	4050	1110
1300	34⅜	26⅜	8	1300	5010	1125
1545	30⅛	52½	11	1545	5400	1125
2015	30 5/32	56½	11	2015	6540	1110

All above models are available with variable speed motors Units for hot water application also available

MODINE COPPER CONVECTORS

(Deluxe Type)

The new Modine Deluxe Convector furnished in four types, Concealed, Recessed, Floor and Wall Cabinet, is designed for use on steam, vapor, vacuum or hot water systems Styled for beauty of line and proportion, the enclosure is adapted to easy color application. Interchangeable, die-cast grille segments in four patterns make it possible to design a grille which will harmonize with any interior Grilles also available in plated finishes —**Catalog 238.**

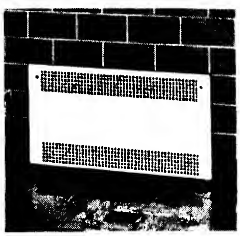
(Standard Type)

The popular copper radiators for commercial and public buildings, low cost houses, etc —wherever the benefits of copper convector heating are desired. Attractive

enclosures with removable fronts. High capacity copper heating units. Enclosures available in three types of Recessed and Floor and Wall Cabinet types. **Catalog 238-A.**

INSTITUTIONAL TYPE

Designed for use where special construction features, common to institutional heating, are specified. This line incorporates many features which have heretofore been considered "expensive specials." Available in two types of Recessed and in Wall Cabinet. **Cat. 238-A.**

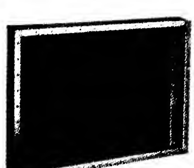


MODINE BLAST HEATERS

Made in over 250 sizes, types and capacities to meet the specific demands for heat transfer service. Outstanding features are: (1) Expansion Bend. (2) All steam carrying passages are cylindrical for greatest possible strength. (3) From inlet to outlet condenser is of copper or copper alloy. (4) Copper fins are bonded metallically to tubes. **Cat. 338.**

MODINE COOLING COILS

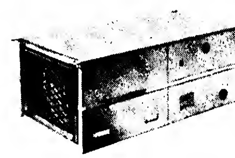
For use in central system cooling and air conditioning plants, Modine Cooling Coils, Cold Water Type, are installed with a blower fan and duct work. Adaptable where cold water or non-corrosive brine is used as the cooling medium.—**Catalog 538.**



UNIT COOLERS (Propellor)

This unit is intended for cooling stores and offices in the summer—and heating them in the winter. Cold water is the cooling medium. Inexpensive and easy to install. No ducts or change in building construction required **Bulletin 438.**

UNIT COOLERS (Blower)



For stores and offices. This unit cools, cleans, dehumidifies and circulates the air. Equipped with powerful, yet quiet blower, extra deep cooling coils and large-area air filters. May be installed with or without duct work. Choice of cold water or Freon cooling coils. **Bulletin 438-A.**

AIR CONDITIONER (Apartment House Type)

A compact unit performing every function of complete winter and summer air conditioning—for apartments, hotel suites, residences, offices, and shops. Its compactness allows installation in a closet above shelving or in a hall above a false ceiling. Uses steam or hot water for heating; cold water or Freon for cooling. Two sizes. **Bulletin 638-B.**

AIR CONDITIONER (Large Central Type)

For residential and commercial year-'round air conditioning—may be used in straight air conditioning or split systems. Uses steam or hot water for heating and cold water or Freon for cooling. **Catalog 638-A.**

The Unit Heater and Cooler Co.

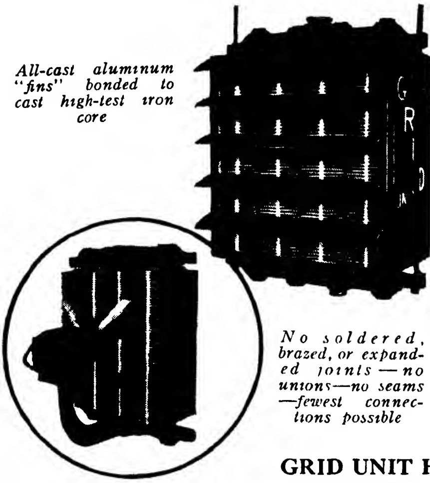
Wausau, Wisconsin

Offices in Principal Cities

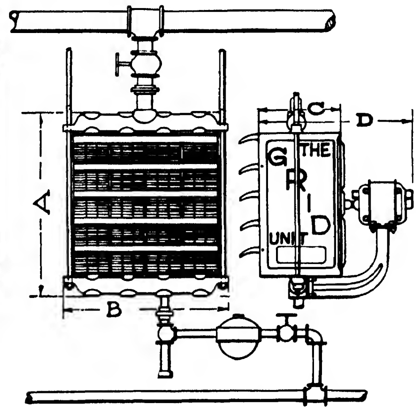
MANUFACTURERS OF THE GRID UNIT

(PATENTED)

All-cast aluminum "fins" bonded to cast high-test iron core



No soldered, brazed, or expanded joints—no unions—no seams—fewest connections possible



Typical Installation of Grid Unit Heater

GRID UNIT HEATER DATA

Model No.	Dimensions Inches				Face Area Sq Ft	Motor		Vol. at Fan	Capacities 5 Lb Steam 60° Air		Approx. Shipping Weight	Pipe Sizes	
	A	B	C	D*		Hp	Rpm		Btu	Final Temp		Supply	Return
1000	16	12¾	9⅜	16	0 85	1/20	1700	578	29,400	107	90	1¼"	1¼"
1200	18	14½	11¼	17½	1 04	1/20	1700	711	46,000	119	120	1¼"	1¼"
515	23½	18	11½	20	1 67	1/10	1750	1290	59,600	102	180	1½"	1¼"
1500	23½	18	11½	20	1 67	1/10	1750	1450	77,500	109	210	1½"	1¼"
1520	27	18	11½	20	2 2	1/10	1750	1700	104,000	113	250	1½"	1¼"
520	27	23¼	11½	21½	2 8	1/6	1150	2500	102,300	97	280	2"	1¼"
2000	27	23¼	11½	21½	2 8	1/6	1150	2500	148,000	114	320	2"	1¼"
2025	32	23¼	11½	21½	3 6	1/6	1150	2875	177,000	115	370	2"	1¼"
525	32	28½	11½	28	4 5	1/2	1150	4200	166,400	94	390	2"	1¼"
2504	32	28½	11½	28	4 5	1/4	1150	3200	210,000	118	400	2"	1¼"
2500	32	28½	11½	28	4 5	1/2	1150	4200	225,000	108	440	2"	1¼"
2530	36	28½	11½	28	5 3	1/2	1150	4650	282,000	115	500	2"	1¼"
530	38	33	13¾	29	6 5	1/2	1150	5300	260,500	105	600	2½"	1¼"
3000	38	33	13¾	29	6 5	1/2	850	6350	341,000	109	690	2½"	1¼"
3000	38	33	13¾	29	6 5	1½	1150	8100	394,000	104	725	2½"	1¼"

*Varies with type of motor.

GRID UNIT HEATERS ARE NOT AFFECTED BY ELECTROLYTIC ACTION

No leaks—no breakdowns.
Low maintenance expense.
More air changes per hour.
Positive "directed" heat.

Lower outlet temperatures.
Larger air volume.
Reduced fuel cost.
Applicable to either low or high steam pressure lines.

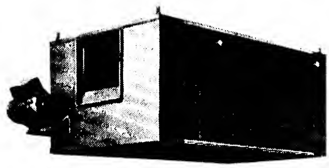
Send for Bulletin on Units not listed above.

YOUNG RADIATOR Company

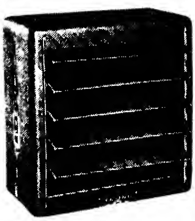
Offices in all Principal Cities **Racine, Wis.**

Write For Literature

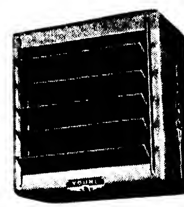
Engineering Information At Request



Air Conditioning Unit



Model SH Unit Heater

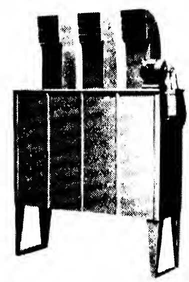


Unit Cooler

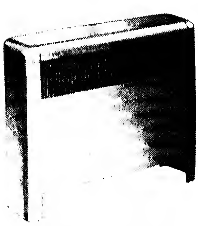
Air Conditioning Units—are made in five physical sizes for home and industrial installations. Designed to do a complete air-conditioning job, heating, cooling, humidifying, and dehumidifying.

Unit Heaters—Model SH and TH. A complete line of single and twin fan suspended units, 28 sizes ranging in capacities from 18,000 to 658,000 Btu per hr.

Unit Coolers—A complete line of suspended units, 9 models varying in capacities from .39 to 2 tons. Designed for use with water brine, or any common refrigerant.



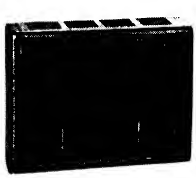
Model FH Unit Heater



STREAMAIRE Convector

STREAMAIRE Convectors—are available in six distinct types of cabinet enclossures—free standing, wall hung, partially recessed, fully recessed, bathroom cabinets, and plastered-in enclosures. Designed to operate on one pipe steam, two pipe steam, vapor, vacuum and gravity hot water heating systems.

"FH" Unit Heaters ten models ranging in capacities from 120,000 to 800,000 Btu per hr. These units are equipped with two, three, and four blower type fans, driven by single or variable speed motors.



Type FC Heating Units

Type "W" Cooling Coil

Type "K" Cooling Coil

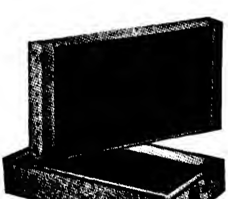
"FC" Heating Units—combine the appearance of a STREAMAIRE convector with the higher capacity of YOUNG Unit Heaters. Available in four physical sizes, equipped with single and two speed motors.

Cooling Coils—Type "W" with continuous tubes, type "K" with removable headers for use with water or brine.

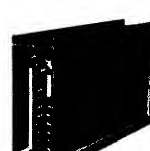
Evaporators—Designed for mechanical refrigeration systems using Freon or methyl chloride.

Commercial Units—Designed for use in connection with heating, cooling, and air-conditioning units where a compact, efficient, heating surface is desired.

Blast Units—as encased heating surface for use in connection with forced air, heating, and cooling systems.



Blast Units—Commercial Units



Evaporators

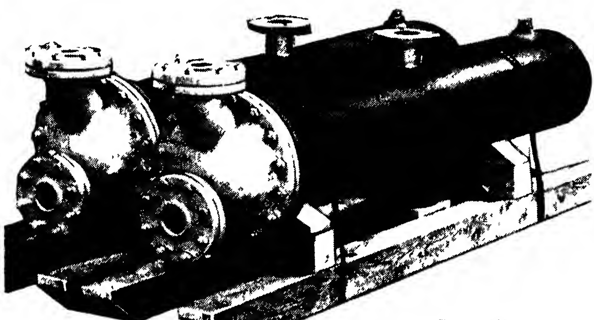
The National Pipe Bending Company

Incorporated 1883

104 River Street, **New Haven, Conn.**

Instantaneous Water Heaters
Caustic Liquor Heaters
Feed Water Heaters
Fuel Oil Heaters
Storage Heaters
Head-Exchangers
Coils, Pipe, Tube
Welded Heaters

NATIONAL HOT WATER HEATERS



Type NI Instantaneous Heater

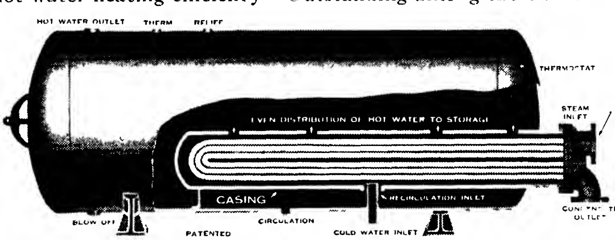
Instantaneous Water Tube Type

Heats water as it flows, either continuously or intermittently Built for steam pressures up to 150 lb per sq in U-Bend copper tubes expanded into steel tube plate and enclosed in steel shell. Made in sizes from 210 to 30,000 gal of hot water per hr at temperatures of from 40 to 180 F.

NATIONAL SUPER-SERVICE STORAGE HEATERS

This heater embodies exclusive, new features of design and construction which make possible the utmost in hot water heating efficiency Outstanding among these features:

Provides for intimate contact of all water with all heating surfaces at all times Entering water passes directly into bottom of casing completely enclosing heating element No short circuiting



Note Heating Element is Completely Enclosed by Casing

NATIONAL FEED WATER HEATERS COIL TYPE

Type BW. Coils are internal manifold type encased in cast iron or steel shell with ends brazed into bronze manifolds Coils rigidly secured by copper straps to bar-iron supports to prevent vibration, abrasion of tubes and electrolytic action. Heads having steam and drip connections are bolted to shell, either cast iron or flared and dished steel.

FACILITIES FOR COILS AND BENDS

Designs, specifications and estimates submitted for any requirements All "National" Coils are continuous, without couplings, or screwed fittings, unless specified. Iron and steel coils have electrically welded joints, brass and copper, brazed joints

tubing, units designed with steel headers on opposite ends, the ends of the "S" bends being connected thereto by compression nuts, the bends taking care of the expansion and contraction of the tubing. Recommended where close control is desired

Headers--Pressed steel

Tubing—1 in O D Copper, admiralty or aluminum

Casings—Copper, aluminum or galvanized iron.

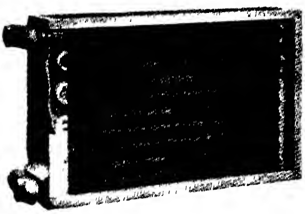


Fig 4

High Pressure Aerofin (Fig 4) is of continuous tube design, being recommended where extremely high pressures of steam are used

Headers—Pressed steel

Tubing—1 in O D. Copper, aluminum or admiralty

Casings Copper, aluminum or galvanized iron

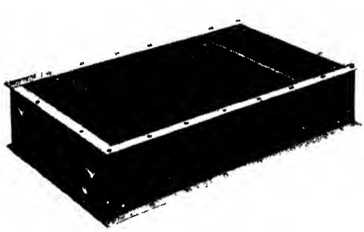


Fig 5

Booster Aerofin (Fig. 5) is of the continuous tube design, recommended where small volumes of air are used, or to raise the air temperatures in branch ducts, etc.

Headers— Cast iron

Tubing—5/8 in. O D. Copper or aluminum.

Casings--Copper, aluminum or galvanized iron

Aerofin Encased Booster Units: (Fig 5) For horizontal or vertical air flow Six sizes, 150 to 1624 cfm

For either horizontal or vertical air flow

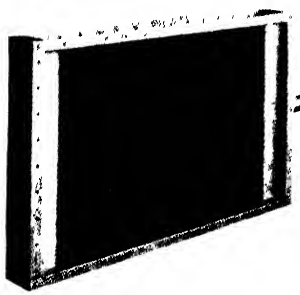


Fig 6

Narrow Width Aerofin: (Fig 6) recommended for water cooling or for flooded Freon systems Made in straight tubes only with headers on opposite ends, joints between headers and tubing being brazed Construction similar to Flexitube AEROFIN

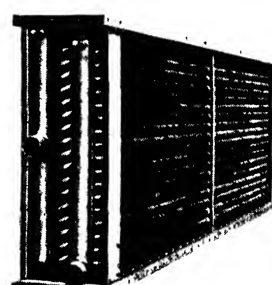


Fig 7

Aerofin Continuous Tube Water Coils (Fig 7) are designed for air cooling by circulating cold water through the AEROFIN and air over extended fin surface Made for either horizontal or vertical air flow.

Tubes and fins are copper, completely tinned with permanent metallic bond

between fin and tubes. Headers are made of one-piece cast bronze and casings of heavy galvanized iron or copper.

Units tested to 1000 lb hydrostatic pressure.

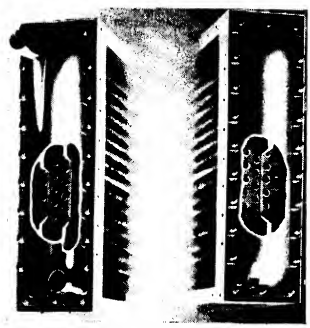


Fig. 8

Aerofin Cleanable Tube Units (Fig. 8) for cooling only and all made with headers removable to permit cleaning out tubes. Recommended for use where sediment or scale forming chemicals are present in the cooling water.

Headers—Cast iron.
Tubing—Copper or admiralty.
Casings—Copper or galvanized iron.

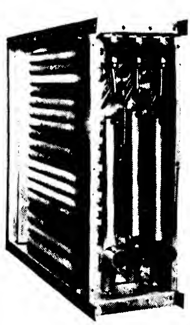


Fig. 9
End plate removed showing distributing and suction headers.

Aerofin Direct Expansion Units: (Fig. 9) Row Control Type—Recommended for use where cutting on or off rows of tubes in direction of air flow is desired. Suitable for use with Freon or Methyl-Chloride.

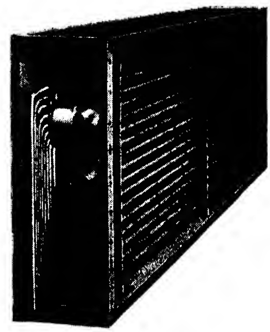


Fig. 10

Aerofin Direct Expansion Units: (Fig. 10) Centrifugal Header Type—Recommended where control of rows in direction of air flow is not required.

Advantages: Weighs but 9 to 16 per cent of same equivalent cast iron surface and occupies one-third of the space. Eliminates expensive foundations and building re-inforcement. Can be suspended from roof beams or trusses if necessary.

AEROFIN Sizes

Flexitube: 13 standard lengths, three widths, one and two rows deep.

Narrow: same as Flexitube.

Universal: 17 standard lengths, two widths, one and two rows deep.

ContinuousTube: 13standardlengths, three widths, 2-3-4-5 and 6 rows deep.

Cleanable Tube: 17 standard lengths, one width, 2 and 4 rows deep.

Direct Expansion: Row Control—11 standard lengths, 3 widths, 1-2-3 rows deep. Face Control—11 standard lengths, 3 widths, 2-3-4-5-6 rows deep. Centrifugal Header—11 standard lengths, three widths, 2-3-4-5-6 rows deep.

Steel Supporting Legs: 18 in. and 24 in. high. Punched same bolt hole centers as standard casings. Quickly attached. No other foundation required.

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus. List upon request.

Write Syracuse for Heating Bulletin G-32; Direct Expansion Bulletin DE-34 on refrigeration type units; Continuous Tube Bulletin C. T. 34 for Water Cooling Coils; or phamplet on Cleanable Type AEROFIN for cooling.

The G & O Manufacturing Company

138 Winchester Avenue **New Haven, Connecticut**

G&O

SQUARE FIN TUBING
STRAIGHT LENGTHS—U-BENDS—CONTINUOUS COILS

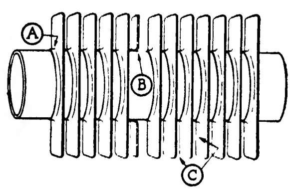
RADIATING ELEMENTS FOR ALL HEAT TRANSFER PURPOSES

THE use of INDIVIDUAL fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface.

Fins of any size or shape may be obtained giving any desired proportion of primary and secondary surface.

A square fin has about 30 per cent greater surface than a round fin of a diameter equal to one side of the square.

Individual fins permit of any fin spacing, also, of using fins in groups at intervals along tubes.

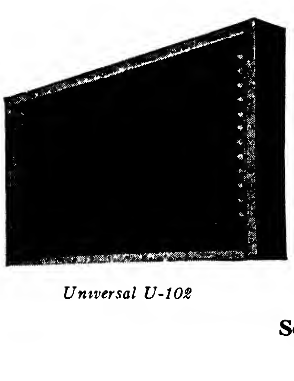


A—Generous Fin Collar provides large contact area between Tube and Fin.
B—Tube expanded against Fin Collar; insures mechanically tight joint, made permanent by bond of high temperature alloy—perfect thermal contact.
C—Free air-flow passages, non-clogging.

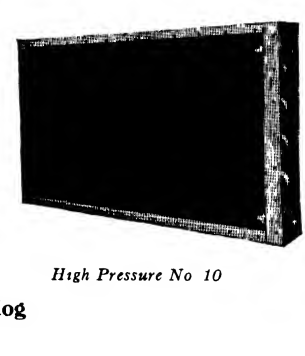
STANDARD SIZES

OD of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
3/8″	7/8″ sq.	6	0 80 sq ft
3/8″	7/8″ r'd	6	0 60 sq ft
3/4″	1 1/2″ r'd	6	1 55 sq ft
3/4″	1 5/8″ sq	6	2 40 sq ft
1″	2 1/8″ sq	6	4 00 sq ft

G & O Finned Radiation Coils for all industrial applications are available in a wide range of sizes for high and low pressure operation.



Universal U-102



High Pressure No 10

Send for Catalog

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Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

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PRODUCTS AND SERVICES—

Complete Service on materials to Specification on Power Plant Piping, Industrial Piping, and Industrial Heating Systems; Prefabricated Piping including Pipe Cutting and Threading, Pipe Bends, Welded Headers, Welded and Welding Fittings, Lap Joints and the Grinnell line of products for Super Power.

Grinnell Equiflo Valves for forced hot water heating systems; Grinnell Adjustable Pipe Hangers and Supports; Grinnell Cast Iron and Malleable Iron Pipe Fittings; Grinnell Malleable Iron Unions; Grinnell Welding Fittings; Grinnell Thermoliers (Unit Heaters); Grinnell Thermofin (Convectors); Thermoflex Traps and Heating Specialties.

Also Humidifying Systems; Constant Level Size Circulating Systems; Piping for acids and other special materials.

Malleable Iron, Brass, Bronze and other Castings; Brass, Cast Iron, Wrought Iron and Steel Pipe; Seamless Steel Tubing in Iron Pipe Sizes.

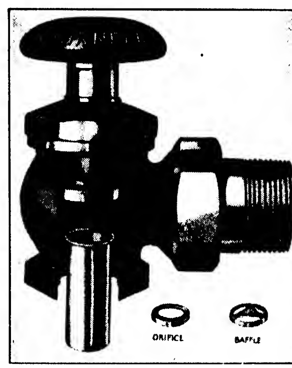
Valves: Check, Globe, Pressure Reducing and Regulating, Quick Opening, Safety and Y.

Automatic Sprinkler Systems; Stand Pipes; Underground Supply Mains; Hydrants; Fire Pumps; Pressure and Gravity Tanks.

Grinnell "Junior" Automatic Sprinkler Systems for Basements and other hazardous areas of Dwellings, Small Apartment Buildings, Schools, Churches, Stores, etc.

Grinnell Equiflo Valves

For Forced Hot Water Heating



Equiflo Valve

The designing of forced circulation hot water heating systems is so simplified by the Grinnell Equiflo Valve that they can be laid out and installed as easily as vapor or steam systems. This valve consists of a regular type packless radiator valve with a cartridge or tube made up of a series of orifices and baffles capable of setting up any required frictional resistance. This method of establishing any desired resistance does away with elaborate calculation of pipe sizes. Grinnell guarantees perfectly balanced circulation to each and every radiator where these valves are installed throughout the system.

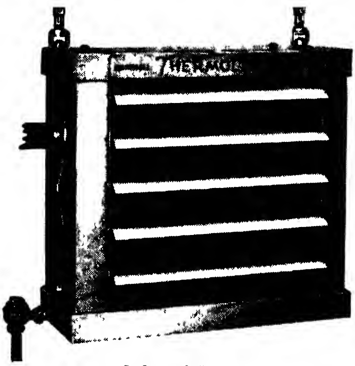
Equiflo Data Book sent to interested parties.

For Data on Thermoflex Traps and Heating Specialties, see page 1087

THERMOLIER

Patented

THE GRINNELL UNIT HEATER



Industrial Type

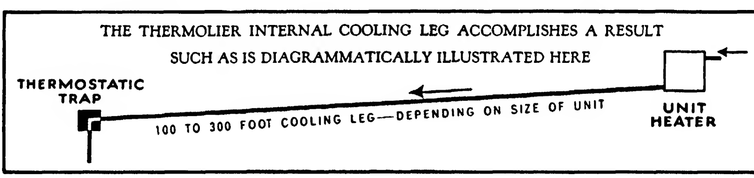
De Luxe, Industrial and Factory Types—125 Lb W.S.P.

Thermolier is a ruggedly built unit heater whose efficiency and dependability have been proved by actual performance in field service. Thousands of them are installed in industrial buildings and commercial structures of all types of occupancy.

Thermolier has 14 points of superiority, the most outstanding of which is the internal cooling leg built right into the unit, an exclusive Thermolier feature. See drawing below.

Radiation is from brass-finned seamless copper U-tubes rolled into a cast-iron tube sheet. No solder is used for strengthening joints and there are no flat horizontal surfaces to catch dirt.

Units may be controlled manually or automatically, singly or in groups. Installation and piping are extremely simple and inexpensive, hence the unit may be moved from one



location to another at small cost if found desirable on account of changes in building or occupancy. The complete line includes 22 Models in DeLuxe Type, and 30 Models each in Industrial and Factory Types.

Thermoliers provide maximum distribution of heat without objectionable drafts.

Specifications

Fan—Grinnell special of rugged construction **Motor**—heavy duty, oversize, enclosed, moisture-proof. **Housing**—Art Metal Slate gray finish with chromium trim on DeLuxe Types Copper on Industrial Type with rubbed lacquer finish; steel on Factory Type finished in gray duco. **Frame**—Heavy pressed steel, providing rugged support for motor and fan. **Special Features**—Adjustable swivel hanger rod couplings, louvers rigid, but easily adjustable: integral cooling leg insuring perfect drainage through one thermostatic trap for pressures up to 25 lb

For pressures not exceeding 125 lb, a thermostatic trap of proper construction can be used and should be attached directly to the unit.

CAPACITIES

60 F Entering Air Temperature—2 lbs Steam Pressure

Model Nos.	Btu per Hour	Model Nos.	Btu per Hour	Model Nos	Btu per Hour
20	35,000	50	90,700	90	189,200
20L	26,900	50L	67,100	90L	151,600
25	40,500	60	104,800	100	234,000
25L	30,900	60L	77,700	100L	196,000
30	47,800	65	129 500	110	259,000
30L	35,200	65L	110,100	110L	211,600
40	69,400	70	142,000	140	320,000
40L	53,300	70L	117,000	140L	271,000
45	81,200	80	164,600	180	368,000
45L	62,600	80L	139,300	180L	294,000

Data Book covering other pressures and temperatures, dimensions and complete installation information on application. Address GRINNELL COMPANY, INC., 277 West Exchange Street, Providence, R. I.

GRINNELL ADJUSTABLE PIPE HANGERS AND SUPPORTS

One of the chief advantages of Grinnell Adjustable Hangers is that they permit adjustment of pipe lines after installation, thus obviating the necessity of turnbuckles or the removal of hangers. Their time and trouble-saving qualities during installation are equally exceptional. Below are shown a few Grinnell Hangers and Supports of particular interest to heating engineers. Send for Hanger Catalogue showing complete line.

Adjustable Swivel Rings (Patented)

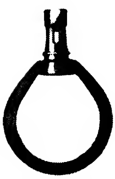


Fig. No. 101
Solid Ring

These Malleable Iron Adjustable Swivel Rings can be used with Coach Screw Rod or Machine Threaded Rod in connection with practically any type of Ceiling Flange, Expansion Case, Insert, etc.

Adjustment of at least 1½ in. is secured by turning Swivel Shank. Swivel Shank automatically locks, preventing loosening due to vibration in the pipe line.

The Split Ring permits adjustment either before or after Ring is closed. A wedge type pin is loosely but inseparably cast into the hinged section for fastening this section after pipe is in place.

Fig. No. 104
Split Ring

Adjustable Swivel Pipe Rolls (Patented)

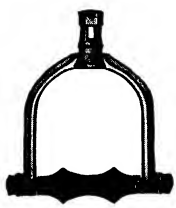


Fig. No. 174
Swivel Pipe Roll

An adjustable type of pipe roll using a single hanger rod. Swivel Shank allows vertical adjustment and automatically locks, preventing loosening from vibration.

CB-Universal Concrete Inserts (Patented)

Made of air furnace malleable iron, in one body size, to take a special removable nut, tapped for ⅜ in., ½ in., ⅝ in. or ¾ in. rod as required. Nuts automatically lock by means of V-type teeth on both insert and nuts.

Fig. No. 282
CB-Universal Insert

GRINNELL WELDING FITTINGS

90° Elbow, Long Turn

Grinnell Welding Fittings are made from Seamless Steel Pipe or tubing and possess the same physical characteristics as standard, extra strong and o.d. steel pipe or seamless steel pipe of comparable size. They can be used under the same conditions, pressures and temperatures as the pipe itself.

Welding faces for all plain circumferential Butt Welds are scarfed or beveled to the regulation 45 deg. angle with 1/16 in. square end on inside of fitting. Angles of bevel other than 45 deg. can be furnished on special orders.

Welding Outlet

Welding Tee

Lap Flanged Welding Neck

Threaded Outlet

The Herman Nelson Corporation

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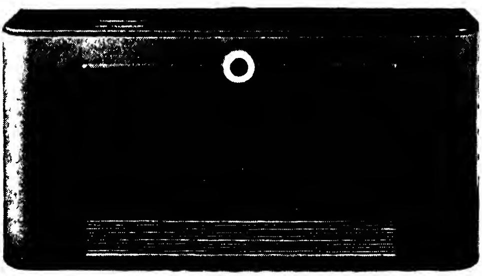
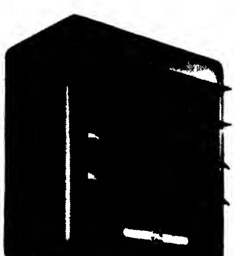
New Herman Nelson Air Conditioner For Schools



Herman Nelson Air Conditioners for maintaining comfortable and healthful conditions in school classrooms consist of heating element, filter, cabinet, air intake, fan and motor unit and two sets of dampers —all designed and arranged to produce efficient heating and ventilating at low cost One Herman Nelson Air Conditioner is all that is normally required in a classroom. Other Herman Nelson equipment especially designed for large rooms, auditoriums and gymnasiums is available

Herman Nelson hiJet Heaters



Propeller-Fan Type

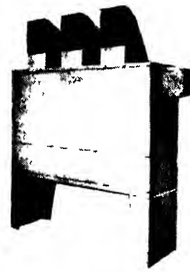
The Herman Nelson Corporation manufactures a complete line of hiJet Unit Heaters in two types—propeller fan and blower fan, for use with steam or hot water systems

Propeller-Fan Type hiJet Heater

Available in 18 different sizes and a complete range of capacities for full, medium or slow speed operation

Blower-Fan Type hiJet Heater

Manufactured in 16 models and a complete range of capacities High or low capacity heating element available Direct-connected motor or V-belt drive



Blower-Fan Type

Herman Nelson Automatic Heat and Air Conditioning

Oil Burning Air Conditioning Furnace
Gas Burning Air Conditioning Furnace
Conversion Oil Burner
Oil Burning Boiler
Automatic Stoker
Coal Burning Air Conditioning Furnace
Self-Contained Cooling Unit
Year Around Air Conditioning Unit

CATALOGS AVAILABLE on The New Herman Nelson Air Conditioner for Schools Herman Nelson hiJet Heaters . . Herman Nelson Automatic Heat and Air Conditioning. Write to THE HERMAN NELSON CORPORATION, MOLINE, ILLINOIS.

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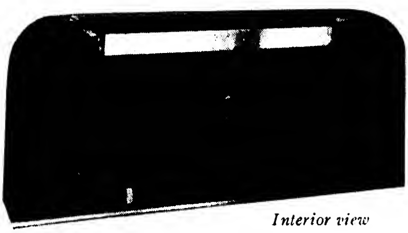
NESBITT HEATING SURFACE with Steam-distributing Tubes and
NESBITT SERIES H HEATING SURFACE, sold by leading manufacturers of fan-system apparatus;
WEBSTER-NESBITT UNIT HEATERS (for details see page 1101), distributed in the U.S.A. by Warren Webster & Company.



Interior view

The Nesbitt Syncretizer—Series 400

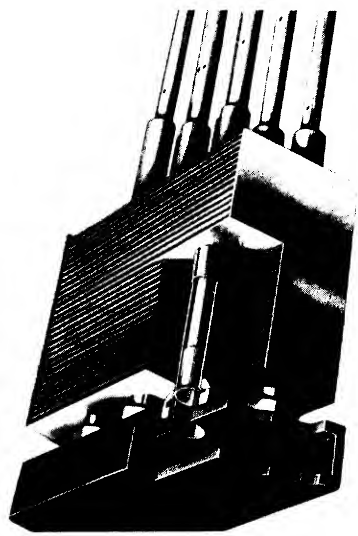
The last word in heating and ventilating units for schoolrooms, offices, etc., where the continuous introduction of outdoor air is desired. For engineering data, get Publication No. 225; for "The Story of Syncretized Air," Publication No. 231.

Nesbitt Series B Thermovent

For heating and ventilating auditoriums, gymnasiums, assembly halls, and similar gathering places. Publication No. 227.

Nesbitt Series H Heating Surface ALL COPPER

A lightweight, enduring, highly efficient blast-coil heating surface of tube-and-fin construction, made entirely of copper and copper alloy, designed for use with steam pressures up to 200 lb gauge. Large headers of seamless copper tubing with collars extruded from the header body, providing large areas to which the condensing tubes are silver brazed. Seven types, each in eight fin widths and up to sixteen finned lengths—a total of 784 sizes from which to select. Sold by leading manufacturers of fan-system apparatus (list upon request). Send for Publication No. 232 for complete engineering data.



Nesbitt Heating Surface with Steam-distributing Tubes

Copper tube-and-fin heat transfer surface perfectly adapted to close, continuous automatic control with modulating steam valves. Revolutionary steam-distributing tubes within the condensing tubes carry the steam equally to all parts of the section, thus assuring UNIFORM discharge temperatures, even under throttled steam supply; eliminating temperature stratification; preventing tube freezing without preheaters; giving ideal system results. The illustration above shows a cut-away view of Nesbitt Surface.

Strong and lasting, built to withstand test pressures up to 200 lb gauge. Available in cased or uncased units of many sizes and capacities, to meet every need.

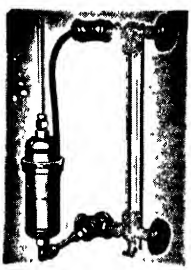
Sold by leading manufacturers of fan-system apparatus (list upon request). Send for Publication No. 229-1 which contains full particulars and engineering data.

Maid-O'-Mist, Inc.

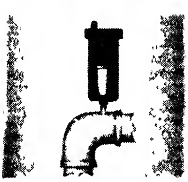
180 N. Wacker Drive, **Chicago, Ill.**

HUMIDIFIERS, WATER-LINE CONTROL VALVES AND AIR VENTS

No 855 Micro Boiler Protector Feeder and Cut-out



Water-Boy Safety Feeder



No 7 Auto-Vent Air Eliminator for Hot Water Lines

No 200 Auto-Vent Air Relief Valve for Hot Water Radiators

All **Maid-O'-Mist Products** are built entirely of Monel metal, brass, nickel silver and copper with nickel plated finish. They are low in price

The **"Micro" Boiler Protector** is small in size and finished to conform with the modern jacketed boiler They are for steam pressure up to 15 lb

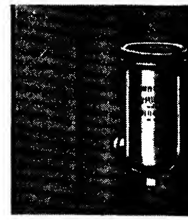
No. 200 Auto-Vent for hot water radiation. Will operate on pressures up to 25 lb Size 1½ in. x 3½ in. No. 7 Auto-Vent is used on concealed radiators, over head hot or cold circulating water lines on pressures up to 75 lb Shells are brass N.P. finish, size 2⅛ in x 4 in.

No. 95 Auto-Vent Humidifiers for low pressure steam heating quickly vent radiator and supply humidity as well. It may be connected in series for commercial installation. The amount of humidity is adjustable N P finish, size 12⅛ in. x 4½ in.

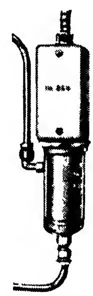
The **Zephyr Humidifier** is designed for forced air circulation The patented wings deflect the heated air over the water surface. The pan is built of bronze Made in eleven styles, 26 in and 36 in lengths.

Midget Water Feeders are made in eight types for humidifying and air conditioning. Self closing, metal to metal or Neoprene seats. Length 7 in. over all. N P. finish. These valves can be furnished with a 2½ in. x 2½ in x 6 in enameled reservoir and cover, saddle valve and copper supply tubing.

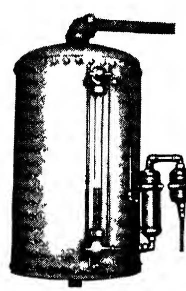
Send for Catalog G for complete information.



Auto Vent Humidifier for Steam Radiators

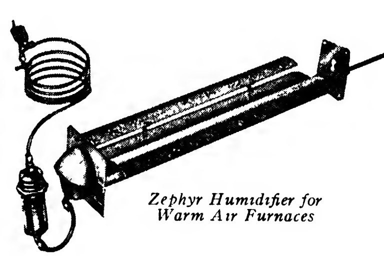


"Micro" Low Water Cut-out



Automatic Expansion Tank open or closed

Water Pan Feeder Length 7 in over all Made in 8 types



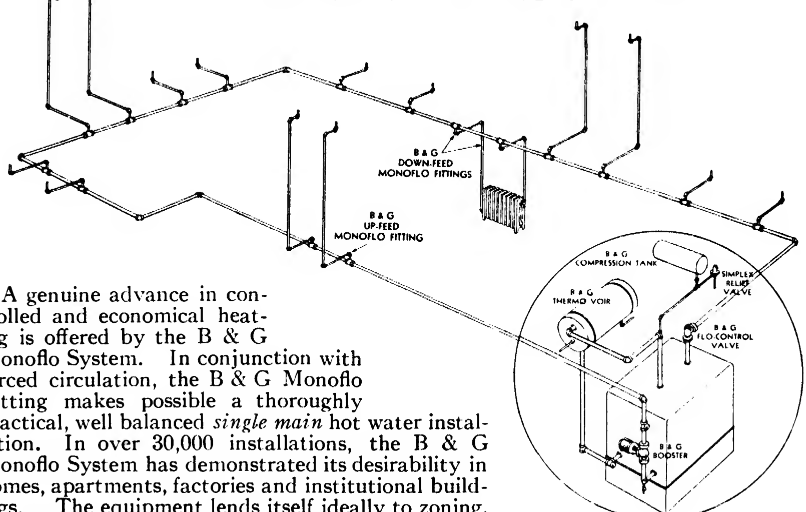
Zephyr Humidifier for Warm Air Furnaces

Bell and Gossett Company

3000 Wallace Street **Chicago, Ill.**

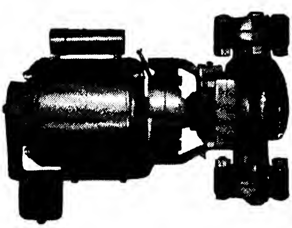
HOT WATER SYSTEMS AND SPECIALTIES

B & G MONOFLO HEATING SYSTEMS



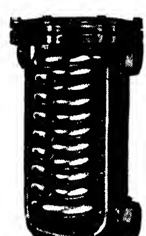
A genuine advance in controlled and economical heating is offered by the B & G Monoflo System. In conjunction with forced circulation, the B & G Monoflo Fitting makes possible a thoroughly practical, well balanced *single main* hot water installation. In over 30,000 installations, the B & G Monoflo System has demonstrated its desirability in homes, apartments, factories and institutional buildings. The equipment lends itself ideally to zoning, yet is exceedingly simple in application.

EQUIPMENT REQUIRED



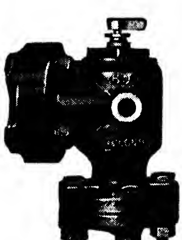
B & G Booster

An electrically-driven centrifugal pump, which mechanically circulates hot water through the system—distinguished by genuine oil lubrication, patented water-tight seal and precision manufacture throughout.



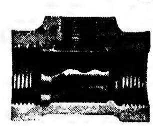
B & G Indirect Water Heater

Any one of five B & G Heater types can be installed to furnish year around domestic hot water at smallest possible cost.



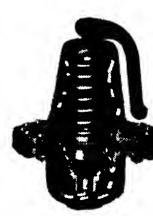
B & G Angle Flo-Control Valve

This valve, installed in the main, controls circulation of hot water to radiators, permitting summer operation of the Indirect Water Heater. It also helps maintain a uniform room temperature during the heating season.



B & G Monoflo Fitting

A correctly engineered fitting, installed in the main at radiator connections, which diverts water into the radiators. Its design assures a balanced distribution of water without introducing excessive resistance.



Simplex Relief Valve

For boiler protection.

See Current B & G Catalog for Complete Engineering Data

C. A. Dunham Company

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450 E. Ohio Street, **Chicago, Ill.**

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TORONTO, ONT , CANADA

COMFORT CONDITIONING by DUNHAM

C. A. Dunham Co., Ltd.,
(Of the United Kingdom)
Morden Road, LONDON, S W 19,
ENGLAND

The accumulated experience of the entire Dunham Organization is put at the disposa of the Heating Ventilating and Air Conditioning Engineer This cooperation is available for *Modernization Work*, as well as for new construction in industrial, commercial, housing and other projects

"Dunham Heating Service" in local classified telephone directory in all principal cities

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Dunham Sub-atmospheric Steam Heating is provided by a simple two-pipe heating system in which all the essentials of circulation, distribution and control are coordinated. Steam volume and temperature resulting from variation of pressure are the fundamentals of the Sub-atmospheric System Control of the temperature of the steam in the radiators is accomplished by controlling the pressure or vacuum of the steam in the supply piping and radiators.

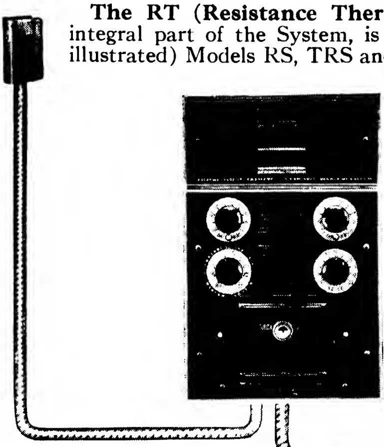
The Dunham Sub-atmospheric System distributes a varying supply of heat equally, automatically and continuously through the heated space Desirable building temperatures are automatically maintained under varying weather conditions. A positive *continuous* circulation is maintained as a fundamental function of the system This maintains unusually constant temperature levels throughout the building. At a control station which may be located in the boiler room, remote readings of building temperatures and operating conditions may be taken.

***The Control is fully automatic.** Beginning with a maximum radiator heat-output obtained by steam circulation at a pressure of 2 pounds and a temperature of 218 F or more as required, the output is progressively reduced according to the demands of the weather, by a reduction in the *rate* of steam admission to the system, which automatically causes a reduction in steam pressure and temperature so that steam may be circulated at varying tem-temperatures down to about 133 F

*Can also be installed for manual control.

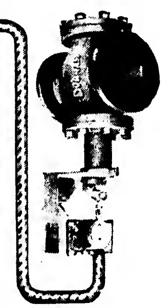
Further reduction in heat-output is obtained by partial filling of radiators with sub-atmospheric steam until the point is reached at which the need for heat ceases and the supply of steam is completely shut off

Measuring Heat Demand. The demand for heat is measured by the resistance thermometer principle Resistance thermometers measure and indicate temperatures or temperature changes, they are uniquely sensitive, accurate and long lived, *operating without moving parts or wear* Variations in temperature at control points create variations in electrical resistance in control circuits The supply of heat is varied with the demand by using these variations in electrical resistance through Wheatstone Bridge circuits to actuate the control valve which governs steam supply. The distribution of the steam supply is automatically maintained under all variations in supply by the coordinated functioning of the Traps, Pump, Differential Controller and Regulating Orifices at radiator inlets



The RT (Resistance Thermometer) control equipment, which is an integral part of the System, is available in three models. Model T (which is illustrated) Models RS, TRS and TRST. **The Model T** for indicating and controlling steam supply in proportion to the demand as measured by room temperatures, consists of a Panel, a Control Valve and one or more Thermostats. It gives:

1. Temperature control so sensitive that a fraction degree F change at the Thermostat changes the valve opening a slight amount.
2. Temperature control so stable that the control valve cannot take sudden swings of full opening or closing under automatic control.
3. Thermostat setting *at the Panel* and not at any other point.
4. Room temperature indication at the Panel.
5. An Operating Guide permanently attached to the Panel.



The MASTER SWITCH with 10 selective stations provides:

Automatic Control Stations

1. Day Temperature—Range 68 to 86 F.
2. Night Temperature—Range 50 to 68 F.
3. Clock used when Time Switch is added to automatically switch from Day Temperature to Night Temperature.

Semi-Automatic Control Station

4. Remote control of Valve position—Range 0 to 100% open.

Manual Control Stations

5. Rapid opening of Control Valve.
6. Rapid closing of Control Valve.

Indicating Stations

7. Room temperature—Upper Range 68 to 86 F.
8. Room temperature—Lower Range 50 to 68 F.
9. Valve Position—Range 0 to 100% of full open.
10. "Off" Station—All control and indicating functions discontinued.

Model RS for indicating and controlling steam supply with balance between the demand as measured by the Selector (window thermostat) and supply as measured by the Heat Balancer (heat-rate thermostat) consists of a Panel, a Control Valve, a Selector and a Heat Balancer.

Models TRS and TRST for indicating and controlling steam supply in proportion to the demand as measured by room temperatures within the limits established by Heat Balancer and Selector, consist of a Panel, a Selector, a Heat Balancer and one or more room Thermostats. These are the only controls on the market that provide fully automatic control in response to the effect of outside weather and to building heat loss.

TWO OTHER OUTSTANDING PRODUCTS

Temperator A compact, efficient and attractive all purpose Heating and Cooling Unit for the home, store or office. Whether for winter or summer service or both the Temperator may be installed with access to outside atmosphere. In winter service this unit may be used with sub-atmospheric heating or with pressure steam or hot water system, providing clean, properly warmed and correctly humidified air with quiet air movement. In summer service instead of adding heat and humidity it dehumidifies and cools. The change from heating to cooling is made by switching from a boiler connection to a chilled water supply or a refrigerating unit which may be at a remote point.

Sav-T-heat System (using gas fuel). A practical, efficient and economical method of residential heating and air conditioning. May be installed as an air conditioning system, as a vapor radiator system, as a hot water system or a combination of these.

The Bristol Company

Waterbury, Connecticut

Branch Sales and Service Offices

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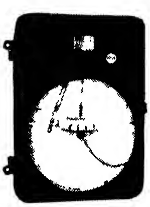
The Bristol Company of Canada, Ltd , 64 Princess Street, Toronto, Ontario
Bristol's Instrument Co , Ltd , North Circular Road, London, N W. 10, England

A COMPLETE LINE OF INSTRUMENTS FOR RECORDING, INDICATING AND CONTROLLING

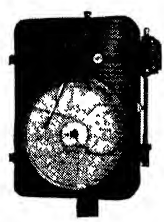
TRADE-MARK
BRISTOL'S
REG. U. S. PAT. OFFICE

FLOW METERS, ELECTRIC AND MECHANICAL

For indicating, recording, automatically controlling, and integrating flow of steam, air, gases, liquids. Electric Flow Meters telemeter measurements of flow any distance up to several hundred miles *Catalog No. 1050.*

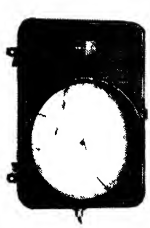


Electric Flow Meter, Model M1040 M



Mechanical Flow Meter, Model 1140M

RECORDING PRESSURE GAUGES



Recording Pressure Gauge, Model 40M

For securing continuous records of pressure, vacuum or liquid level for steam, air, gas, liquids. For ranges from full vacuum to 12,000 lb per sq in *Catalog No. 1014.*

RECORDING AND INDICATING THERMOMETERS

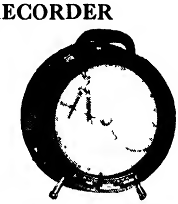
Recording Thermometer, Model 240M

Recording Thermometer: For all commercial ranges from 60 F below to 1000 F above zero. For wall, flush or panel mounting. *Catalog No. 1250.*

Recording and Indicating Resistance Thermometers: For securing, at a central point, readings of a number of temperatures at distant points. *Catalog No 1451 and Bulletin No 997.*

DIRECT READING RELATIVE HUMIDITY RECORDER

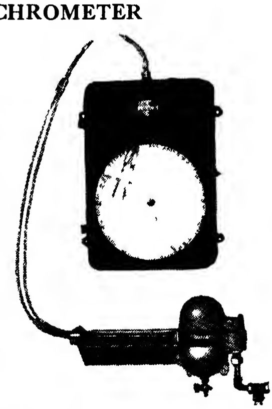
Shows trend of humidity condition. No calculations or humidity tables needed Requires no water, no fan Portable, corrosion-resisting case. *Catalog No. 1250.*



Right. Thermo-Humidigraph, Model 4069

WET- AND DRY-BULB PSYCHROMETER

Records wet- and dry-bulb temperatures in kilns, drying rooms, air ducts Self-contained and distance types Moisture-, fume-, and dust-proof case. For wall, and flush panel mounting. *Catalog No 1250*



Wet- and Dry-Bulb Psychrometer, Model 4240M

INDUSTRIAL STEM THERMOMETERS

Plain or red background. With fixed thread, union connection or socket design. Straight form, or rear oblique, right side and left side angle forms. Also portable stem thermometers with ring top, ring handle top, or adjustable stem handle forms. Fenestrated guard or plain bulb. *Catalog No. 1250.*

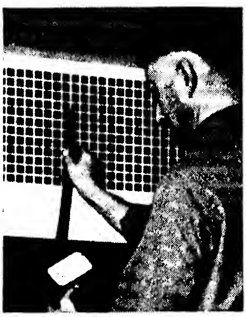
Straight Form

Illinois Testing Laboratories, Inc.

422 N. LaSalle Street, **Chicago, Illinois**

TESTING ENGINEERS AND MANUFACTURERS

Pyrometers—Portable—Wall Type—Surface Temperatures
Distant Reading Resistance Thermometers
Automatic Temperature Controllers
Air Velocity Meters



Velometer with averaging jet used for checking velocity from supply grille.

"ALNOR" VELOMETER

The Only All-Purpose Air Velocity Meter

The Velometer is a versatile direct reading air velocity meter which gives instantaneous readings of the speed of air measured in feet per minute.

Anyone can use the Velometer. No mathematical calculations, no leveling—no timing.

As its movement is actuated by the pressure or impact of the air against a swinging vane, it is essentially a pressure instrument—thus it can be scaled to not only read velocities directly, but also to read static or total pressures when using suitable jets.

Made in three standard ranges for velocity readings from 20 fpm to 6000 fpm and 3 in. static or total pressure. Special ranges available up to 18,000 fpm velocity and 20 in. pressure.

Jets—Several standard jets are offered providing a wide application.

Spot jets—for velocities over very small area.

Averaging jets—for obtaining average velocities over a definite area or grille face.

Duct jets—for determining velocities directly within ducts or pipes.

Static pressure jet—for static pressures in inches of water.

Total pressure jet—for total pressure in inches of water.

Other jets—Standard jets offered in several lengths and sizes.

Special jets—can be designed for unusual applications.

Ask for Bulletin No. 2448-B

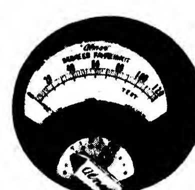
"Alnor" Distant Reading Electric Thermometers

The use of "Alnor" multi-point resistance type thermometers is rapidly increasing not only in air-conditioning installations, but also for heating plants.

The instrument can be located in the machinery room or boiler room with the elements located on various floors in any part of the building, or outdoors, thus providing the engineers with constant and convenient temperature readings.

"Alnor" thermometers are made in several styles and sizes, both portable and mounted types.

Ask for Bulletin No. 2451-A



"Alnor" round type multi-point resistance thermometer with built-in switch.

Leeds & Northrup Company

General Office and Works: 4941 Stenton Avenue, **Philadelphia, Pa.**

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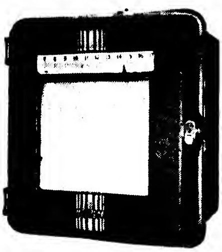
RUGGED, NULL-TYPE INSTRUMENTS THAT ARE RELIABLE



Model S Micromax Recorder

Records from 1 to 16 points on a single strip-chart. Extremely open record. Can operate signals. (About 1/15th size)

Model R Micromax Recorder

Records 1 or 2 points on a round-chart. Has extremely readable dial. Can operate signals. (About 1/15th size)

Switchboard Indicator

Hand-operated. Can be connected through selector switches to any number of points. (About 1/10th size)

Electrical Thermometers for Air Conditioning

No method for measuring temperatures fits the specific needs of air conditioning as does the three-lead null-type resistance thermometer method. It is independent of distance and disregards all temperatures except those right at detector locations. The detectors (resistance thermometer bulbs called Thermohms), can be placed anywhere—in rooms, air ducts or water lines. They are connected by simple electrical wiring to instruments at a central location. Instruments may be: Micromax Recorders, Model S for up to sixteen Thermohms; Micromax Model R, for related pairs such as wet and dry bulb; indicators with switches for any number of Thermohms; or indicating and recording combinations.

This equipment is fundamentally reliable. Instruments and Thermohms are highly responsive, yet rugged in construction. A complete system is easy and economical to install, regardless of distances. It is easy to operate and demands minimum maintenance. Thermohms and instruments are interchangeable, and can be replaced without disturbing wiring or returning anything to the factory.

L&N Resistance Thermometers make it possible to operate efficiently; to maintain comfort or correct process atmosphere constantly . . . so that maximum return is realized on the conditioning investment.

Electrical Instruments for the Heating Plant

The facts needed to operate a modern heating plant so as to save fuel, to protect equipment, and to operate efficiently at varying loads are provided reliably by rugged L&N instruments. Readings can be indicated or recorded or both. Recorders can be equipped to operate signals or alarms that warn the operator of extreme conditions. In some cases the instruments control automatically.

Micromax Model S provides a permanent record of conditions at from 1 to 16 points on one wide-scale chart. Micromax Model R concentrates on conditions at one point, provides a permanent record, and has a giant indicating dial that can be read at a glance. The switchboard indicator provides intermittent checks on conditions at one or several points.

In the heating plant, L&N measuring, signalling or controlling equipment is used for:

Metermax Combustion Control.
Furnace Pressure Control.
Smoke Density Analysis.
Flue Gas Analysis (Percent CO_2).
Flue Gas Temperatures.
Steam and Water Temperatures.
Boiler-Furnace Temperatures.
Electrolytic Conductivity of Water.

J-N-225(2)

The Liquidometer Corporation

38-16 Skillman Ave.

Long Island City, N. Y.

Tank Gauges, Liquid Level Controls
Remote Reading Thermometers

ADVANTAGES OF LIQUIDOMETER AND LEVELOMETER TANK GAUGES

(1) Accurate check on liquid deliveries and consumption provided. Correct inventory constantly available.

(2) Inaccurate and dangerous "gauge sticking" or "taping" methods eliminated.

(3) Tanks can be gauged without loss of pressure or vacuum.

(4) Overflowing of tanks—and consequent losses eliminated.

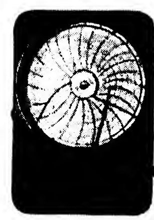
(5) Extra handling of liquids in many instances saved.

(6) Costly shut-downs due to lack of fuel prevented.

(7) Approved by Underwriters Laboratories, for gauging hazardous liquids—fire and labor hazards cut down.

LIQUIDOMETER—THE 100% AUTOMATIC READING TANK GAUGE

Senior Liquidometer



Liquidometer Recorder

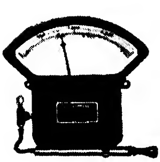
Liquidometers are self-contained. Large clock-faced dials provide completely automatic readings without the use of pumps, valves, electrical energy, or other outside source of power, and they are not affected by specific gravity differences.

They will gauge practically any liquid that will flow, light or heavy. Used on fuel oil tanks in residences, apartments, public buildings, industrial plants, U. S. Army and Navy air and water craft, on freight and passenger vessels, and for other liquid gauging requirements.

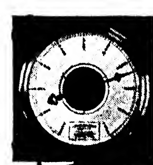
Direct Reading Models are used where remote reading is not required. On this type, the dial indicator is located on the tank itself, and a metallic bellows is used to seal the indicator from the tank, thus preventing escape of liquid, gas or pressure.

Models are available to automatically control pumps, motors, signals or other devices for maintaining desired minimum or maximum liquid levels.

LEVELOMETER TANK GAUGES (HYDROSTATIC)



Large Model Levelometer



Master Model Levelometer

Levelometers utilize the principle of hydrostatic balance between the head of liquid to be measured and a dial indicator—there is no liquid in the indicator to expand or contract and create error.

Dial indicators have large graduations and figures and, if desired, one dial indicator can be arranged to gauge a group of tanks.

In this instrument the tank liquid traps and compresses air in an "air chamber"—the pressure depends upon the height of liquid in the tank, and this pressure is transmitted through tubing to a metal bellows in the indicator unit; the slightest movement of the bellows operates the dial indicator. As air is required to insure correct readings, a manually operated pump is generally supplied, but, where compressed air is available, a valve may be substituted for the pump, thus providing a continuous reading gauge.

Levelometers are made in a number of different models to suit various tank capacities and desired dial calibrations. Models for tanks under pressure also are available.

We will welcome the opportunity to make recommendations for perplexing liquid level gauge problems.

Manning, Maxwell & Moore, Inc.

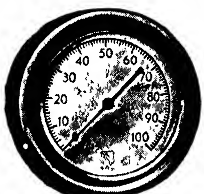
Bridgeport, Conn.

BRANCHES IN PRINCIPAL CITIES

Makers of AMERICAN INDUSTRIAL INSTRUMENTS—Since 1851

Manufacturers of Indicating and Recording Gauges; Gauge Testers; "U" Gauges; Draft Gauges; Indicating and Recording Thermometers; Tachometers; Dial Thermometers; Pressure and Temperature Controllers; Electric Temperature Controllers; Pop Safety and Water Relief Valves; Steam Traps; Absolute Pressure Gauges.
Also manufacturers of Bronze, Cast Steel and Forged Steel Valves, Engine Room Clocks; Barometers; Mercury Column Gauges; Gauge Boards.

Ashcroft American Gauges—Ashcroft American Gauges are made in all sizes from 2½ to 12 in., for pressures from 8 oz to 25,000 lb and also for vacuum. Cases are cast-iron or cast brass. The movements are heavy duty and all bearings are Monel Metal. Write for Catalog No. A-59.

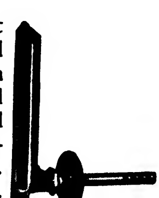


Also Duragauges—accurate to within ½ of 1%. Stainless steel movement. In Phenol Cases in 4½ in., 6 in. and 8½ in. dial sizes.

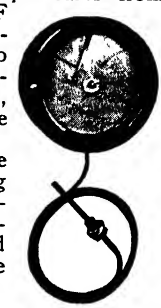
For Mercury Pressure and Vacuum Gauges, "U" Gauges, Draft Gauges and Mercurial Barometers, write for Catalog B-59.

Recording Duragauges—Recording Duragauges are made for all pressures from 15 in. of water to 10,000 lb and for vacuum. They are made in one size only to accommodate a 10 in. chart, having an effective scale width of 3⅝ in. The case is die cast with a dull black hard-rubber finish and with either bottom or back connection. The pen-arm is made of non-corrosive monel metal and is of the inverted type. Operating instructions are lithographed on the chart plate so that they cannot be lost. Write for Catalog E-59.

American Air Duct Thermometer–Designed especially for both warm and cold air ducts. Fitted with chromium plated frame, glass front. Furnished with 9-in. or 12-in. scale graduated 0-160 F. Write for Catalog F-59.

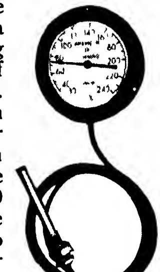


American Recording Thermometers—Made for recording temperatures from minus 40 to plus 1000 F or equivalent C. Very flexible connecting tubing up to 200 ft. One size to accommodate 10 in. chart, with an effective scale width of 3⅝ in.



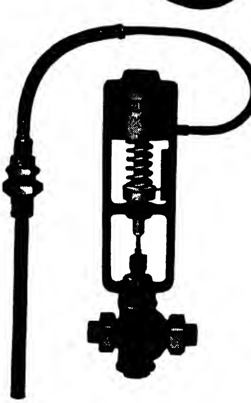
Same case as for the American Recording Gauge, so that all instruments are uniform in appearance when mounted on Gauge Boards. Write for Catalog H-59.

American Dial Thermometers—American Dial Thermometer (mercury-filled) has the accuracy of the standard glass tube thermometer and the reading convenience of a dial face. Entire working mechanism is made of steel, meaning long life.



Six sizes, ranging from 4½ in. to 12 in. diameter dials. Furnished with rigid connection or flexible capillary tubing up to 200 ft. For temperature ranges from minus 40 to plus 1000 F. Write for Catalog G-59.

American Precision Temperature Controllers—Self-operated. For regulating temperatures from 20 to 475 F. For hot water service tanks, water heaters, etc. Size of valve must be specified. Write for R-59 Bulletin.



The Palmer Company

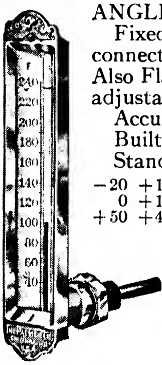
Main Plant: 2506 Norwood Ave., **Cincinnati (Norwood), Ohio**

Canadian Factory: THE PALMER THERMOMETER CO., LTD., King and George Sts., Toronto

Manufacturers and Originators—"Red-Reading-Mercury" Thermometers

THERMOMETERS for Heating and Air-Conditioning with "Red-Reading-Mercury".

Industrial Style Thermometers.



No. 327

7 in., 9 in. and 12 in. case.
STRAIGHT and various ANGLE forms.
Fixed thread fitting, Union connection, Separable Socket. Also Flanges: Fixed, Union and adjustable.
Accuracy Guaranteed.
Built sturdy for long service.
Standard Ranges:

−20 +120F. + 30 +240F.
0 +100F. + 30 +180F.
+50 +400F. +200 +550F.
+200 +750F.

Stem Length: 3½ in. including thread. Longer lengths furnished.
Finishes: Nickel-plate, chrome-plate, Polished brass, Instrument-black, etc.
PALMER thermometers show a bright RED column, seen at a great distance, through steam, smoke, dust, etc. For Genuine mercury thermometer with RED column, Specify: PALMER.

No. 14200

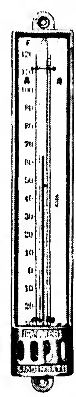
Psychrometers

For testing moisture in the air, this Sling Psychrometer is necessary for all Air-conditioning jobs. Pocket style, in leather case, with wet- and dry-bulb. Convenient for making daily tests.

Because the mercury column in the tubes show a bright RED column, it is much easier to read than plain mercury, after swinging. Range: +30 +110F.

If you want a good reliable thermometer, thoroughly aged for permanent accuracy, specify: PALMER

Write for FREE catalog.



No. 2120 Thermometer

Pocket Thermometers

No. 13240

Every engineer should carry this accurate type thermometer around at all times. Handy and easy to read, with the RED column. Standard ranges: −30 +120 F. 0 +220 F. Can be furnished in maximum-registering style also.

Wall Thermometers

Can be used inside and outdoors. This style made of bronze castings with white duco finish. Also furnished in chrome finish. With the RED column, it can be seen at a great distance. Other styles made. Has a graduated, accurate scale.

Laboratory Thermometers

For tests to be made at any time, these thermometers are very valuable. ROUND or LENS glass. From 6 in. to 16 in. length. Also longer lengths. All ranges furnished, with the RED column. Made of selected glass. Every thermometer PALMER furnishes is annealed and treated so as not to change with age or use. It pays to buy the best.

Precision Thermometers

Have on hand a reliable test thermometer in Fractional Divisions, so as to make very close and accurate tests. In ½ deg, ⅕ deg and $\frac{1}{10}$ deg divisions. With Test Certificate. Easy to read with PALMER "Red-Reading-Mercury" feature. The mercury thermometer with a wide RED column.

No. 10042

Repairs to all makes of thermometers with "Red-Reading-Mercury" at no extra charge.

A trial order will convince you!

Taylor Instrument Companies

Rochester, N. Y., U. S. A.

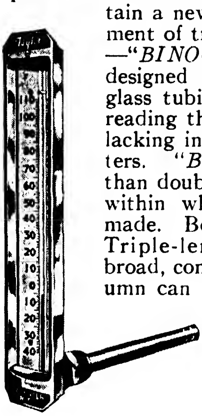
IN CANADA—TAYLOR INSTRUMENT COMPANIES OF CANADA, LTD., TORONTO

NEW YORK	PHILADELPHIA	LOS ANGELES	ST. LOUIS	DETROIT
CHICAGO	PITTSBURGH	INDIANAPOLIS	CINCINNATI	ATLANTA
BOSTON	CLEVELAND	SAN FRANCISCO	TULSA	MINNEAPOLIS

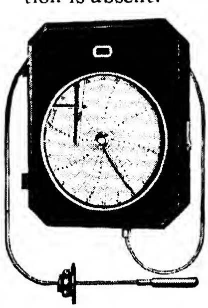
Manufacturing Distributors in Great Britain, Short & Mason, Ltd. London

Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure, Humidity, Flow and Liquid Level

Taylor Industrial Thermometers—with new "BINOC" Tubing—This line of thermometers includes many styles and scale ranges with bulbs for every application. Suitable for air ducts, kiln temperatures and oven temperatures. But these thermometers contain a new and radical development of tremendous importance—"*BINOC" Tubing*. This newly designed and optically correct glass tubing assures an ease of reading that has been generally lacking in industrial thermometers. "*BINOC*" *Tubing* more than doubles the angle of vision within which readings can be made. Because of the patented Triple-lens construction, its broad, contrasting mercury column can be read easily and accurately with both eyes at close range and also at greater than normal distances. Bore reflection is absent.



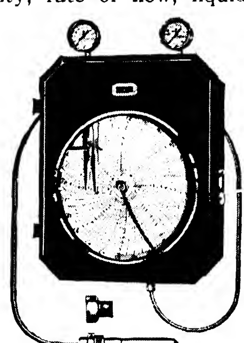
Taylor Recording Thermometers—Temperature ranges and time requirements vary greatly in heating and ventilating work. Taylor Recorders are made in needed scale ranges and time periods. These instruments are beautiful and efficient, particularly adapted for heating and air conditioning applications. They may be had for surface or flush mounting. The polished flanges make an effective panel board installation.



Taylor Electric Contact Temperature Control—These instruments combine in the same case an electrically - operated temperature controller with an indicating thermometer. One tube system operates both units.

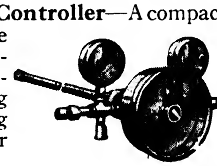
The New Taylor "Fulscope" Recording Controller—An air-operated controller that gives practically any character of process control regardless of *time lag* in apparatus.

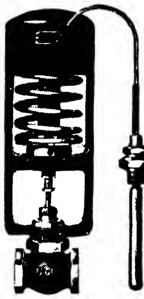
Available for controlling temperature, pressure, humidity, rate of flow, liquid level. Where extreme load changes or badly balanced operating conditions exist, the *Taylor "Dubl-Response Control Unit"* is the only positive means of maintaining control-point. *Write for literature.*



New Taylor Indicating "Fulscope" Controller—For those control applications where a high-quality air-operated controller is desired for close and sensitive regulation, but a record is not required, this instrument is available for temperature, pressure, flow and liquid level.

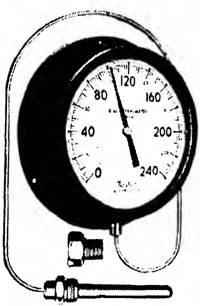
Taylor Type-P Controller—A compact and very sensitive air-operated controller, ideal for air-ducts, air-washing machines, cooling rooms and similar applications.



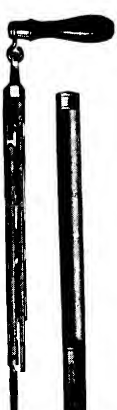


Taylor Self-Acting Temperature Controller — Adapted for use on hot-water storage tanks, etc. It requires no auxiliary motive power, such as compressed air, to open and close the steam valve.

The valve can be closed at any desired temperature, or a throttling action can be obtained. Not practicable on pipe lines having steam pressure over 125 lb.

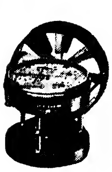


Taylor Dial Thermometers for air ducts or any application where it is desirable to have temperature readings at some distance from the thermometer bulb, as in a central control room. Can be read at a glance as easily as a steam gage.

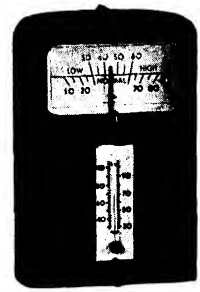


Taylor Sling Psychrometer—The advantage of this form of Wet- and Dry-Bulb Hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as in whirling the bulbs they are subjected to perfect circulation. Consists of two accurate etched stem thermometers mounted on a die-cast frame, with the bulb of one covered with a wick to be moistened.

These thermometers have scales of 0 to 100 F, graduated in ½ deg divisions. A copper case protects the tubes when not in use.

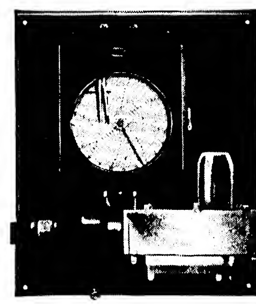


Taylor Anemometer—This instrument is ideal for measuring air velocities with the fan revolutions indicated on the dial. Available in various models for a wide range of air speeds and registration limits.



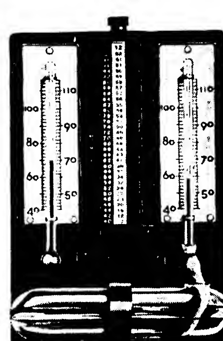
Taylor Hampton-Model Humidiguide (Direct-Reading)—A hygrometer giving direct humidity percentages, in a smart modern case suitable for home, office or public buildings. Finish is rich brown.

The Permacolor Thermometer is filled with non-fading, easy-reading red liquid.



Taylor Recording Hygrometer—This instrument records both wet- and dry-bulb temperatures on the same chart in different colored inks, making comparison very easy.

Type shown above with motor-driven fan for conditioned rooms or passages in which circulation is poor. Can be supplied without fan for installations where circulation across bulb is good.



Taylor Humidiguide—A handsome small hygrometer for the wall of the home, office, school or other building where a neat, easy-reading and inexpensive instrument is desired. It is self-contained, requiring no charts or separate tables. Frame is Mahogany Bakelite.

For complete information on above instruments and others designed for heating, ventilating and air conditioning, send to any of the offices listed on the previous page for new Taylor Catalog Number Five.

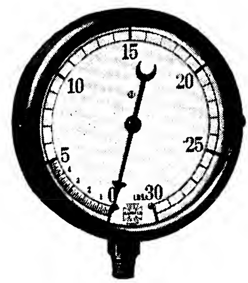
United States Gauge Company

U. S. GAUGE CO. N. Y.

Indicating and Recording Pressure Gauges

44 BEAVER STREET · NEW YORK

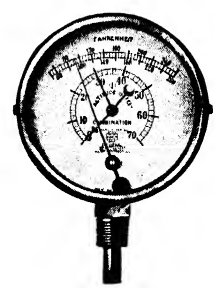
FACTORY · SELLERSVILLE · PENNSYLVANIA
BRANCHES · NEW YORK · CHICAGO · PHILADELPHIA
BOSTON · CLEVELAND · DETROIT · ST. LOUIS
HOUSTON · SEATTLE · LOS ANGELES · MONTREAL



U. S. GAUGES—U. S. Gauges are made in all sizes from 2 to 12 in. inclusive for pressures from 1 lb up to 50,000 lb, and for vacuum. Cases may be cast-iron, cast brass, drawn steel and drawn brass for wall mounting or flush mounting. For severe service long wearing hardened steel or bushed movements may be supplied.

For service on Steam Heating Systems—

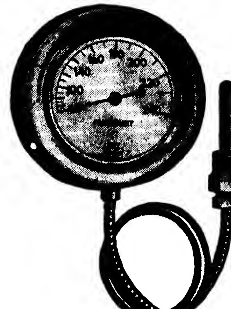
Steam Gauges . . . Compound Pressure and Vacuum Gauges . . . Retard Gauges . . . Compound Retard Gauges . . . Steam Gauges with Internal Siphons.



For Hot Water Heating Systems—

Altitude Gauges . . . Tank-in-Basement Gauges . . . Altitude and Pressure Gauges . . . Combination Altitude Gauges, and (a) Bimetal Thermometers, (b) Glass Tube Thermometers, (c) Vapor Tension Distance Type Thermometers . . . Glass Tube Hot Water Thermometers.

U. S. RECORDING GAUGES—U. S. Recording Gauges are made in 8½, 10 and 12 in. sizes for pressures from 1 lb up to 50,000 lb, and for vacuum. Cases may be cast-iron or cast brass for wall mounting or flush mounting. Pen arms are made of non-corrosive metal. Especially designed clock movements are used. Charts can be furnished for customary time periods.



U. S. DIAL THERMOMETERS—U. S. Dial Thermometers are of the vapor tension type with open scale reading in the central and upper portion of the scale. Cases may be cast-iron, cast brass, drawn steel or drawn brass for wall mounting or flush mounting. Supplied in all sizes from 2 to 12 in. inclusive, for temperature ranges from −40 F to 800 F. Furnished with rigid connection bulb or with flexible capillary tubing up to 100 ft long.

Alfol Insulation Company

Incorporated

155 East 44th St., **New York, N. Y.**

Agents in Principal Cities

INSULATION *for*

Houses
Buildings
Ducts
Blowers
Boiler Jackets
Radiators
Air-Conditioners
Dehumidifiers
Boilers
Hot Water Tanks

At temperatures up to 1200 F.

ALFOL

PAT. IN 34 COUNTRIES

INSULATION *for*

Refrigerators
Refrigerator Cars
Refrigerator Trucks
Cold Storage Rooms
Buses and Coaches
Passenger Cars
Locomotives
Ships
Ovens
Ranges
Tanks
Stills
Towers
Pipes

TECHNICAL DESCRIPTION OF ALFOL

ALFOL consists of one or more sheets of pure aluminum foil installed to build up heat insulation by providing barriers against the passage of heat. Sheets of ALFOL block air currents and seal against moisture The metallic surfaces reflect heat radiation and eliminate 95 per cent of this form of heat flow. ALFOL *has the dual advantage of being light in weight and low in heat absorption, and conduction through the* separated layers of foil is negligible.

ALFOL is also furnished combined with other materials such as aircell asbestos for specialized applications

PERMANENCE

The outstanding quality of ALFOL is its permanence. It resists fire and water, it is not affected by vibration, shrinking, settling, bulging, warping ALFOL is a lasting material

Every aluminum surface oxidizes as soon as it is formed The oxide is transparent and forms a protective film. On the pure aluminum of which ALFOL consists, this film is continuous and forms a coating which protects against atmospheric conditions. The reflectivity of ALFOL with this oxide film is 95 per cent. After years of service, ALFOL tests at 95 per cent reflectivity

EFFECTIVE AS VAPOR BARRIER

ALFOL is a recognized vapor barrier. Being pure metal it cannot absorb moisture and moisture cannot penetrate it ALFOL is a dry insultaion.

SERVICE EXPERIENCE

The first large applications of ALFOL were made in this country in 1931 and in Europe four years earlier Eight years here and twelve years abroad have proven ALFOL's reliability and permanence Millions of square feet of ALFOL aluminum foil in ships, refrigerators, refrigerator cars, trucks, locomotives, pipes, ovens, houses, etc , prove as efficient today as when installed. The high insulating efficiency of ALFOL is constant.

Data on Heat Savings Effected by Alfol

Construction	Heat Transmission Btu/Hour/Sq Ft/F			Heat Transfer Stopped . . . %	
	Not Insulated	1 layer ALFOL	2 layer ALFOL	1 layer ALFOL	2 layer ALFOL
Wood Stud Wall—4-in studs	0 25	0 11	0.09	56%	64%
8-in. Brick Wall—Furred	.30	.12	.09	60	70
Open Attic Floor	.62	13	.10	79	84
Wood Shingle Roof (no lath and plaster)	.46	.13	09	72	81
Concrete Roof—6-in slab—suspended ceiling	35	.13	.10	63	72
Composition Shingle Roof (no lath and plaster)	.56	15	.11	73	81

Alfol Actively and Instantly Reflects Radiant Heat

Aluminum Aircell Insulation Co.

Curtis Bldg., **Detroit, Mich.**

INSULATION FOR

Air Conditioners
Brooders
Buildings
Chicken Houses
Dairy Barns
Fruit Storage
Homes
Hot Houses
Incubators
Refrigerated Rooms

REFLECT-O-CELL

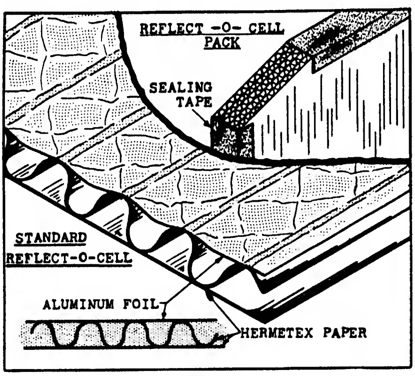
Licensed under Patents:
1,757,479—1,890,418—1,934,174
others pending

INSULATION FOR

Automobiles
Beehives
Busses
Floral Shipping Boxes
Heaters
Ovens
Ranges
Shipping Containers
Trailers
Trucks

	Heat Transmission Btu/Hour/sq ft/F			Reflect-O-Cell Saves per cent	
	Not Insulated	1 layer Reflect-O-Cell	2 layers Reflect-O-Cell	1 layer Reflect-O-Cell	2 layers Reflect-O-Cell
Wood Stud Wall—4 in. studs	0.26	0.13	0.09	50	65
Open Attic Floor	0.69	0.19	0.12	69	82
Wood Shingle Roof (no lath and plaster)	0.56	0.18	0.10	68	81

Thermal Efficiency determined at the University of Detroit under humidity conditions prevailing in actual use and also at the University of Toronto, Toronto, Canada.



REFLECT-O-CELL is the modern insulating material based on the famous Dewar Principle of insulation, the outstanding example of which is the efficient Thermos Bottle.

Light in Weight, not depending on thickness nor density, REFLECT-O-CELL employs this Dewar Principle by means of its polished Aluminum Foil backed by corrugated Hermetex paper. (Hermetex paper was formerly used ALONE as an insulator by leading refrigerator manufacturers).

The Structure of Reflect-O-Cell permits both surfaces of the foil to reflect radiant heat.

Moisture Sealing, which effectively reduces summer dehumidifying load and winter humidifying requirements by preventing vapor pressure losses.

Windproofing all studding spaces similar to metallic window weatherstrip effect.

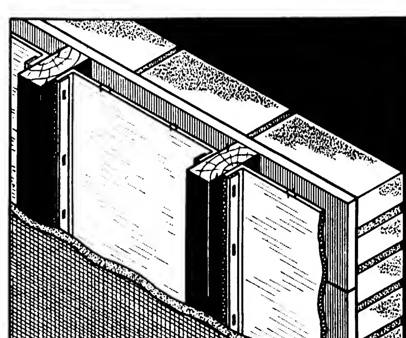
Soundproofing provided by high hysteresis air cell construction.

Especially Indicated for exposed application to eliminate air-conditioning shock and to reduce cooling load.

REFLECT-O-CELL is used in the production of insulated Buses, Trucks, Trailers, Ranges, Heaters and others. Similar satisfactory service has also been rendered in residences, industrial buildings and processing equipment.

REFLECT-O-CELL is supplied in conveniently scored continuous Roll Sheets or in Packs and is easily installed between wall studs, ceiling joists and roof rafters by stapling through its flanged edges.

Weighs only 38 lb per 1000 sq ft.



Single layer wall application.
Also installed in two layers.

Armstrong Cork Company

Building Materials Division

Lancaster, Pennsylvania

Branch Offices

Albany	Cincinnati	Houston	Minneapolis
Atlanta	Cleveland	Indianapolis	New York
Boston	Columbus	Jacksonville	Omaha
Buffalo	Dallas	Kansas City	Pittsburgh
Charlotte	Des Moines	Louisville	Rochester
Chicago	Detroit	Milwaukee	St Louis

Distributors

Appleton, Wis	Northwestern Asbestos and Cork Insulation Co
Baltimore, Md	John R Livezey
Boston, Mass	T R Nunan Company
Charleston, W Va	Capital City Supply Company
Cleveland, Ohio	Clark Asbestos Company
Denver, Colo	Stearns-Roger Manufacturing Company
Eau Claire, Wis	Horel-George Company
Evansville, Ind	Tri-State Asbestos & Magnesia Co
Green Bay, Wis	Northwestern Asbestos and Cork Insulation Co
Jamestown, N Y	Laco Roofing & Asbestos Company
Joplin, Mo	Joplin Cement Company
Kingsport, Tenn	Kingsport Lumber Co
Little Rock, Ark	Fischer Cement & Roofing Company
Los Angeles, Calif	Van Fleet-Freear Company
Manitowoc, Wis	Northwestern Asbestos and Cork Insulation Co
Memphis, Tenn	Grant Brothers, Inc
New Orleans, La	H T Steffee
Oklahoma City, Okla	Kelley Asbestos Products Co
Philadelphia, Pa	John R Livezey
Portland, Oregon	Asbestos Supply Company
Providence, R I	Rhode Island Covering Company
Richmond, Va	John R Livezey
San Angelo, Texas	San Angelo Building Materials Co
San Francisco, Calif	Van Fleet-Freear Company
Seattle, Wash	Asbestos Supply Company
Spokane, Wash	Asbestos Supply Company
Springfield, Mass	Johnson Asbestos Company
Springfield, Mo	Southwestern Insulation Company
Tacoma, Wash	Asbestos Supply Company
Terre Haute, Ind	The Hartmann Company
Tulsa, Okla	Kelley Asbestos Products Company
Washington, D C	John R Livezey
Wichita, Kans	Ludeman Insulations Company

For detailed technical information, samples, and descriptive literature, ask any office or representative Complete specifications appear in each of Sweet s Catalog files

PRODUCTS—Armstrong's Corkboard, Armstrong's Cork Covering, Armstrong's Vibracork, Armstrong's Corkoustic, Armstrong's Temlok, Armstrong's Temcoustic, Armstrong's Insulation Sundries.

Corkboard

Insulating Efficiency

The thermal conductivity of Armstrong's Corkboard, depending on the density, is 0 27 to 0 29 Btu per hour, per degree temperature, per inch thickness at 90 F mean temperature (U.S Bureau of Standards)

The value of adequate and efficient insulation is covered in (Chapter 5) of this book and the tables on pages 102 to 114 indicate the savings which can be effected by using 1½ in or 2 in of corkboard in standard wall and roof construction

Armstrong's DI Corkboard is recommended for the insulation of ducts where the problem is to prevent condensation It is furnished in sheets 12 in. x 36 in x ½ in and has a waterproof mastic finish on one face

Sizes and Thicknesses

Armstrong's Corkboard is furnished in rigid boards 12 in. x 36 in., 12 in. x 32 in., 18 in. x 36 in., 24 in. x 36 in., and 36 in. x 36 in., in several thicknesses: 1 in , 1½ in., 2 in , 3 in., 4 in., and 6 in

Cork Covering

Armstrong's Cork Covering is made of pure cork in sizes to fit all standard pipe sizes The inside surfaces of each piece are machined to assure an accurate fit, free from moisture-catching air pockets. Cork covering is rigid and will not sag. Thicknesses are: Ice Water (1 20 in to 1 93 in.); Brine (1 70 in to 3.00 in); and Special Thick Brine (2 63 in to 4.00 in)

Armstrong's Fitting Covers are rigid and are designed to fit accurately standard ammonia and extra heavy fittings, both screwed and flanged, of all types

Vibracork

Armstrong's Vibracork, made in two densities, is ideal for the elimination of noise and vibration transmission and is of primary importance in air conditioning work It does not take a set, is not affected by atmospheric moisture, and will not deteriorate in service.

For aid in the solution of any technical problems involving insulation, isolation, or acoustical treatment, and for literature and prices, get in touch with an Armstrong district office or distributor or the Armstrong Cork Company, Building Materials Division, Lancaster, Pa

The Barrett Company

2800 So. Sacramento Ave., Chicago, Ill.

40 Rector Street
New York, N. Y.

Fairfield, P. O.
Birmingham, Ala.

Barrett ROCK WOOL

BATTS: Paper-backed and Standard. Full thick, and semi-thick. 15 in. x 48 in. and 15 in. x 23 in. Also Demi-Batts, full thick, 9 in. x 16 in.

GRANULATED: For pneumatic application. Weight per bag, 35 lb.

LOOSE LONG FIBRE: For hand packing and irregular spaces. Weight per bag, 35 lb.

CHARACTERISTICS

Barrett Rock Wool is a mineral wool made by melting a high silica limestone. The resulting liquified rock, at between 2500 and 3000 F, is "blown" with a jet of steam, forming a fibrous, wool-like material which contains over 90% dead air cells. These cells, which resist penetration by cold, heat and sound, give Barrett Rock Wool unusually low conductivity. The Barrett Company, with a background of 85 years of building experience, believes with many architects, engineers and builders that this type of Rock Wool offers high insulation value for all types of buildings.

PERMANENT . . . LIGHT WEIGHT . . . SAFE

Barrett Rock Wool will not deteriorate with age. It is fireproof, vermin-resistant, odorless, clean, and a non-conductor of electricity. It is approved by the Underwriters' Laboratories, Inc. Its light weight and rigidity assure ease and economy in handling. It is supplied only in high and uniform quality—free from impurities and extraneous material.

VB BATTS

Barrett VB Batts are furnished with a special water-repellent, vapor-resisting paper backing which has a 1½-inch flap on all edges for lapping over adjacent batts and building framing to assure a vapor seal. (See cut.) These improved-type batts represent the last word in dependable, efficient insulation.

SPECIFICATIONS FOR APPLICATION

Detailed specifications for furnishing and installing Barrett Rock Wool Batts and Barrett Granulated Wool by the pneumatic method will be furnished upon request. 'Phone, wire or write our nearest office for complete information.

Barrett's new Paper-Backed VB Batt. Overlapping flange permits easy application to studding. Special water-proof and vapor resistant Barrett Building Paper anchored to wool extends 1½ in. beyond the edge of batt on all four sides.

Granulated Barrett Rock Wool for pneumatic installation. Exceptionally light weight promotes easier handling, more economical application. Ideal for insulation of existing buildings and other enclosed areas.

THE CELOTEX CORPORATION

919 N. Michigan Ave., **Chicago, Illinois**

Mills: MARRERO, LA. AND METUCHEN, N. J.

LONDON, ENGLAND

Boston, Mass.
Minneapolis, Minn.
Philadelphia, Pa.
Denver, Colo.
New York, N. Y.
Cleveland, Ohio
Los Angeles, Calif.
St. Louis, Mo.

Seattle, Wash.

Sydney, Australia
Paris, France
London, England
Buenos Aires, Argentina
Durban, South Africa

CELOTEX
BRAND
INSULATING CANE BOARD
REG. U. S. PAT. OFF

Builds - Protects - Insulates - Decorates - Subdues Noise

Building Board	**Tile Board**	**Cemesto**
Lath	**Roof Insulation**	**Thermax**
Vapor-seal Sheathing	**Vapor-proofed Low Temperature Insulation**	**Ornaments-Mouldings**
Vapor-seal Lath	**Insulation Blocks**	**Key Joint Units**
Finish Plank		**C-X Rock Wool Products**

Celotex Products also include Roofing . . . Gypsum . . . Asphalt Impregnated Products . . . Hardboards . . . Flexcell Expansion Joint . . . Pottsco.

Celotex Cane Fibre Insulation

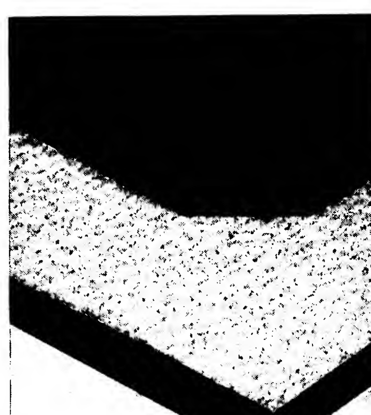
In the manufacture of Celotex, long tough fibres of bagasse (cane) are properly refined, thoroughly sterilized, effectively waterproofed, firmly felted and securely interwoven to produce large boards of maximum strength consistent with the light weight necessary to assure high heat retarding value. Careful technical control assures uniform high quality.

The thermal conductivity of Celotex is 0.33 Btu per hour per square foot per 1 F per inch thickness (based on a mean temperature of 70 F.) Tests conducted at Armour Institute of Technology and other recognized laboratories confirm this figure.

The Celotex Corporation maintains an Engineering and Research staff which is available for all types of insulation investigations. Engineers are invited to address their problems to The Celotex Corporation, Chicago, Ill.

Celotex Vapor-seal Insulating Sheathing

A sheathing that has wide acceptance because it provides unusual advantages for use on outside walls under any type of exterior. Designed to meet the advancement of modern building construction. All surfaces and edges moisture proofed with a coating of special asphalt . . . One side additionally treated with a bright aluminum compound as an added vapor-seal — this side is applied facing the studs and interior . . . Coated on the surfaces—not integral—no impregnation—thus maintaining full insulating efficiency . . . Dry Rot and Termite Proofed by the exclusive Ferox Process (patented) . . . Greater rigidity and bracing strength than ever before . . . Backed by Celotex written 10 POINT Life-of-Building Guarantee. Same thickness as the wood sheathing it replaces—1 in.—S2S to 25/32. 4 ft wide; 7 ft, 8 ft, 8½ ft, 9 ft, 9½ ft, 10 ft and 12 ft long.

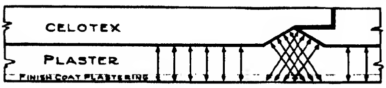


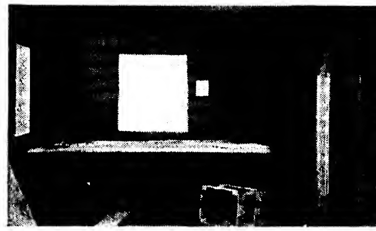
Celotex Vapor-seal Insulating Lath

In combination with Celotex Vapor-seal Insulating Sheathing this insulating plaster base, with a vapor-seal on the warm side of the wall, protects against harmful moisture condensation within the walls. All the superior qualities of Celotex Insulating Lath are retained in this material. Size: 18 in. x 48 in.; ½ in., 1 in. thicknesses.

Celotex Insulating Lath

A natural bond for plaster . . . a continuous surface eliminates lath marks and reduces cracking . . . beveled edges to reinforce plaster (patented) . . . tests prove a bonding power of 1,000 lb per square foot. Shiplapped joints (see diagram). Size: 18 in. x 48 in.; ½ in., ¾ in. and 1 in. thicknesses. Backed by Celotex written 10 point Life-of-Building Guarantee.





Celotex Vaporproofed Low Temperature Insulation

Waterproofed and vaporproofed low density insulation for low temperature requirements. Coolers (beer, meat, creamery, etc.,) Fruit and Vegetable Storage Rooms, Packing Plants, Fur Storage, Air-conditioned Spaces, General Cold Storage Rooms and Freezers. Each block enrobed in a sealed odorless membrane. Ferox Treated. Conductivity 0.30 Btu per inch. Sizes: 12 in. x 36 in., 18 in. x 18 in., 18 in. x 36 in., 9 in. x 36 in. Thicknesses: 1 in. ,1½ in., 2 in. or any multiple of ½ in. up to and including 4 in.

Celotex Building Board

The original cane fibre insulation. Suitable for general use where an insulating board is desired. Can be beveled, paneled, grooved or painted to provide attractive walls that insulate. Neutral tan color. Double surfaced—one side smooth sanded, the other a tapestry-like texture. ½ in. and 1 in. thick. Sizes: 4 ft wide and 4 ft, 5 ft, 6 ft, 7 ft, 8 ft, 8½ ft, 9 ft, 9½ ft, 10 ft and 12 ft long. Backed by Celotex written 10 point Life-of-Building Guarantee.

Celotex Roof Insulation

Selected to insulate wood, concrete, steel, unit tile or poured gypsum roof decks under average conditions. Size 22 in. x 47 in. Full ½ in. thick. Also furnished laminated in thicknesses, 1, 1½, and 2 in. Used to prevent ceiling condensation and conserve fuel; also to prevent excessive expansion and contraction of concrete roof decks. Makes possible a reduction in the size of heating plants.

Celotex Insulating Tile and Finish Plank

Tile Board and Finish Plank—For attractive wall and ceiling treatments . . . may be applied over existing plaster or as a new finish. Neutral tan color or several pleasing tints on reverse side that simplify decoration. Type Double-A joint permits alternated surfaces if desired. Type A, furnished ½ in. and 1 in. thick has beveled square edges; otherwise the same as Type Double-A. Type Double-A joint ½ in. thick only, and 1 in. thick on special order. Sizes range from 12 in. x 12 in. to 24 in. x 48 in. Celotex Finish Planks, ½ in. thick. Sizes: 6 in. to 16 in. wide; 8 ft to 12 ft long. Backed by Celotex written 10 point Life-of-Building Guarantee.

C-X Rock Wool Products

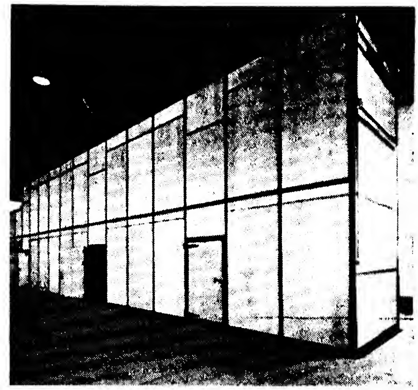
Effective insulation made from molten rock. Incombustible, vermin proof and permanent. Rock Wool Batts 15 in. x 23 in; Prepacked Batts 9 in. x 15 in; Paper Backed Batts 15 in. x 23 in. and 15 in. x 48 in.; Pads 9 in. x 15 in.; Paper Backed 2 in. Batts 15 in. x 23 in. and 15 in. x 48 in. Loose and Granulated to spread between ceiling joists also available.

The Ferox Process

All Celotex Cane Fibre Products are manufactured under the exclusive Ferox Process (patented) and therefore effectively resist damage by Fungus Growth, Dry Rot and Termites (White Ants). It is not a surface treatment—it is integral—insoluble in water—non-volatile—odorless—permanent.

Celotex Cemesto

Celotex Cemesto is composed of genuine Celotex Cane Fibre Insulation Board core with a surface coating of 1/8 in. cement-asbestos adhered to one or both sides with a moisture-proofed adhesive. It forms a rigid board designed to assure permanent insulation and resistance to fire, moisture and weather. It is used for homes, pavilions, cabins, garages, filling stations; vent ducts, conditioning rooms, dyers, kilns, ovens, spandrels; steel frame buildings and hangars. The standard insulation value of Celotex, 0.33 Btu per inch, is maintained in Celotex Cemesto. It is recommended for insulation where the temperature is not constantly in excess of 200 deg—it is not however, a high temperature insulation. The Celotex core in Cemesto is Dry Rot and Termite Proofed by the exclusive (patented) Ferox Process and the hard asbestos-cement coating provides permanent mineral protection which neither rusts nor decays.



A dryer built of 1 in. S2S Cemesto. Bolted to steel framing.

Celotex Cemesto is sawed, drilled, and otherwise handled with ordinary tools. It may be painted if desired or used in its natural light grey finish which provides a smooth surface with excellent light reflecting qualities. Celotex Cemesto is surfaced on one or both sides and comes in sizes 4 ft wide; 4 ft, 6 ft, 8 ft, 10 ft and 12 ft long; 5/8 in., 1 1/8 in., 1 5/8 in. and 2 1/8 in. over-all thickness, when surfaced (S1S) on one side; and 3/4 in., 1/4 in., 1 3/4 in. and 2 1/4 in. over-all thickness, when surfaced (S2S) on both sides.

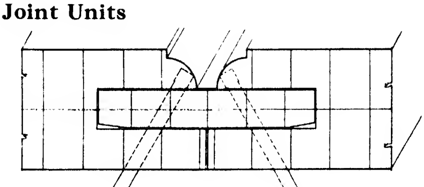
Thermax Structural Insulating Slab

Application of Thermax Structural Insulating Slabs.

A fire-resistant insulating slab suitable for wall sheathing—furring, partitions, ceilings and load-bearing roof decks. Thermal conductivity is 0.458 Btu per hour, per square foot, per degree Fahrenheit, per inch thickness. It provides an excellent plaster base; and when used over large auditorium spaces, the under side is often left exposed for sound absorption treatment. Thermax is manufactured of shredded wood fibres bound together with fire-resistant cement. It is nailed, sawed or set in masonry walls by ordinary mechanics with ordinary tools. Standard sizes: 1 in., 2 in., and 3 in. thicknesses in slabs 20 in. wide x 32, 48, 64, 72 and 96 in. long. Special lengths up to 9 ft can be furnished on request.

Celotex Key Joint Units

New, improved insulating interior finish units, grooved on all edges with splines furnished. These splines form a joint that becomes part of the ceiling and wall decoration. Designed for speedy and low cost application, the units are adaptable to standard framing practices and offer the advantages of less noticeable nailing and self-alignment of units. Paint cracks are less discernable at the joint because of depth of joint and resulting shadow. Cutting and fitting is reduced to a minimum and no special tool work is necessary. The units come in interchangeable sizes, permitting innumerable designs, even on small areas. Sizes: 16 in. x 16 in., 16 in. x 4 ft, 16 in. x 8 ft, 4 ft x 4 ft, 4 ft x 8 ft. Thickness: 3/4 in.



Chamberlin Metal Weather Strip Co., Inc.

General Offices, Detroit, Mich. **Factories, Detroit, Mich., Peru, Ill.**

Factory Sales—Installation Branches

Atlanta, Ga.	Dallas, Texas	Milwaukee, Wis.	Portland, Ore.
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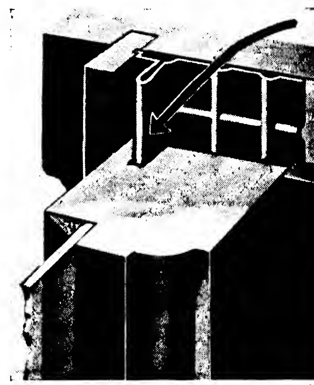
CHAMBERLIN HEAT-SAVING PRODUCTS
Weather Strips, Calking, In-Dor-Seals, Insulation and Insulate-Windows.

Weather Strips

Modern weather strip service, in which engineers and architects can have greatest reliance, is one that can responsibly fulfill conditions of a contract. For problems of infiltration or circulation of air, gases or mixtures, leakage of rain, filtration of dust, sand or soot—there's a Chamberlin Weather Strip remedy regardless of climatic or construction conditions. Particularly helpful to the engineer is the knowledge that the job can be entrusted, regardless of location, to a Chamberlin Factory Branch employing experienced mechanics.

Modern problems of heating, ventilating and air conditioning can be approached with utmost confidence in this field with the aid of Chamberlin's proven 45 years of experience and specialized training. Write for *A.I.A.* catalog of standard details and specifications. Consult nearest branch for equipments, surveys, quotations.



Oldest and most extensively successful weather strip principle—tongue in groove. 80 to 95 per cent efficient. No failure or upkeep.

Calking

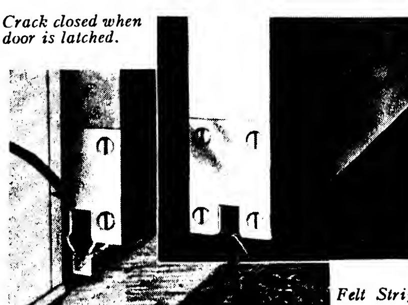
Chamberlin Plasti-Calk is essential in the sealing of construction joints in wood, metal, glass, stone, tile, concrete and brick. It is waterproof, permanently elastic, nonstaining and noncorrosive. It provides durable adhesion and will not sag, pucker, or shrink under extremes of heat or cold, dryness or moisture. Chamberlin Plasti-Calk is specially prepared with porous pigments capable of retaining oil indefinitely, and does not contain tar or asphalt. Supplied in various colors. It is all important that a calking compound be specified thoroughly on the basis of stringent physical properties. Such specifications will be furnished upon request.

Calking applied with hand tools, hand or power gun.

Automatic Door Bottoms

In-Dor-Seals eliminate under-door drafts and heat losses in rooms adjacent to cold areas, corridors and halls—as in a sleeping room door—virtually an "outside" door at night when windows are open. They are used to light-proof X-Ray and dark rooms; to sound-proof private offices, studios, laboratories; to resist room-to-room circulation of odors, fumes and dust.

Crack closed when door is latched.



Felt Strip—automatically raised the instant door opens.

Chamberlin Metal Weather Strip Co., Inc.

General Offices, Detroit, Mich. **Factories, Detroit, Mich., Peru, Ill.**

CHAMBERLIN HEAT-SAVING PRODUCTS

Weather Strips, Calking, In-Dor-Seals, Insulation and Insulate-Windows.

Rock Wool Insulation

Having an average conductance coefficient of 0.24 to 0.27 Btu, Chamberlin Rock Wool is one of the most efficient insulations available today. Long in fibre and clean of "shot", it insures effective insulation with low density and light weight. It is available in several forms adaptable to both new and existing construction—in blowing, commercial or loose wool fibres; in wall-thick or 2 in. thick batts, with or without vapor-proofing coverings. Among its physical properties Chamberlin Rock Wool averages 38 per cent silica, 32 per cent lime and only 0.04 per cent sulphur. Its loss on ignition is less than 0.1 per cent. Wherever possible pneumatic application, by means of such modern high-powered and efficient transient units as illustrated, is preferable. Here again, the engineer will do well to consider the value of faithful, responsible performance in the art of application—the hidden factor that means so much in obtaining the utmost efficiency from insulation.

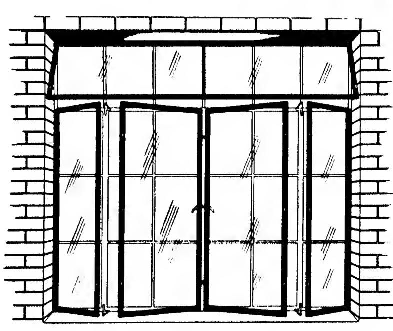
Typical Chamberlin Insulation unit. Equipped with powerful, gas-powered, pneumatic blower capable of projecting granular grade "A" rock wool a minimum distance of more than 200 ft through a large rubber hose.

Insulate-Windows

In addition to other major heat loss controls, Chamberlin has developed a product that reduces 40 to 60 per cent of the heat loss of glass in standard window construction.

Thousands of Chamberlin Insulate-Window (registered tradename) units have been applied to wood and metal windows in existing buildings, including offices, banks, residences, etc.—they are *not* double glazed units. They are single glazed and custom made *to supplement* existing windows of practically any type. Depending upon characteristics of the areas and sash to be "insulated," Insulate-Windows are hinged, stationary or sliding. They are framed with a patented rolled section of antique-finished, cold-rolled bronze embodying ingenious features of mechanical glazing in place of putty. Chamberlin "insulation" of windows is a high grade, mechanical service involving design and installation of durable, secondary window units, readily accessible for cleaning and removal when necessary—no interference with sash operation or ventilation. Insulate-Windows provide a year 'round heat stop that gives practical relief without the technical difficulties involved in double-glazed windows.

At right—near view of "insulated" residential metal casement windows. Chamberlin Insulate-Windows on outside. Below—battery of Insulate-Windows showing how they can be hinged. All 5 units can be unlatched from within the room by simply opening the two casement ventilators.



See Chamberlin exhibits in The Home Building Products Building at the New York World's Fair.

The Philip Carey Company

Manufacturers of Heat Insulation and Asbestos Products

Lockland **Cincinnati, Ohio**

Sales Offices

Atlanta, Ga.
Baltimore, Md.
Boston, Mass.
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Norfolk, Va.
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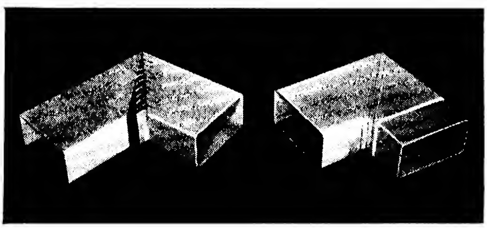
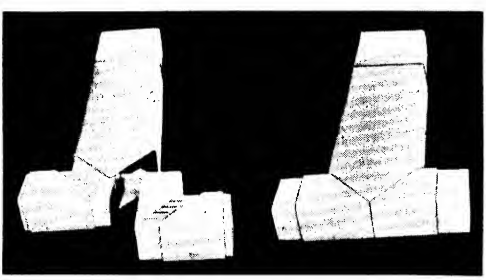
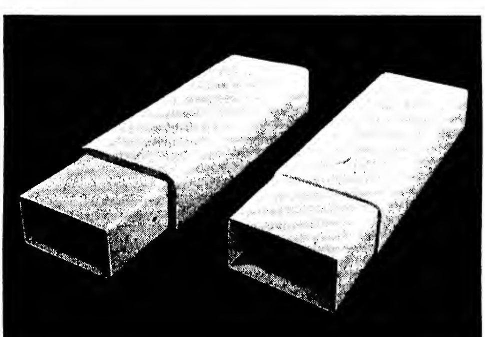
Wheeling, W. Va.



Standard 90 deg Elbow assembly. Left: Core opened to show duct vanes. Right: The completed fitting.



Two standard 90 deg Elbows nested in a larger standard section to form a tee.



Standard 1 in. and ½ in. thick Careyduct sections with core extended.

Careyduct is a new prefabricated insulated duct built entirely of asbestos. The double layer construction consists of an inner core of hard, rigid asbestos, and the outer jacket is made of multiple layers of a fine corrugated asbestos structure. The combination results in great strength, is an excellent insulator, and has a definite sound deadening effect.

Careyduct fittings are made from standard sections of duct, and may be made in the field with comparative ease by men without special training. A simple mitre cut plus a few standard accessories make a complete fitting thus keeping costs at a minimum. Prefabricated fittings may be ordered from the factory if desired.

The telescopic assembly method practically eliminates leaks that are commonly found in other construction.

The standard sizes of Careyduct are designed so that a combination of smaller sizes will exactly nest in a larger size. All tees and take-offs are a combination of ells and straight duct.

Grilles and dampers are installed according to the accepted standard practice. Careyduct gives high insulating value. It materially reduces the transmission of extraneous and equipment noises. Careyduct costs decidedly less than properly insulated metal duct and compares very favorably with sheet metal duct of standard quality.

For more detailed information and prices, write for illustrated catalog.

The Eagle-Picher Lead Company

General Offices: Temple Bar Building, Cincinnati, Ohio Offices in All Large Cities

EAGLE HOME INSULATION

Eagle Insulation for homes is a fluffy, woolly material spun from mineral rock. In both granulated form for pneumatic application and in batt form, it is extremely lightweight, non-corrosive, fireproof.

According to U.S. Bureau of Standards tests, Eagle Insulation in applied thickness of 3⅝ in. has a thermal conductivity rating of only 0.074 Btu at 103 F. (Overall conductivity is considerably lower.)

Fuel Savings Up to 40 Per Cent

Eagle Insulation literally "seals" furnace heat inside the home. Records show that fuel savings run up to 40 per cent of the season's total. In summer, Eagle Insulation keeps homes up to 15 deg cooler than outdoor temperatures.

Reduces Fire Hazard

Unlike many insulating materials which are described as "fire-safe" or "fire-resistant", Eagle Insulation is fireproof to 1200 F. It will not burn. Fire hazard is greatly reduced. Hollow spaces in walls, which ordinarily act as flues once a fire starts, are completely filled with a material that even a blow torch cannot ignite.

Eagle Insulating Wool easily blown into spaces between wall studdings

The Bureau of Buildings of the City of New York has given full approval to use of Eagle Insulation where fire-retarding is required under the Building Code and the Multiple Dwelling Law. Also approved by Underwriters' Laboratories, Inc. as being a non-conductor of electricity.

Water Repellent

Samples of Eagle Insulation exposed to completely saturated atmosphere for 100 hours gained less than 0.2 per cent in weight due to moisture absorption. This is important. Insulating materials which absorb water soon lose much of their insulating efficiency.

Being made of mineral wool, Eagle Insulation contains no vegetable matter to attract insects or pests. It is rotproof —will not deteriorate. It passes the most severe settling tests.

Type H-1. Applied Pneumatically

In granulated form, Eagle Insulation is blown into hollow spaces between wall studdings and between joists in the attic floor by a special pneumatic process. No building alterations are necessary, whether the structure is of frame, brick veneer or stucco construction. Work is done by skilled authorized contractors.

Type H-3. Batts for New Construction

Eagle Insulating Batts are rectangular pads in suitable sizes to fit snugly between studding and joists. Equipped with a waterproofed paper back. Batts are easily cut to fit irregular spaces when necessary.

Eagle wall-thick batts quickly installed in new construction

Data and Specifications

For complete specifications and technical data on Eagle Home Insulation, see *Sweet's Architectural Catalog.*

EAGLE INDUSTRIAL INSULATION

The Eagle-Picher Lead Company manufactures a varied line of insulating products effective for a full range of temperatures.

Eagle Super "66" Plastic Insulation provides remarkable heat-saving efficiency for temperatures as high as 1800 F. Trowels easily on all large or irregular surfaces. 100 lb give 50-55 sq ft 1 in. thick dry coverage.

Eagle Blanket Insulation is Eagle Insulating Wool felted and secured between metal fabrics. Easy to cut and fit.

Other products are pipe insulation, roll and felted wools for air conditioning appurtenances and other industrial uses, insulating cements and loose wool.

For specifications and technical data on Eagle Industrial Insulation, see *Sweet's Engineering* or *Power Plant Catalog.*

Ehret Magnesia Manufacturing Co.

Valley Forge, Pa.

EHRET
INSULATIONS

DURANT SYSTEM

Hermetically Sealed Insulated Pipe For Underground and Outdoor Use

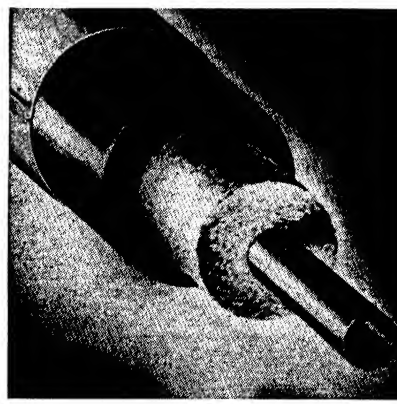
Ehret's DURANT SYSTEM insures unquestionable protection against moisture and soil conditions.

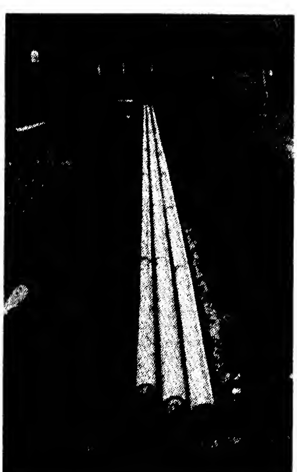
Even if submerged, no water can ever penetrate through to the insulation.

Since no foundations, underdrains, etc. are required, the installation is simply and quickly accomplished.

Ehret's DURANT SYSTEM is a factory made product and is shipped complete with the pipe, in special or mill lengths.

Any size and specification of pipe or tubing can be used.

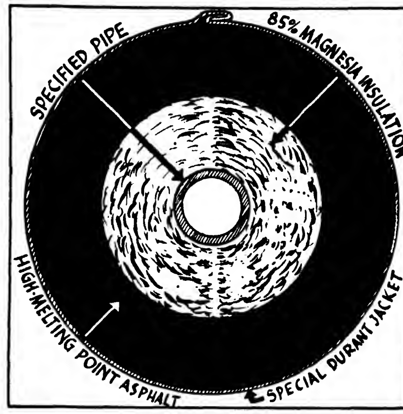




Many large installations of Ehret's DURANT SYSTEM are in successful use by Federal Housing Projects, United States Navy and Coast Guard, and other Government departments, as well as by numerous railroads, public utilities and industrial concerns. A list of these installations will be furnished upon request.

A pure, high melting point asphalt is melted and cast around the insulated pipe, using an accurately spaced metal jacket as the mould. This results in a continuous, seamless and jointless protection which is exceedingly durable. The thickness of the asphalt is usually one in. for underground lines and ½ in. for outdoor lines.

The insulation consists of Ehret's 85% Magnesia which is the most dependable and efficient insulation material in its field. The thickness will depend upon the conditions. It is practical for both low, or cold temperature lines, as well as for heat piping.



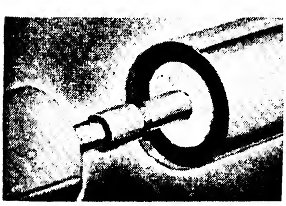
Insulation loses its efficiency if moist or wet and the heat losses are greatly increased by outdoor air currents. Permanent and dependable protection from underground soil conditions and outdoor weather is most important. The absolutely impervious nature of Asphalt, surrounding the insulated pipe in a solid casing, thoroughly seals it against moisture or air infiltration from one end to the other.

The asphalt protection will not crack from the pressure of the soil and it is sufficiently ductile to follow any possible movements of the pipe line. There are no fitted-together and cemented joints which might crack open, if the piping is forced out of line, permitting water to seep in.

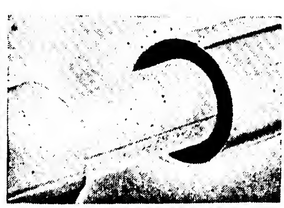
Since each pipe line is separately treated, it can be used as individual units. In the event that it becomes necessary to make changes in any line, such as replacing worn out return piping, the work can be done without disturbing any parallel piping.

Also, in case that any pipe must be replaced by identical sizes, all that it is necessary to do is to tack with the welding apparatus a new piece of pipe to the old piece to be removed, pull the old piece out of the insulation and the new piping will be inserted and ready to be installed in the original line. The only new insulation and waterproofing necessary, will be where the new pipe is joined to the original.

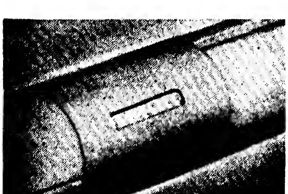
Since the DURANT SYSTEM will not leak, even if placed in standing water, there need be no concern regarding the type of soil it is to be buried in. Back filling of the trench can be effected in one operation as the loose soil can be settled down into place by flooding the trench with water.



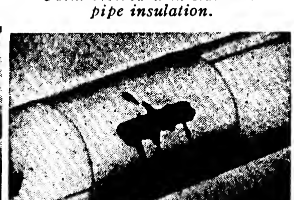
STEP ONE
Field joint ready for inspection.



STEP TWO
Joint covered with standard pipe insulation.



STEP THREE
Special Durant joint casing in place ready for Asphalt.



STEP FOUR
Asphalt poured in slot—a perfect seal.

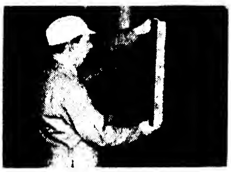
Complete installations are quickly made and field joints and connections are easily and perfectly sealed.

As shown in the illustrations at the left, the method of treating joint and fitting connections actually extends the asphalt casing along the entire pipe system in an absolutely seamless manner. The heat of the melted asphalt poured around the joints thoroughly fuses into the asphalt cast around the insulation.

General Insulating & Mfg. Company

Engineering Offices and Main Plant: ALEXANDRIA, INDIANA | Executive Offices: ST. LOUIS, MISSOURI | Branch Plants: DOVER, N. J. DUBUQUE, IOWA

ROCK WOOL Gimco INSULATION



Gimco Sealal Bats, furnished with waterproof paper backing, fit between standard studdings and joists.

Gimco Sealal Rock Wool Bats—Gimco Rock Wool Bats are made from long, tough rock wool fibres annealed and treated specially by the patented Gimco process. Installed 3⅝ in. thick Gimco's conductivity is only 0.067 Btu per hour per sq ft for that thickness. Gimco provides full "wall-thick" protection . . . keeps inside temperatures as much as 15 deg cooler in summer and pays for itself out of winter fuel savings. Gimco is as fireproof as the rock itself, resists moisture, and will not decay, pack down or dust out. Gimco is as permanent as the house, and offers no attraction to vermin or termites.

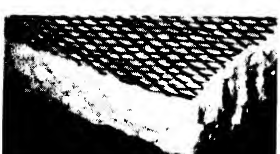
Gimco Bats are Self-Supporting—Gimco bats need only to be pushed between studdings or joists. Their own natural resiliency holds them permanently in place without additional support. Application costs are thus cut to a minimum.

Gimco Insulation for Present Homes—Gimco in granulated form can easily be blown into empty wall and ceiling spaces. It makes a permanent "wall-thick" insulation, and can be installed in any home, regardless of age, size or type of construction. For complete details, write for new book on home insulation.

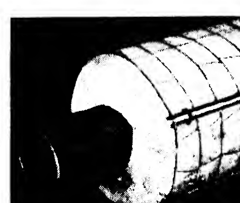
Specifications—Gimco Sealal Bats are furnished with waterproofed paper backing. They are made in three sizes: (1) 15 x 18 in. x wall thick; (2) 15 x 23 in. x wall thick; (3) 15 x 48 in. x wall thick. Ten small bats insulate approximately 20 sq ft of wall or ceiling area; 10 medium size bats insulate approximately 25½ sq ft; 10 large bats insulate approximately 55 sq ft.

Present homes are easily and quickly insulated by blowing Gimco Rock Wool in empty wall and ceiling spaces. Insures a thick, protective blanket of uniform density.

GIMCO ROCK WOOL PRODUCTS FOR INDUSTRY



Flexfelt Blankets—Efficient and adaptable rock wool insulation for boilers, heaters, furnace breachings, hot tanks and other industrial equipment. Standard sizes: 2 ft x 4 ft and 2 ft x 8 ft. Special sizes are also made to order or as planned by our engineers from purchasers' equipment drawings.



Gimco Flexfelt Pipe Coverings—Flexfelt coverings are suited for every type of pipe insulation . . . hot water, steam process, cold water and refrigerant lines. They are made in a variety of sizes and styles, each individually designed to do an efficient insulating job.

Gimco No. 330 Insulating Cement can be easily and quickly applied to any surface. It dries quickly with little shrinkage and has a smooth, hard surface. Cost per square foot coverage is unusually low.

Free Construction and Engineering Service—Gimco engineers, backed by years of research in our own laboratories, render prompt and efficient service in helping you solve insulation problems. Quotations and suggestions on any job (temperatures up to 1500 F) will be gladly given.

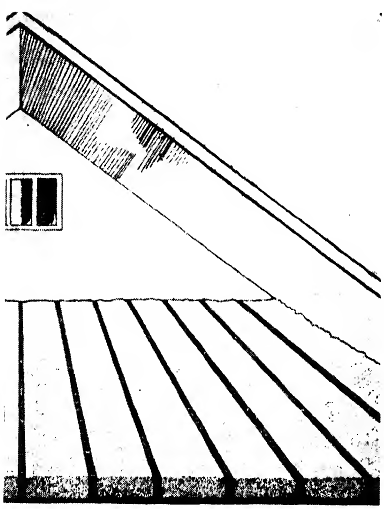
Insul-Wool Insulation Corp.

General Offices, **Wichita, Kansas**

Branches in Principal Cities

Manufacturers and Distributors of Insul-Wool

Insul-Wool is a fibre insulation of the "fill" type—made of wood pulp, a natural insulating material. By the exclusive "Insul-Wool" method the wood pulp is converted into a loose fluffy substance which, when installed in a building, forms a soft heat-resisting blanket having millions of tiny air cells capable of resisting passage of either heat or cold.



Insul-Wool Applied over Ceiling

UNIFORMITY OF PRODUCT

Only one grade of Insul-Wool is made and every "run" is tested at the factory to insure uniformity of product and unvarying high quality. It is free from grit, silicon particles, or "shot."

FIRE PROOF AND VERMIN PROOF

A special "Insul-Wool" method of chemical treatment makes Insul-Wool thoroughly vermin proof and fire proof. It has been approved by the National Board of Fire Underwriters.

"INSUL-WOOL" SERVICE

Insul-Wool is distributed and installed only by specially trained men—direct factory representatives or men in the organizations of the largest insulation material dealers throughout the United States.

ADVANTAGES OF INSUL-WOOL

1. It is made from wood pulp, a natural insulating material.
2. Chemical treatment makes Insul-Wool safe under all conditions and hazards.
3. Approved by the National Board of Fire Underwriters.
4. Its light weight of 2.5 lb to the cubic foot adds very little load to the ceiling rafters.
5. Does not pack or settle and outlasts the building in which it is installed.
6. Does not draw moisture.
7. Cuts fuel costs and reduces Summer temperatures, indoors.
8. Meets U. S. Government requirements on Federal Construction with a thermal conductivity of 0.24 Btu per hour, per square foot, per degree Fahrenheit, per inch thickness.

Analysis of Insul-Wool in Terms of Commercial Thickness

Material	Commercial Form	Comm'l Thickness Inches	D. Wt. Per cu. ft.	C. Conductivity
INSUL-WOOL	Wood Fiber-Loose Type, Fire proofed and Germ proofed.	1 4	2.5	0.24* 0.067**

*Kansas City Testing Laboratory, Inc., February 25, 1938.
**J. C. Peebles, Armour Institute of Technology, April 8, 1937.

Complete data on Insul-Wool Insulating Product will be sent upon request.

The Insulite Company

General Offices

1100 Builders Exchange, **Minneapolis, Minnesota**

Sales Solicitors Offices

New York—101 Park Avenue
Chicago—205 W. Wacker Drive

San Francisco —475 Brannan St.
St. Louis—1206 S. Vandeventer

TWENTY-FIVE YEARS Silver Jubilee **PROVED DURABILITY**

1939 marks the 25th Anniversary of The Insulite Company.

For 25 years engineers and architects have specified Insulite materials in walls, ceilings, floors, roofs, and for duct lining to achieve insulation, acoustics, and sound control. Insulite materials have proved themselves practical through their performance on the job.

You are invited to consult any Insulite office for cooperation on problems concerning insulation, acoustics, and sound control

Modern Materials for Modern Building

Ins-Lite—Natural surface wood fiber board, the color of natural wood. Burlap texture one surface—linen the other. Thermal conductivity 0 33 Btu/hour/square foot/inch/F, based on a density of 16 lb/cu ft. Thickness—½ in. for following sizes· 4 to 8½ ft wide by 6 to 14 ft long with square edges and also 4 x 6 ft up to 4 x 14 ft beveled two long edges of linen textured side Thicknesses ½ and ¾ in for following sizes: 12 x 12 in to 18 x 48 in. with V-W joint on four edges and 6 to 16 in. wide by 8 to 12 ft long with V-W joint, beaded, on long edges

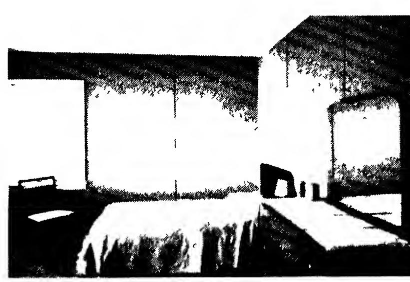
Used on interior walls and ceilings, also as sub-flooring.

Graylite—Asphalt-containing wood fiber board with natural surface and neutral gray-brown color Burlap texture one surface—linen the other Thermal conductivity 0 35 Btu per inch thickness Available in same thicknesses, sizes, and joint fabrication as Ins-Lite Used on interior walls and ceilings, also as a subflooring.

Satincote—A wood fiber board with primed and sealed surface in a rich light color The surface is smooth, hard and durable and offers resistance to abrasion. The sealed surface receives paint finishes without further treatment but may be used in its natural finish if desired Thermal conductivity the same as for Ins-Lite Thickness ½ in. Sizes 4 x 6 ft up to 4 x 14 ft are available either square edged or beveled on long edges of finished surface. Sizes 12 x 12 in. up to 18 x 48 in. are furnished with V-W joint all edges Following sizes are furnished with V-W joint—beaded, long edges: widths 6, 8, 10, 12, and 16 in.—lengths 8, 9, 10, and 12 ft. Used on interior walls and ceilings

Smoothcote—A pre-finished wood fiber board with a smooth, hard surface in a pleasing light cream color. No decorative treatment is required over the surface, however, if desired, the board may be painted. Available in same thickness, sizes, and joint fabrication as Satincote Thermal conductivity same as Ins-Lite Used on interior walls and ceilings.

Lok-Joint Lath—An insulating plaster base, fabricated from Ins-Lite Has a patented "lok" that firmly locks the sheets together between supporting members Thicknesses ½ in , ¾ in , and 1 in. Size: 18 in x 48 in Used as a plaster base for walls and ceilings



Insulite for Interior Walls and Ceilings

Bildrite Sheathing

Bildrite Sheathing is an asphalt-containing wood fiber board manufactured under an exclusive process which provides increased strength and moisture resistance. It is 25/32 in. thick and has a distinctive gray-brown color. Thermal conductivity: 0.36 Btu per inch thickness. Each sheet is marked to indicate proper nail spacing Available in sizes 4 x 8 ft up to 4 x 12 ft. Used as a structural sheathing board and as a roof boarding.

INSULITE WALL OF PROTECTION*

Coefficients of Transmission (U) of various types of frame construction.

Exterior Finish and Sheathing	Interior Finish: No Insulation Between Studding		Interior Finish: Insulation Between Studding
	Plaster (1/2 in.) on plaster board (3/8 in.)	Plaster (1/2 in.) on Lok-Joint Lath (1/2 in.)*	Plaster (1/2 in.) on Lok-Joint Lath (1/2 in.) Wall-Seal Batts (2 in.) between studding—1 air space*
Wood Siding $^{25}/_{32}$ in. Bildrite Sheathing	0.20	0.16	0.074

The above values are typical of results which can be obtained by utilizing Insulite materials in frame construction. For further (U) values refer to Chapter 5, pages 106 and 107.

Ins-Lite Cold Storage Insulation

Fabricated from special low density wood fiber board, 12 lb per cu ft. Average thermal conductivity is 0.30 Btu per inch thickness. Available in eleven standard sizes of 12 x 18 in. up to 24 x 48 in. Thicknesses: 1, 1½, 2, 3, and 4 in. For use as insulation on walls, floors, and ceilings of ice houses, cooler rooms, storage plants, breweries, and wherever low temperature control is necessary.

Sealdslab Cold Storage Insulation

Fabricated from the same low density material as Ins-Lite Cold Storage Insulation; however, the board is given an additional treatment consisting of a continuous impregnation of especially prepared asphalt to a uniform depth of 3/32 in. on all faces and edges which, with the addition of the final asphalt coating applied on the job, provides a superior seal against moisture and vapor. Available in same sizes and thicknesses as Ins-Lite Cold Storage Insulation. Used as insulation on walls, ceilings, and floors of ice storage plants, cooler and freezer rooms, and wherever low temperatures are maintained.

Ins-Lite Roof Insulation

Fabricated from wood fiber board. Size: 22 x 47 in. with either offset or square edges. Thicknesses: ½, 1, 1½, and 2 in. Thermal conductivity: 0.32 Btu per inch thickness. Used as insulation on all types of roof decks under built-up roofing.

Graylite Roof Insulation

An asphalt-containing wood fiber board of gray-brown color. The asphalt treatment not only provides added strength and moisture resistance to the board but also a better natural bond with the bitumen used on the job. Available in the same thicknesses, size, and edge treatments as Ins-Lite Roof Insulation. Thermal conductivity: 0.35 Btu per inch thickness. Used as insulation for all types of roof decks under built-up roofing.

Applying Insulite Roof Insulation

Insulite Fiberock

Insulite Fiberock Insulation is a rock wool product, treated for moisture resistance. It has a low percentage of shot and sufficient resiliency to prevent it from settling. Thermal conductivity: 0.26 Btu per inch thickness. Used as insulation in the intermediate spaces in walls, ceilings, floors, and roofs.

Available in six forms: Loose Fill, for hand packing; Granulated Fill, for pouring into place; Batts, thicknesses 2 and 3⅝ in.; Wall-Seal Batts, thicknesses 2 in. and wall-thick; Pads, wall thick; and Pre-packed, roughly 2 in. thick.

The Batts are 15 x 23 in. and 15 x 48 in. in size, and the Wall-seal Batt is lined on one side with a waterproofed, vapor-proofed kraft paper, flanged on all four edges. Pads and Pre-packed forms are 15 x 9 in. in size.

Hard Pressed Boards

Bronzelite materials are tough, durable, grainless, pressed wood fiber boards with a hard, smooth surface. Available in a range of densities, from 37 to 68 lb per cu ft, and degrees of hardness. Thicknesses are from 1/10 to 5/16 in. and sizes from 4 x 2 ft to 4 x 12 ft.

International Fibre Board Limited

Sales Offices
OTTAWA—MONTREAL—TORONTO—WINNIPEG
Administrative Offices and Mills: GATINEAU, QUE

London Office
THE TENTEST FIBRE BOARD CO. LTD.
ASTOR HOUSE, ALDWYCH, LONDON, W. C. 2., ENGLAND

TEN/TEST
INSULATING WALL BOARD

TEN/TEST is a manufactured lumber made from spruce fibres, solidly pressed under hydraulic pressure into a strong, homogeneous board. The fibres are chemically treated and water-proofed during process of manufacture, until the insulation is non-hygroscopic, free from capillary attraction and moisture-resisting in service commensurate with the maximum degree of insulation obtainable.

Official Tests

Conductivity. TEN/TEST has a conductivity of 0.33 Btu per hour per square foot per degree fahr. per 1 in. thick. Authority: Professor E. A. Allcut, M. Sc M. I. Mech. E Mem. A.S.M E. Professor of Applied Mechanics, University of Toronto. Tests performed by Hot-Plate method. Mean temperature 47.8 deg.

Tensile Strength 228 lb per sq in. Tests made on 7/16 in. board cut to strips 1 in. wide and tested in a Riehle Tensile Testing Machine, the grips being 2 in. apart. 228 lb is the mean average of seven series of tests.

Transverse Strength (equal deflection) is 28.4 lb. Test made on 7/16 in. board, 6 in. wide, 18 in. long, on 12 in. centers, and load being applied to breaking point.

Plaster Bonding Strength 2163 lb per sq ft. Brown and scratch plaster coats were applied to standard 7/16 in. board, and the pull registered in an Olsen Testing Machine. Authority: Columbia University Testing Laboratories, New York.

Moisture Resisting. TEN/TEST, after complete immersion in water for 24 hours, registered 37.5% increase in weight.

NOTE.—Authority for tensile strength, transverse and moisture tests; J. T. Donald & Co., Ltd., Chemical Analysts and Engineers, Montreal, Que.

TEN/TEST Products

TEN/TEST Insulating Building Board. Standard insulation for use as exterior sheathing, interior finish, between walls and under floors for sound deadening. Standard Industrial Insulation for refrigeration and the prevention of condensation. Manufactured in convenient sizes: 4 ft wide and up to 17 ft long, ½ in to 1 in thick or laminated to any desired thickness.

TEN/TEST Notch Board Plaster Base. Insulating plaster base having tongue and groove interlocking joints. Provides an effective bond with plaster without use of metal lath at joints. Sizes· 16 in. wide; 32 in and 47¾ in long. Thicknesses from ½ in. to 1 in. or laminated to any desired thickness

TEN/TEST Roof Board. An effective roof insulation. Manufactured in two sizes: 1 x 4 ft and 2 x 4 ft Thicknesses from ½ in to 1 in or laminated to any desired thickness

TEN/TEST Ashlar Block. For interior decoration and acoustical correction Can be supplied in a variety of designs and sizes to harmonize with any decorative treatment

TEN/TEST Acoustical Tile and Panels with sound absorption coefficients ranging up to 0 53 at 512 Specially designed for churches, schools, auditoriums, theatres, etc.

TEN/TEST Moulded and Shiplap Edge Wall Panels. Conceals joints and provides excellent decorative treatment. Featured in widths of 11 in. to 47¼ in, lengths up to 12 ft.

TEN/TEST Mouldings. An effective trim and finish for joints, corners, etc. Available in widths of ¾ in. to 10 in. and lengths up to 12 ft.

HYDRO/TEST. Water proof, insulating building board, designed particularly for low temperature requirements.

Mundet Cork Corporation

65 S. Eleventh St. INSULATION DIVISION **Brooklyn, N. Y.**

Manufacturers of Corkboard, Cork Pipe Covering, Compressed Machinery Isolation Cork, Natural Cork Isolation Mats, Cork Tile, Cork Bulletin Board, and all kinds and varieties of Cork Specialties.

Mundet Branches

Albany, N. Y.	Dallas, Texas	Los Angeles, Calif.	Philadelphia, Pa.
Atlanta, Ga.	Detroit, Mich.	Memphis, Tenn.	St. Louis, Mo.
Chicago, Ill.	Houston, Texas	New Orleans, La.	San Francisco, Calif.
Cincinnati, Ohio	Kansas City, Mo.	No. Cambridge (Boston), Mass.	Syracuse, N. Y.

Mundet Agents

Baltimore, Md.	The McCormick Asbestos Co.	Norfolk, Va.	F. H. Gaskins Corp.
Buffalo, N. Y.	Claxton Asbestos Co.	Oklahoma City, Okla.	Standard Roofing & Material Co.
Charlotte, N. C.	Geo. A. White & Co.	Portland, Oregon	Pacific Asbestos & Supply Co.
Cleveland, Ohio	Standard Asbestos Mfg. Co.	Richmond, Va.	Virginia Insulation Co.
Richmond, Va.	Virginia Insulation Co.	Salt Lake City, Utah	Louis A. Roser
Hartford, Conn.	The Hartford Cement Co.	Seattle, Wash.	Pioneer Sand & Gravel Co.
Minneapolis, Minn.	Asbestos Building Materials Co.	Tulsa, Okla.	Standard Roofing & Materials Co.
Nashville, Tenn.	John Bouchard & Sons Co.	Utica, N. Y.	George Weisenberger

Engineering and Specification Service

Our engineering department is at the service of Architects and Engineers at all times to assist and advise in the preparation of specifications pertaining to cork. This service is also available to any one who has a cold insulation or a vibration isolation problem, and is rendered without obligation. Our complete catalogue is filed in Sweet's Architectural Catalogue, and will be sent on request. It is replete with valuable information and data that should always be within reach of every specification writer whose field touches our products.

Mundet Contract Service

We contract for the erection of our products. In this way we may be certain that our material is installed in accordance with best established practice. This gives a definite advantage to an owner, in that divided responsibility for a given installation is eliminated. No contract involving cork is too large, too small or too far away. All materials and workmanship are unqualifiedly guaranteed.

Mundet "Jointite" Corkboard

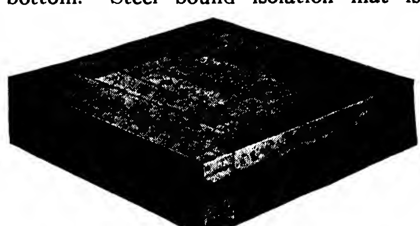
Mundet "Jointite" Corkboard is 100 per cent pure cork, fabricated in accordance with the U. S. Government Master Specification, and is unsurpassed in its field. It is used for all cold insulation services and for acoustical correction. We manufacture only one grade of corkboard. Mundet "Jointite" Corkboard is sold in the standard 12 in. x 36 in. sheet. Standard thicknesses are ½ in., 1 in., 1½ in., 2 in., 3 in., 4 in. and 6 in.

Mundet "Jointite" Cork Pipe Covering

Mundet "Jointite" Cork Pipe Covering is the complement of Mundet "Jointite" Corkboard and is used for all types of cold lines. The three thicknesses in which it is manufactured make it suitable for pipes carrying sub-zero to 50 F temperature. The pipe covering comes in sections 36 in., long. A complete line of standard covers is available in the three thicknesses.

Mundet Cork Vibration Isolation

The transmission of machine vibration can be easily and permanently prevented by the use of Mundet cork isolation. The machines commonly associated with the heating and ventilating industry are best isolated with Mundet Natural Cork Isolation Mat. This form of isolation is fabricated from blocks of pure cork. These blocks are held together within a rigid steel frame or bound with asphalt paper applied with hot asphalt top and bottom. Steel bound isolation mat is



Above is shown a Steel Bound Mundet Natural Cork Isolation Mat. Note the natural cork strips within the steel frame.

usually used under exposed mounts, and asphalt paper bound isolation mat, under concrete foundations, of the envelope type. Mats are constructed to fit under any type of machine foundation.

For loads exceeding 2,000 lb per square foot, we manufacture Mundet Machinery Isolation Cork, which is a board form of compressed granulated cork and comes in three densities.

Both types of isolation are furnished in 1 in., 1½ in., 2 in., 3 in., 4 in. and 6 in. thicknesses, depending on class of service.

Johns-Manville

Executive Offices: 22 East 40th Street, **New York, N. Y.**

Offices in All Large Cities

Johns-Manville Home Insulation

Johns-Manville Home Insulation is a light, fluffy mineral wool, highly efficient in heat-proofing practically any building, old or new. It is durable, rot-proof, fire-proof and odorless, and will not corrode or settle. Full stud thickness of this material will cut fuel costs up to 30 per cent in winter, and, help keep rooms up to 15 deg cooler in hottest weather. J-M Home Insulation is furnished in two forms: for new construction, in easily handled batts; for existing buildings, in loose, nodulated form to be installed pneumatically.

Applying J-M Type B batts in new home

For New Construction
Types B, C and Junior Batts

Type B Home Insulation is furnished in pre-fabricated batts of uniform density, in both full stud thickness and semi-thick, in sizes 15 x 23 in. and 15 x 48 in., designed to fill completely the space between studs, joists and rafters on the usual 16 in. centers. Type B batts are backed with waterproof, vapor-resistant paper, extending on the long sides in a $1\frac{1}{2}$ in. wide flange, by which the batt is fastened in place and which also aids in sealing the joints. This backing protects against penetration of moisture from wet plaster and also resists infiltration of moisture vapor from the house into the wall.

Junior Batts are similar to Type B batts except that they are not paper-backed and are furnished only in size 12 x 15 in. by full stud thickness.

Type C Home Insulation is an improved form of loose wool, in pieces 8 x 15 in., without paper backing, which readily fluffs to full wall thickness when installed.

Both Junior Batts and Type C can be readily installed in irregular spaces since they are easily cut or torn with the hands.

For Existing Homes and Buildings
Type A "Blown" Rock Wool

Type A Rock Wool is blown pneumatically into the spaces between studs in outer walls and between rafters or joists in roofs or attic floors. Insulation thickness in walls corresponds to stud depth, approximately $3\frac{5}{8}$ in.; density does not exceed 10 lb per cubic foot. This type of insulation is installed only by Approved J-M Home Insulation Contractors, who are equipped with the necessary apparatus and trained crews.

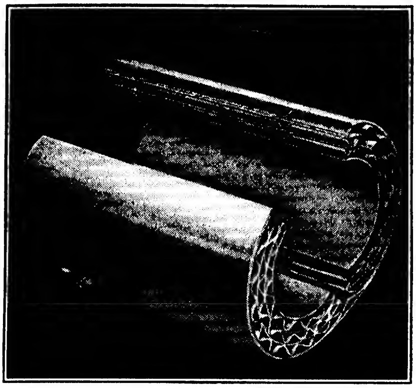
Write for Details

Complete information on all types of J-M Home Insulation will be furnished on request.

J-M Airacoustic Sheets for lining Air-Conditioning Ducts

J-M Airacoustic Sheets, for duct linings of air conditioning systems, are rigid, fire-proof, highly sound-absorbent and moisture-resistant, with a surface which will not materially increase friction losses in the duct system. Write for DS Series 275.

Johns-Manville Pipe and Boiler Insulation



J-M Pre-Shrunk Asbestocel Pipe Insulation

J-M Pre-Shrunk Asbestocel Pipe Insulation

J-M Pre-Shrunk Asbestocel is a radically improved insulating material for hot water or low pressure steam piping, which, since it is made of moisture-proofed asbestos paper, minimizes objectionable shrinkage.

Supplied in canvas, asbestos paper or aluminum finishes. All types furnished in 3-ft sections in standard thicknesses of 2 to 8 plies, each ply approximately ¼ in. thick, for all commercial pipe sizes.*

J-M 85% Magnesia

Recommended as the most efficient insulation of the molded type for temperatures up to 600 F. Pipe insulation is furnished in sectional or segmental form for all commercial pipe sizes, in thicknesses up to 3 in. Blocks are 3 in. by 18 in. and 6 in. by 36 in., flat or curved, from ½ in. to 4 in. thick, for all commercial pipe sizes.*

J-M Pre-Shrunk Wool Felt Pipe Insulation

Due to its Dual-Service Liner—an asphalt-saturated felt—J-M Pre-Shrunk Wool Felt is equally effective and durable on either hot or cold water service piping. By the use of waterproofed felts, shrinkage troubles have been minimized.

Supplied in two finishes, the regular canvas and a smooth, dull-coated aluminum. In either finish, it is furnished in 3-ft sections in thicknesses of ½ in. ¾ in., 1 in., Double ½ in., and Double ¾ in., for all commercial pipe sizes.*

J-M Asbesto-Sponge Felted Pipe Insulation

Recommended on all high pressure steam piping at temperatures up to 700 F where insulation may be subjected to rough usage or where maximum efficiency and durability are desired. Furnished in 3-ft sections up to 3 in. thick, for all commercial pipe sizes.

J-M Superex Combination

Superex Combination Insulation (an inner layer of high temperature Superex and an outer layer of 85% Magnesia) is recommended where temperatures exceed 600 F. Superex and Magnesia are both furnished in sectional and segmental pipe covering, and in block forms.

J-M Asbestocel Sheets and Blocks

Asbestocel Sheets and Blocks are used for insulating warm-air ducts, flues, heater casings and fan housings in the ventilating system. Temperature limit 300 F. Furnished 6, 9, 12, 18 and 36 in. wide by 36 and 72 in. long, from ½ in. to 4 in. thick.

J-M Rock Cork Sheets and Pipe Insulation

J-M Rock Cork is made of mineral wool and a moisture-proof binding ingredient molded into sheets for insulating refrigerated rooms and air conditioning ducts; and into sectional pipe insulation with an integral waterproof jacket, for all low temperature service. It is strong, durable, and will not support vermin. Because of its unusual moisture resistance, its high insulating efficiency is maintained in service.

Furnished in sheets 18 in. by 36 in., in 1½, 2, 3 and 4 in. thicknesses; also 18 in. by 18 in. by 1 in. thick. In pipe covering form, in ice water, brine and heavy brine thicknesses, for all commercial pipe sizes.

Details on Request

Write for complete information on any Johns-Manville insulating material.

*J-M Magnesia, Pre-Shrunk Asbestocel and Pre-Shrunk Wool Felt can also be supplied in sections to fit straight runs of copper pipe or tubing with the following outside diameters: ⅜ in., ½ in., ⅝ in., ⅞ in., 1⅛ in., 1⅜ in., 1⅝ in., 2⅛ in., 2⅝ in., 3⅛ in., 3⅝ in., 4⅛ in., 5⅛ in., and 6⅛ in.

The Ruberoid Co.

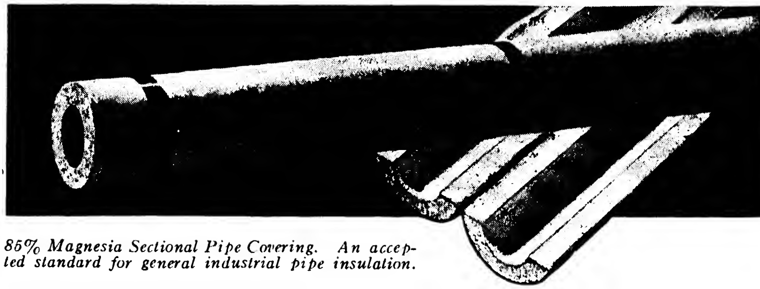
INSULATING PRODUCTS

Executive Offices

500 Fifth Avenue, **New York, N. Y.**

Divisional Offices

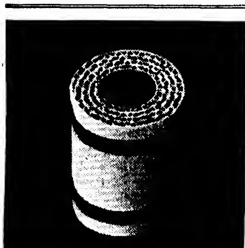
NEW YORK — CHICAGO — BOSTON (Millis) — ERIE — BALTIMORE — MOBILE



85% Magnesia Sectional Pipe Covering. An accepted standard for general industrial pipe insulation.

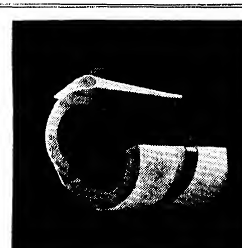
The desire for increased efficiency of heating equipment as well as the need of fuel conservation prompts the engineer to seek the product that provides him with the most economical operating plant. The following Ruberoid Insulating Products are tabulated to enable you to choose quickly the one for the correct purpose. Greater detail and description are provided in the Ruberoid-Watson Catalog on "Heat or Cold Insulating Products," which will be sent on request.

Product	Temp. Limit	Suggested Use
Hi-Temp	to 1600 F	Protective inner layer for low temperature insulations.
Pyrfelt	to 1000 F	Breechings and flues — withstands vibration.
Aristo	to 750 F	Superlative efficiency particularly on oil refinery lines.
Sponge Felt	to 750 F	For vibrating pipes and underground insulation — excellent efficiency.
85% Magnesia	to 600 F	Combined efficiency and reasonable cost — General use in industrial work
Imperial	to 500 F	For temporary lines that require efficiency and constant removal of insulation.
Watcocell	to 350 F	For a low-cost, medium pressure industrial steam line.
Air Cell	to 350 F	Standard insulation for residential pipes.
Woolfelt	to 180 F	For cold and hot water lines. Recommended especially for air conditioning work.
Anti-Sweat	to 120 F	For cold water lines to prevent condensation.
Frost-proof	30 F to 100 F	To assist in the prevention of freezing in circulating water pipes exposed to cold.



Air Cell Pipe Covering—A low-cost insulation for residential use.

Woolfelt Pipe Covering—For the insulation of pipes carrying hot or cold water—also prevents condensation under normal operating conditions.



Sheet and Block Insulations

All of the above products are also made in sheet and block form to whatever thickness may be required. Standard sizes are usually 6, 12, 18 or 36 in. wide x 36 in. long. In this form they are used for insulating flat or irregular surfaces, such as tanks, breechings, furnaces, etc.

Insulating Cements

For the finishing of sheet and block insulation and the insulating of irregular surfaces, such as valves, unions, flanges, etc., the Ruberoid-Watson line of insulating cements is complete. This group of plastics not only uses as its base asbestos, but also takes advantage of such excellent natural products as magnesium, mineral wool and Vermiculite.

Asbestos Cements—Factory Prepared—Grades 203, AA, A, HF.

Asbestos Cements—Mine Run—Grades 115, 214.

Magnesia Cement—85% Magnesia.

High Temperature Cement—Grade H.T.

Mineral Wool Cement—Grade R-W.

Vermiculite Cement—Grade A-11.

RU-BER-OID Insulating Products for Residential Use

RU-BER-OID AIR-MET
Reflective Type

Ruberoid also offers AIR-MET, a reflective insulation consisting of two sheets of pure aluminum. Has 4 reflective surfaces, 4 air spaces and 3 vapor barriers. Installation does a three-fold job—turns back 95 per cent of the radiant heat striking its aluminum surfaces; the air cells retard heat losses by convection; its light weight reduces the loss by conduction to a minimum.

Moisture and verminproof. Simple to apply. Made in folded strips opening up like an accordion. Strips are 80 to 100 ft with scale of feet and inches indicated. Flanges are readily tacked to the sides of studs, joists or rafters. Cartons 8 in. x 18 in. x 31 in., weighing approximately 40 lb, contain enough material for 1000 sq ft of area.

AIR-MET can be stocked in a small space. Twenty cartons provide the insulation coverage of a carload of rock wool. Its insulation value is equal to the best and satisfies the government specifications, as well as meeting the thermal conductivity factors required by heating and air conditioning engineers and public utilities.

Type 2 AIR-MET has one layer of foil, 2 reflective surfaces, with a water-repellent vapor-resisting paper cemented to the opposite side. Suitable for sidewall insulation and can be applied very rapidly.

RU-BER-OID AIR-MET Reflective Insulation

Giant Bats RU-BER-OID Rock Wool

RU-BER-OID ROCK WOOL
Mass or Fill Type
Bats—Loose—Granulated

This indestructible wool is an efficient insulating material. Absolutely fireproof, verminproof and inert toward moisture. Affords excellent sound-deadening and acoustical qualities.

RU-BER-OID Kraflined Rock Wool Bats are carefully fabricated. They are well tailored, uniform and easily handled. They are clean and sufficiently dense to prevent dusting and deterioration. Each bat is backed with a moisture-resistant paper that prevents the infiltration of vapor into the insulated space. RU-BER-OID Kraflined Bats provide "four flap" protection—each edge having an extension that allows adjoining bats to be covered preventing any exposed seam, thus effectively resisting the vapor flow.

Recommended for all exposed spaces, such as sidewalls of new houses or under the roof, either in the roof rafters or the floor joist over the top floor ceiling. Bats without the Krafliner can be furnished if desired.

Packages Contain

Kraflined Standard Bat	15 in. x 23 in. x wall thickness	8 bats—19.16 sq ft	27 lb
Kraflined Demi-Bat	15 in. x 23 in x 2 in. thick	12 bats—28.75 sq ft	30 lb
Kraflined Giant Bat	15 in. x 48 in. x wall thickness	5 bats—25 sq ft	45 lb
Kraflined Giant Demi-Bat	15 in. x 48 in. x 2 in. thick	8 bats—40 sq ft	45 lb

RU-BER-OID Wal-Pac Pads are insulating units 9 in. x 15 in. that can be fluffed up when applied to nearly fill the studding space. Furnished in cartons weighing 25 lb containing 20 pads that should cover 20 sq ft area.

RU-BER-OID Loose and Granulated Rock Wool is also available. Furnished in paper bags containing 35 lb each.

The Pacific Lumber Company

PALCO WOOL INSULATION

100 Bush Street, San Francisco — 59 E. Van Buren St., Chicago — 5225 Wilshire Blvd., Los Angeles — 122 East 42nd St., New York

WHAT IT IS

PALCO WOOL is a loose fill insulating material made from the bark of the Redwood tree, the protective covering of the world's oldest living thing. It is highly refined into an insulating material of light weight wiry fibres of springy resilience. Recent improvements in manufacturing have made it clean, dustless and lighter in weight. In practical use PALCO WOOL has proven to be ideal for all types of construction, large or small, where resistance to conduction of heat is required. It is continuously efficient and reasonably priced, thus assuring economical performance.

USES

PALCO WOOL is suitable for any type of domestic or commerçial construction, in fact every place where an insulating material is required to effectively resist the transmission of heat.

INSTALLATION

Approximately 1 lb of PALCO WOOL is required per square foot of 4 in. thickness. It is easily put in place by hand. Between 100 and 150 lb can be applied per hour per man. It comes in bales weighing approximately 100 lb. Size 24 in. x 24 in. x 26 in.

Send for Data Folder and Sample

Send for "Comfort that Pays Its Own Way," new 16-page folder with comparative data charts and complete information on PALCO WOOL.

PALCO INSULATION WOOL

8 PROPERTIES that make it AN IDEAL INSULATION

1. Thermal Efficiency: The established conductivity of PALCO WOOL is 0.255 Btu per hour per sq ft per inch of thickness per degree F difference in temperature by the Flat Plate Method.

2. Non-Settling: The fibres of PALCO WOOL possess such resilience that no settlement in a wall can occur under the most severe conditions of vibration.

3. Moisture Resistant: The fibres of PALCO WOOL are entirely lacking in capillarity, and have little attraction for moisture, enabling it to remain dry and efficient when in use.

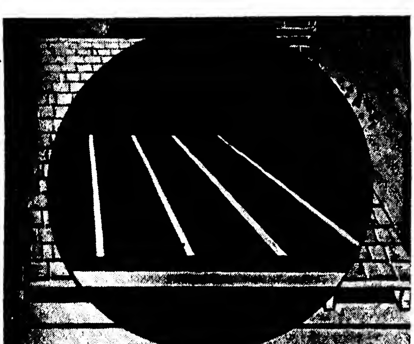
4. Permanent: The inherent antiseptic qualities of PALCO WOOL make the existence of fungus impossible. The fibres retain their resilience indefinitely.

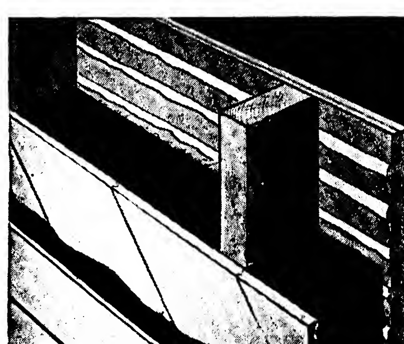
5. Vermin Proof: PALCO WOOL is distasteful and repellent to rodents and insects.

6. Fire Resistant: PALCO WOOL will not readily support combustion and is fire resistant.

7. Odor Proof: PALCO WOOL is odorless itself and does not absorb or give off odors.

8. Economical: PALCO WOOL is light in weight and low in density, offering exceptional thermal efficiency per dollar invested.





The Standard Lime & Stone Company

First National Bank Building — **Baltimore, Maryland**

Manufacturers of Capitol Rock Wool Home Insulations

Franchised Distributors in all Principal Cities

The Standard Lime and Stone Company, prominent in the building materials industry since 1888, manufactures Capitol Rock Wool Insulations, all types of lime products, fluxing and crushed stone, Capitol Portland Cement, etc.

A new manufacturing technique produces a refined, longer, more flexible fibre—a more effective form of Rock Wool.

CAPITOL ROCK WOOL IS EFFICIENT

Transmission of both heat and sound is virtually eliminated by the effective tiny dead air cell structure of Capitol Rock Wool. CAPITOL ROCK WOOL WILL NOT BURN.

Capitol Rock Wool in Existing Homes

Capitol Rock Wool Grade "A" Blowing Fibre is pneumatically introduced into the wall air spaces and between rafters or joists in roofs or attic floors of homes already built, whether the construction is shingle, clapboard, brick veneer, stucco or half-timbered. This "blown" installation is the same for any type of construction and is performed by franchised blowing contractors. Installation leaves no telltale marks.

Installing
CAPITOL ROCK WOOL BATTS
in a New Building

Capitol Rock Wool in New Structures

New homes or buildings are generally insulated at the time of erection by placing prefabricated Batts between the studding and roof rafters. Many new structures however, are effectively insulated by pneumatically installing Grade "A" Blowing Fibre after the scratch coat of plaster is applied.

CAPITOL ROCK WOOL BATTS
A Distinctive Product

Moisture-proofed. The processing of Capitol Rock Wool Batts gives the fibres a moisture resisting characteristic.

Vaporproofing Membrane. To protect the exposed surface of the Batts against moisture from wet plaster and excessive interior humidity, a tested membrane is enclosed separately in each carton. It is 17½ in. wide to make tacking quick and easy, and of sufficient length to give a smooth continuous membrane-protected surface without open joints between Batts.

Cuts Cost. The Capitol Rock Wool Batt is semi-rigid. Batts are one size, 15 in. x 24 in. Because of their size and stiffness, they "spring fit" between framing members spaced either 16 in. or 24 in. on centers. They cut readily to fit irregular shaped spaces.

Two Thicknesses. Wall-thickness, for maximum efficiency; also 2 in. thickness.

Permanency. Capitol Rock Wool is permanent, non-deteriorating. It brings lasting efficiency, fire-protection and comfort to the home with fuel savings that ultimately return the investment many times over.

Send for catalogs and samples of Capitol Rock Wool Home Insulations.

"Look for the Capitol Dome on every package."

United States Gypsum Company

300 W. Adams Street, **Chicago, Ill.**

Sales Offices in Principal Cities

PRODUCTS

Strip Wool	**Insulating Building Board**	**Insulating Plank**
Bat Wool	**Insulating Lath**	**Insulating Mouldings**
Junior Bat Wool	**Metal Reinforced Insulating Lath**	**Roof Insulation**
Granulated Wool	**Insulating Tile**	**Asphalt Coated Sheathing**

RED TOP INSULATING WOOL

Description

Red Top Insulating Wool is an extremely light, fluffy mineral fiber insulation—a fireproof material.

The nature of the raw materials used, permits accurate manufacturing control—the product is uniform and long-fiber wool. It contains a minimum of non-insulating materials. It is springy and resilient.

Low Thermal Conductivity

The heat conductivity of Red Top Insulating Wool (1-½ lb density) is 0.266 Btu per inch thickness, per square foot per hour, per degree Fahrenheit difference in temperatures. (Tests by Professor Peebles, Armour Institute of Technology).

Light Weight (Density)

In its standard density, Red Top Insulating Wool weighs but 1-½ lb per cubic foot. It is furnished in full and semi-thickness.

Applying Red Top Strip Wool between studs. Strip Wool has a waterproof paper backing with a 2 in. flange for nailing to studding. Used as shown it prevents the entrance of moisture into the insulation.

WEATHERWOOD BOARD

Weatherwood Insulating Board is a felted wood fiber product that has been treated to make it highly moisture resistant. It is a homogeneous board, free from laminations and tendencies to split through the center.

Low Thermal Conductivity

The average established in tests is 0.33 Btu per hour, per square foot per inch thickness, per degree Fahrenheit difference in temperature.

Resistance to Moisture

Tests show that this board, after being submerged in water for a period of two (2) hours, has an average water absorption of less than 15 per cent.

Structural Strength

The tensile strength is over 300 lb per square inch, and the modulus of rupture, over 500 lb.

Durability

There is nothing in this board that will deteriorate and impair the original insulation value. The Moisture Resistant chemical makes the board distinctly distasteful to rodents and insects.

Uniformity

Adequate laboratory control and mill inspection assure uniformity in density and structural strength.

Applying Asphalt Coated Sheathing

Wood Conversion Company

First National Bank Building, **St. Paul, Minn.**

NEW YORK CHICAGO TACOMA DALLAS

BALSAM-WOOL AND NU-WOOD INSULATIONS

BALSAM-WOOL	**NU-WOOD**	**NU-WOOD**
Sealed Insulation	**Interior Finish**	**Lath**
Acoustical Blanket	**Plank**	**Roof Insulation**
Sound Deadening	**Tile**	**Industrial Insulation**
Industrial Insulation	**Board**	**Refrigerator Insulation**
Refrigerator Insulation	**Sheathing**	**KOLOR-TRIM Pre-decorated Moldings**

BALSAM-WOOL—The Original Moisture Barrier Insulation

The Moisture Barrier, which is universally recommended by engineers and architects, has been incorporated in Balsam-Wool for 16 years. This Barrier, improved as construction and equipment demanded, now consists of double layers of asphalted kraft—a heavier liner being used on the warm side. Encased between this protective covering is an insulating mat of fleecy wood fibres, chemically treated to resist fire, rot, termites and vermin. 92 per cent of the mat volume is dead air.

*Balsam-Wool Spacer Flange**

Balsam-Wool SEALED Insulation is fabricated at the factory to a controlled density of 2.2 lb per cubic foot. The mat has a coefficient of .25 Btu per hour, per square foot, per 1 degree F difference in temperature, per 1 in. thickness.

As applied, factory efficiency is assured. The Spacer Flange* on each edge folds over and is fastened to framing members with a staple hammer, assuring important air space, front and back.

Balsam-Wool is available in ½ and 1 in. thicknesses in widths of 12, 16, 24 and 33 in.—Wallthick in widths of 12, 16, 20 and 24 in.

*Pat. Applied For.

Application is quick and easy

NU-WOOD INTERIOR FINISH — STRUCTURAL INSULATION

Nu-Wood Interior Finish (Tile, Plank, Board and Wainscot) is applicable either to new construction or to existing buildings. It offers varied and pleasing decoration, also insulation and acoustical value.

Nu-Wood Insulating Lath has several times the bonding strength of wood lath—continuous surface eliminates dirty lath marks, reduces cracks. V-joint resists trowel pressure in both directions—assures unbroken insulation value.

Nu-Wood Insulating Sheathing is surfaced on both sides with double coats of special moisture proofing compound. Large boards, marked for nailing—speed erection—stronger, windproof, insulated construction.

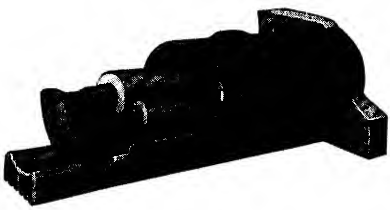
AMERICAN DISTRICT STEAM COMPANY

PRODUCTS for STEAM SERVICE

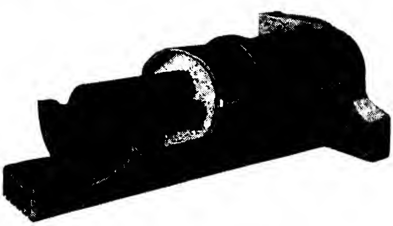
NORTH TONAWANDA, N.Y.

IN BUSINESS OVER SIXTY YEARS

Branches and Agents in Principal Cities



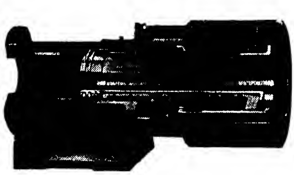
Conduit with Asbestos Insulation



Conduit with ADSCO-Corning Insulation

Internally Guided Joint

Internally-Externally Guided Type



Piston-Ring Type Joint

ADSCO-BANNON TILE CONDUIT

For Underground Pipe Lines

A vitrified, salt-glazed, separable tile conduit, with or without base drain, combining strength, durability and simplicity in connection with one or more bare or insulated pipes in underground steam or hot water lines. It provides high insulating efficiency in a conduit that can be installed at small labor cost and is adaptable to varied installation conditions.

The pipe supports are held in fixed position on reinforced conduit sections. They do not pierce the conduit wall and are suitable for use with sectional molded insulation or filler insulations, particularly ADSCO-Corning Filler Insulation. Write for Bulletin No. 35-67AG.

INTERNALLY GUIDED JOINT

An internal guide ring provides full guiding for entire travel of the slip, limit stops prevent overtravel of the slip in either direction. No metal to metal contact against polished slip surface. Small over-all dimensions make it suitable for use in restricted spaces. Pressures to 300 lb and temperatures to 750 F. Write for Bulletin No. 35-30G.

INTERNALLY-EXTERNALLY GUIDED JOINT

A completely guided slip type joint. Both ends of slip guided throughout entire length of travel by an internal guide ring and an external guide in hood. Slip cannot pull out of body. No metal to metal contact against slip surface. Pressures to 300 lb and temperatures to 750 F. Write for Bulletin No. 35-20G.

PISTON-RING EXPANSION JOINT

Piston rings in the guide ring attached to the inner end of the slip hold the line pressure, enabling the joint to be unpacked and repacked under full operating pressure without interruption to service. A fully guided joint for pressures to 400 lb and temperatures to 750 F. Slip cannot pull out of body. Write for Bulletin No. 35-15G.

(*See also Page 966*)

H. W. Porter & Co.

INCORPORATED

Newark, New Jersey

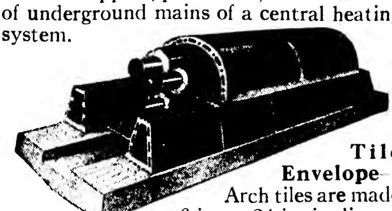
Permanent Protection and Insulation for Underground Pipe Lines

BALTIMORE, MD. CHARLOTTE, N. C. RICHMOND, VA. WASHINGTON, D. C.

Therm-O-Tile
REG. U.S. PAT. OFF.

STEAM CONDUIT SYSTEMS

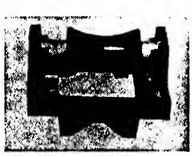
For Central Heating—Therm-O-Tile is a complete conduit system for the permanent support, protection, and insulation of underground mains of a central heating system.



Tile Envelope—Arch tiles are made 6 in. to 24 in. in diameter, and with 5 different size base tiles they produce 27 different conduit cross sections.

Foundation—The base of the system is a thick concrete slab poured directly in the trench bottom, reinforced with steel when installed over a filled or boggy ground.

Drainage—The drainage system of the conduit is entirely internal, open to inspection at manholes, and of ample capacity to keep the pipe space dry at all times without any possibility of becoming clogged with silt or vegetation.



Pipe Support for Single Pipe.

Pipe Support for Three Pipes.

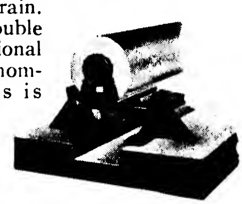
Pipe Support—All pipes are supported on cast-iron adjustable supports resting directly on the concrete base independent of the tile envelope.

Accessibility—All piping is installed before tile is placed, giving complete accessibility for welding, testing and insulation. Pipe fitters work on concrete slab "walkway."

Strength—Due to immovable concrete base and arch construction of extra heavy tile members, conduit will sustain any roadway traffic load usually encountered without extra reinforcement.

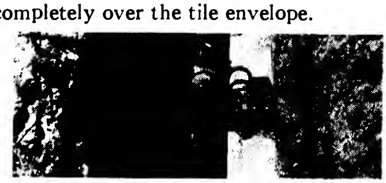
Insulation—Either sectional pipe covering or Thermobestos waterproof fibre filling may be used for insulation, as the insulation space is kept dry at all times, by the internal drain.

For single or double pipe lines, sectional insulation of economical thickness is recommended; for multiple pipe lines, a filler type of insulation is usually more economical in first cost.



Pipe Saddle. Permits full thickness of insulation between pipe and roller.

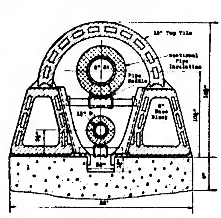
Waterproofing— Under normal soil conditions, this conduit is waterproof. If marshy ground or partially submerged conditions are encountered, the conduit may be made completely waterproof by the use of membrane waterproofing applied under the slab on a sub-base and carried completely over the tile envelope.



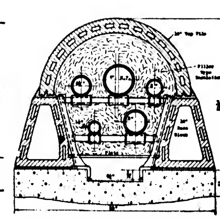
Anchor Block. Fits directly in line with Base Tiles.

Efficiency—The degree of thermal efficiency secured depends upon the type and thickness of insulation used. This conduit, due to its sealed air chambers in the tile and dry insulation space, adds to the normal efficiency of the insulating material on the pipe lines.

Representatives—Therm-O-Tile is also sold and installed locally by Johns-Manville Construction Units.



Single or Double Pipe Lines Using Sectional Pipe Insulation.



Multiple Pipe Lines Using Filler Type Insulation.

The Ric-wiL Company

Agents in Principal Cities | RIC-WIL (REG. U. S. PAT. OFF.) | **ESTABLISHED IN 1910**

CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

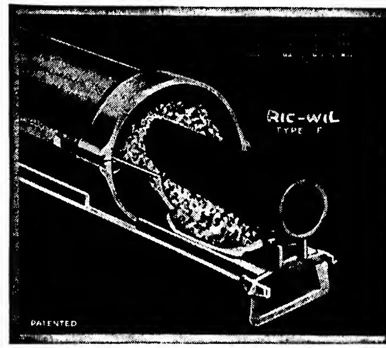
Union Trust Bldg., **Cleveland, Ohio** — **New York, Chicago**

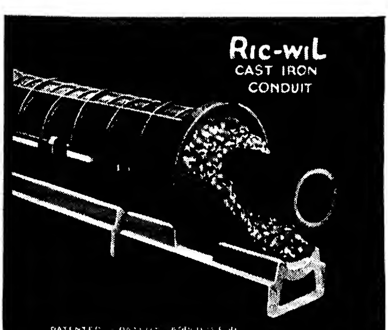
Conduit—Standard conduit is vitrified salt glazed tile or cast-iron with Loc-liP Side Joints, bell and spigot type, lined or unlined. Tile in 24 in. sections, sizes 4 to 27 in. inside diameter. A Super-Tile conduit to support any average traffic load is also available. For extra heavy duty under railroads or where conduit is subject to extreme loads, Ric-wiL is made of cast-iron in 2 or 4 ft sections. Where reduced labor cost is not essential, Ric-wiL Universal Type System is recommended (details on request).

Base Drain—Standard Base Drain is vitrified salt glazed tile for tile conduit and extra heavy tile or cast-iron for the cast-iron conduit, in 24 in. lengths. Made in three sizes to support and drain **properly** all conduit sizes.

Pipe Supports—Standard pipe supports will carry from one to five or more pipes and are ordinarily spaced 12 ft apart. They are strong, made of cast-iron, rust-proofed, and interlock with the base drain foundation, imposing no load on the conduit itself. No movement of pipes can disturb them. Special pipe supports available for different conditions. Information on request.

Insulation—Ric-wiL Dry-paC Waterproof Insulation is high-grade long fibre asbestos processed so that it is permanently water repellant. Of unusually high efficiency and great natural strength, it will not slump away from pipes and is non-corrosive. Sample will be sent gladly for testing. Ric-wiL No. 11 Insulation (same as Dry-paC except non-waterproof) or sectional pipe covering can also be furnished. For lined conduit, diateomaceous earth mixture is molded and keyed inside the tile.





Accessories—Shutter sleeves, alignment guides, filter cloth, manhole covers, and other accessories will be furnished as desired.

Engineering Service—Full cooperation with architects and engineers. Installation supervision if desired.

Technical Data—Tabulated Steam Heating Rates, Test Reports, Service Detail Bulletins, Catalog Bulletins, Central and District Heating Bulletins and Architects and Engineers Detail Sheets available upon request.

Ric-wiL Unit Steam Main, a prefabricated, ready-to-install unit, 13½ ft long, including conduit (Armco Iron), pipe, insulation, and accessories. Ideal for speed and economy on district heating projects.

Underground Steam Construction Co.

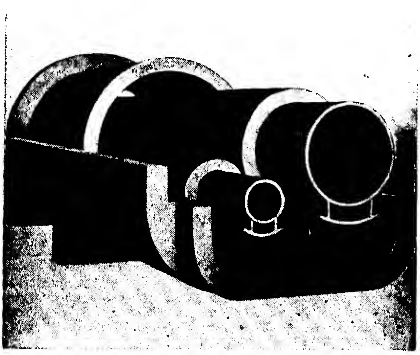
75 Pitts Street, **Boston, Mass.**

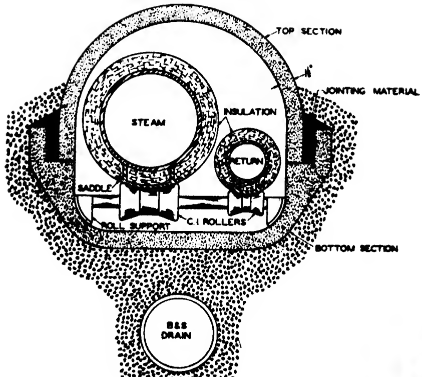
PRODUCTS—Engineering and Contracting of Steam Line Installations. Underground Steam Conduits.

USCCO PRE-CAST CONCRETE CONDUIT

This unit marks a long step forward in the economics and mechanics of laying underground steam lines.

Its Outstanding Features Are:





Strength resulting from flat reinforced bell; from the longitudinal joints, and from the fact its sections are 4 ft long, reducing the usual number of joints necessary.

Shape which allows 12 per cent greater inside capacity than circular conduit of like diameter. This permits larger pipes in a given size conduit and more room for drainage, pipe supports and rolls.

Ease of Installation resulting from the fact that **all** top and bottom halves mate. Bottoms may be laid for any distance, the piping installed and then the tops brought up and laid. This speeds up the work and protects idle halves from damage. They can be stored away from the job.

Materials are high strength cement with suitable aggregate, amply reinforced with wire mesh.

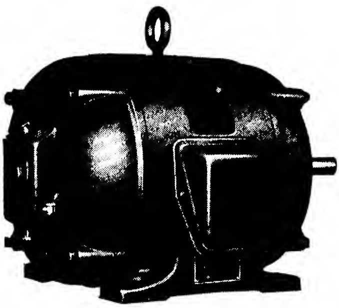
Pipe Supports are of cast-iron and quickly installed. They may be placed anywhere in the conduit. The weight of the pipe holds them in place.

Joints may be of any standard accepted brand of jointing compound or they may be of standard Portland cement mortar.

Century Electric Company

1806 Pine Street, St. Louis, Mo.

Offices and Stock Points in Principal Cities



Century motors have been developed specially to meet the exacting requirements of the Air Conditioning Industry —Quiet Starting—Quiet running—Remarkably free from vibration.

CENTURY MOTORS

For Compressors, Pumps, Fans, Blowers, Refrigerators, Stokers, Oil Burners

Type of Motor	Horse Power Range	Starting Duty	Remarks	Applications
POLYPHASE MOTORS				
SCH—Squirrel Cage	3 to 200	Heavy	Low Starting Current High Starting Torque	Refrigerators, Piston or Plunger Pumps, Compressors, etc.
SC—Squirrel Cage	1/8 to 600	Medium	Normal Starting Current Normal Torque	General Purpose Motors, Blowers, Pumps.
SCN—Squirrel Cage	7½ to 200	Medium	Lower Starting Current than SC Normal Torque	
AS—Automatic Start	1 to 60	Heavy	Lower Starting Current than SCH High Starting Torque	Refrigerators, Piston or Plunger Pumps, Compressors, etc.
SR—Slip Ring	1 to 350	Heavy	For Frequent Starting and/or Speed Control	Fans, Blowers, Centrifugal Pumps, Compressors, etc.
SINGLE PHASE MOTORS				
RS—Brush-Lifting	1/8 to 40	Heavy	Low Starting Current, High Starting Torque	Piston or Plunger Pumps, Refrigerators, Stokers, Compressors, etc.
CPH—Cap. Start and Run	1/8 to 10	Heavy	High Starting Torque	
CSH—Cap. Start	1/8 to 3/4	Heavy	High Starting Torque	
CSN—Cap. Start	1 to 10	Medium	Normal Starting Current	Fans (Belted or Direct Connected) Centrifugal Pumps, etc.
CPX—Cap. Start and Run	1 to 10	Light	Must be Loaded to at Least 50 per cent Capacity	
SP—Split Phase	1/20 to 1/3	Medium	Unrestricted Starting Current	Oil Burners, Unit Heaters, Blowers, Fans, Small Tools, etc.
SP—Split Phase	1/20 to 1/3	Light	Restricted Starting Current	
DIRECT CURRENT MOTORS				
DM-DN-R Shunt Wound Constant Speed	1/20 to 300		Torque is limited only by Commutation. A Direct Current Motor has ample Torque to start any load that it can carry when up to speed. Starting Current is limited by Controller to about 150 per cent of full load current for light starting torque requirements with corresponding increases in current for increased starting torque	Fans, Blowers, Centrifugal Pumps, Machine Tools, etc.
DM-DN-R Compound Wound Varying Speed	1/12 to 300			Reciprocating Pumps, Compressors and Machines with Flywheels, etc.
DN-R Shunt Wound Adjustable Speed	1/2 to 200			Fans, Blowers, Machine Tools, etc.

Size Range up to 600 horse power

See also pages 908 and 909

The American Brass Company

General Offices: Waterbury, Conn.

Manufacturing Plants:

Ansonia, Conn Torrington, Conn. Waterbury, Conn. Buffalo, N Y Detroit, Mich Kenosha, Wis

Offices and Agencies in Principal Cities

ANACONDA
from mine to consumer

CANADIAN PLANT Anaconda American Brass Limited, New Toronto, Ontario

PRODUCTS—Anaconda Deoxidized Copper Tubes and Fittings; Anaconda "85" Red-Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air conditioning equipment

ANACONDA COPPER TUBES AND FITTINGS

For Heating, Plumbing and Air Conditioning

Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Fittings offer an unusual combination of advantages in hot water heating systems at a cost only slightly higher than black iron and approximately the same as wrought iron pipe. These advantages may briefly be summarized as follows:

Low Friction Loss—Because the inside surfaces of copper tubes are inherently smoother than those of pipe and tubes made of ferrous materials and also because they do not become roughened by the formation of rust, these tubes offer a minimum resistance to flow. In addition, the long radius turns of Anaconda Elbows and the smooth inside surface of Anaconda Wrought Copper Fittings further reduce friction losses.

These factors naturally increase the efficiency of the system, particularly when it includes a forced pressure circulator.

Low Heat Loss—The bright copper tubes radiate less heat than black iron pipe of the same size.

Ease of Installation—In many places the flexibility of copper tubes simplifies connections that ordinarily would be awkward and expensive to make with rigid pipe and threaded fittings. Anaconda Solder Fittings are compact. They can be installed in constricted space where the use of a wrench would be impossible.

Architects and builders naturally object to large holes and notches cut in the framing members of a building for the passage of piping. Anaconda Copper Tubes can be installed with a minimum of cutting in the structure, although holes should be large enough to permit movement of tubes due to expansion and contraction.

Appearance—Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Solder Fittings present an attractive appearance. It is customary practice to clean the tubes after they are installed and apply a coat of clear lacquer or similar substance. This keeps the tubes bright and makes an installation of which both plumber and owner can be proud.

Temper and Gauges—Anaconda Copper Tubes are made in both hard and soft temper and in three types as to wall thickness. Designated as types K, L and M, they meet the requirements for these types of tubes in U. S. Government Specification WW-T-799* and A.S.T.M. Specification B-88-33*. Type K, the heaviest, is recommended for heating lines and general piping. Type L tubes are suitable for interior plumbing.

Accuracy of Dimensions—Anaconda Deoxidized Copper Water Tubes are all finished to the close tolerances required by the A.S.T.M. and Federal Specifications, which have been found essential for efficient assembly with solder fittings.

Permanent Identification—For permanent identification, the name "Anaconda" and the letter designating the type of tube is stamped in the metal at intervals of approximately 18 in., throughout every coil or straight length of tube.

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*Specifications for Type M tubes include hard drawn tubes only.

The American Brass Company

Availability—Anaconda Copper Tubes, in all standard sizes, are carried in stock by distributors of Anaconda Pipe, located in the principal trading areas of the country. These tubes, in sizes up to and including 1¼ in. are furnished soft in 30, 45 and 60-ft coils; also hard and soft in 20-ft straight lengths. Sizes over 1¼ in are furnished, hard or soft, in straight lengths only.

ANACONDA SOLDER FITTINGS

Anaconda Solder Fittings are available in both wrought copper and cast bronze. They are made to the exacting dimensions so essential for sound, leak-proof joints. Smooth inside surfaces permit quick, thorough cleaning which is necessary for satisfactory soldered connections. Deep cups, with adequate shoulders for the tubes to butt against, provide a maximum area for the solder bond.

All Anaconda Solder Fittings are tested to 90 lb per sq in. air pressure under water, which is the equivalent of 400 to 450 lb per sq in. water pressure. They are so designed as to offer a minimum of resistance to flow.

Anaconda Wrought Copper Solder Fittings—Anaconda Wrought Copper Solder Fittings provide copper to copper connections. They are uniformly true to size, of one piece, seamless construction, and are free from porosity—features which make these fittings ideal not only for heating lines but also for air conditioning and refrigerating installations, where the penetrating power of refrigerants demands freedom from porosity.

The American Brass Company offers Anaconda Wrought Copper Solder Fittings, including elbows, tees, couplings, unions, and a complete range of reduction and adapter combinations from ¼ in. to 4 in. inclusive.

Anaconda Cast Bronze Solder Fittings—Anaconda Cast Bronze Solder Fittings are extensively used for interior plumbing. They are available in all standard sizes from ¼ in. to 12 in. (sizes up to 4 in. are carried in stock), in a complete line of elbows, tees, couplings and unions, including all standard reduction and adapter combinations. With such a broad range of Cast Bronze Solder Fittings, any usual type of connection can be made without resorting to the trouble and expense of using combinations of fittings.

Literature — Anaconda Deoxidized Copper Tubes, Fittings, Solder and Accessories are discussed at length in Anaconda Publication 11 B-1, 11th Edition. Copies will be mailed to Engineers on request.

ANACONDA "85" RED-BRASS PIPE

Anaconda "85" Red-Brass Pipe, in standard pipe sizes, is offered as the highest quality corrosion-resistant pipe commercially obtainable at a moderate price and is recommended for steam return lines.

Anaconda "85" Red-Brass Pipe contains 85 per cent copper and conforms to government specifications for Grade "A" water pipe. The words "Anaconda 85 Red-Brass" are stamped in the metal at intervals of one foot throughout each length.

EVERDUR

***Everdur Tanks**—Everdur Copper Silicon Alloy is a special non-rust alloy which combines high strength and complete immunity to rust with ready weldability. It is an ideal material for durable, rustless water tanks of every description—from domestic range boilers to giant storage heaters for hotels, laundries, hospitals, textile plants, schools or breweries.

Everdur is made in all commercial shapes, including tank plates which have physical properties as given in A.S.T.M. tentative specifications B96-36T. For additional data, and names of fabricators, address our nearest office or agency.

Everdur for Air Conditioning Equipment—Everdur Metal has been used with marked success for fans and blowers, ducts, humidifiers (air washers) and for various cast and wrought parts of other equipment items subject to corrosive influences.

Because of its strength and welding properties, Everdur may be substituted for steel, fabricated by substantially the same methods, and with the same equipment.

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*"Everdur" is a trademark of The American Brass Company registered in the U. S Patent Office

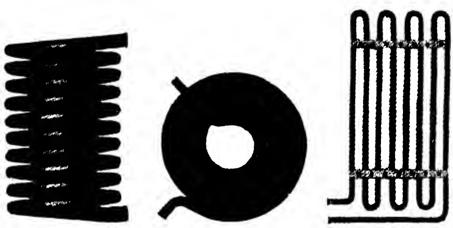
Arthur Harris & Co.

210-218 N. Aberdeen (formerly Curtis) Street **Chicago, Ill.**

ENGINEERS — FABRICATORS OF NON-FERROUS METALS AND STAINLESS STEEL

PRODUCTS—Apparatus for Brewers, Distillers, Dyers, Paper Mills, Pharmaceutical Manufacturers, Manufacturers of Acetic Acid, Grain and Wood Alcohol, Cider, Confectionery, Gelatine, Glucose, Glue, Glycerine, Lacquer, Malted Foods, Meat Extracts, Milk Products, Preserved Fruits, Sugar, Tan Liquor Extract, Turpentine, Varnish, Vinegar, etc. Bulletin on request.

Metals Fabricated—Aluminum, Block Tin, Brass, Bronze, Copper, Everdur, Monel, Nickel and Stainless Steel.

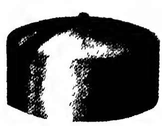


Coils

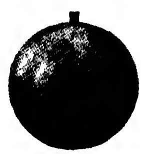
We have special equipment for making coils in all shapes and sizes from pipe or tubing—copper, brass, aluminum, stainless steel, monel, block tin and pure nickel Standard or special fittings Send sketch, blue-print or old coil

Metal Floats

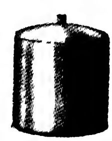
Cylindrical



Flat Cylindrical



Ball



Cylindrical

Column

Made of copper, plain steel, stainless steel, aluminum, brass, monel, pure nickel, and Everdur, for open tank and all pressures

Seamless copper ball floats carried in stock in diameters of 4 in , 5 in., 6 in , 7 in , 8 in., 10 in. 12 in. for open tank and pressures of 25, 50, 100 and 150 lb Floats in special sizes and pressures—made to order Stainless steel ball floats 2½ in. to 12 in. for high pressure and corrosion carried in stock—special stainless steel floats made to order—stainless steel ball floats larger than 12 in. diameter can be made up specially. Float catalog sent on request.

Copper Expansion Joints

For low pressure and vacuum Made in two styles—convex and concave. Sizes 4 in. to 60 in. diameter Cast iron or steel flanges. Flanges drilled to American standard unless otherwise ordered.

Bends

We make bends in every shape from all sizes of pipe and tubing in copper, brass, aluminum, stainless steel, monel, tin and nickel. Standard or special connections. U-bends for storage water heaters.

Also special pipe work for industrial installations, plumbing, heating and brewing. Perforated pipe, double pipe coolers, etc.

Jones & Laughlin Steel Corporation

AMERICAN IRON AND STEEL WORKS

Jones & Laughlin Building, **Pittsburgh, Pa.**

WELDED AND SEAMLESS STEEL TUBULAR PRODUCTS

J & L Welded Pipe

Jones & Laughlin manufactures Standard Weight, Extra Strong, and Double Extra Strong Welded Pipe, Black and Galvanized, for steam, gas, air, water, refrigeration and sprinkler work. Sizes: 1/8 in. to 16 in. O.D. inclusive

J & L Copper-bearing Steel Pipe, when specified, can be supplied in standard weight, or extra strong, black or galvanized. Use of this product is recommended for long life, where piping is to be exposed to the atmosphere or other alternate wet and dry conditions.

Jones & Laughlin Steel Pipe is made of soft, weldable steel rolled from solid ingots made to a special analysis The steel pipe produced is soft and ductile, free cutting, strong at the welds, and free from excess scale J & L Pipe is commercially straight and free from blisters, cracks or other injurious defects

Careful attention is given the threading of the pipe with good clean-cut threads fitted with sound couplings correctly tapped to give a tight joint. Soft, ductile steel of free cutting quality enables the contractor to cut clean, sound threads on the job

The Jones & Laughlin process of galvanizing assures a thorough coating and insures against pipe being clogged with spelter The galvanized coating adheres strongly and does not tend to flake off

J & L Seamless Pipe

J & L Seamless Pipe is made in three weights, standard, extra strong and double extra strong. Sizes: 3/8 in. nominal to 14 in O D. inclusive.

J & L Seamless Steel Pipe is pierced from a solid billet—there are no welds The result is dependable and uniform wall strength. The method of manufacture, and the use of only specially selected steel, assure exceptional ductility, a quality that is essential to successful coiling and bending, and flanging for Van Stone joints

J & L Seamless Pipe can be used with full satisfaction in either threaded joint or completely welded installations

Ductility, strength and safety—make this product especially adaptable for air, steam, gas and gasoline lines, boilers, refineries, dry kilns, refrigerating systems and other exacting applications

J & L Hot Rolled Seamless Steel Boiler Tubes

J & L Seamless Boiler Tubes are manufactured in accordance with the *A.S.M.E.* Boiler Code and comply with the *A.S.T.M.* Specifications and the rules and regulations of the Bureau of Navigation and Steamboat Inspection of the U. S. Department of Commerce. They are supplied in a full range of standard sizes, from 1 in. O.D. to 6 in. O D inclusive.

The process by which Jones & Laughlin manufactures seamless boiler tubes is largely responsible for the unusually high ductility of the product. It is a process in which a forging action is predominant. It makes J & L Boiler Tubes stronger yet more pliable and, therefore, more easily formed in a cold state

Other J & L Tubular Products

J & L also manufactures Reamed and Drifted Pipe in sizes 1 in. to 6 in. inclusive, Dry Kiln Pipe, Pipe for Refrigeration Service, Water Well and Irrigation Casing, Line Pipe and a complete line of Oil Country Tubular Products in welded and seamless.

Also J & L Flat Galvanized Sheets for Air Conditioning and Ventilating Work

Pipes, ducts, stacks and trunk lines made of J & L Flat Galvanized Sheets give a lasting, neat looking job These sheets have a tight galvanized coating that will not spall or flake off during forming operations J & L Flat Galvanized Sheets provide uniform resistance to corrosion Because of their uniform ductility, even surface and flatness, J & L Flat Galvanized Sheets meet the most exacting specifications for severe bending and forming operations.

Wolverine Tube Company

SEAMLESS COPPER, BRASS AND ALUMINUM

Main Office and Mill: 1411 Central Avenue, **Detroit, Mich.**

NEW YORK OFFICE 420 Lexington Avenue

Sales Offices

ATLANTA, GA	631 Spring St	MINNEAPOLIS, MINN	529 S Seventh St
BALTIMORE, MD	121 S Gay St	NEWARK, N J	965 Broad St
BOSTON (CAMBRIDGE), MASS	195 Albany St	PHILADELPHIA, PA	231 North 12th St
CHICAGO, ILL	129 S Jefferson St	PITTSBURGH, PA	1000 Columbus Ave , N S
CLEVELAND, OHIO	1740 East 12th St	PORTLAND, ORE	524 N W 14th Ave
DALLAS, TEXAS	2813 Canton St	ST LOUIS, MO	4565 McRee Ave
DAYTON, OHIO	Route No 9	SAN FRANCISCO, CALIF	7 Front St
DENVER, COLO	1210 California St	SEATTLE, WASH	1005 E Pike St
LONG ISLAND CITY, N Y	47-31 31st Place	WASHINGTON, D C	1108 16th St
LOS ANGELES, CALIF	1015 East 16th St	TORONTO, ONT	137 Wellington St , W
LOUISVILLE, KY	125 S 5th St	WINNIPEG, MAN	80 Lombard St
MILWAUKEE, WIS	647 W Virginia St	EXPORT H M ROBINS CO , 120 Madison Ave ,	Detroit, U S A

The experience of over 20 years of seamless tube manufacture, the use of fine, up-to-date equipment, and strict adherence to Government and customer specifications, are responsible for the uniform, high quality of Wolverine products.

Immediate Shipment From Large Stocks

DEOXIDIZED COPPER WATER TUBE

Government Type K

Recommended for Air Conditioning, Refrigeration, Oil Burner, and Plumbing and Heating installations Also for Gas, Steam, Oil Lines, and industrial uses and where water conditions are severe.

Government Type L

For Oil Burner, Air Conditioning, Refrigeration, and general plumbing uses Suitable for normal water conditions

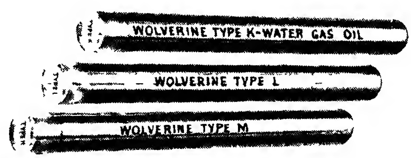
Types K and L furnished in hard or soft temper in straight 20 ft lengths, soft temper in 30, 45, and 60 ft coils

Government Type M

Suitable for Air Conditioning and Refrigeration installations and for interior plumbing and heating purposes.
Furnished in hard temper in straight 20 ft lengths only.

WOLVERINE COPPER WATER TUBE

Types K, L, and M (made to U S Government WW-T-799 and *A S T M* B 88-33 Specifications), for air conditioning and refrigeration installations, make this product suitable for piping in accordance with all refrigeration codes



Wolverine copper air conditioning tube is the Standard for Air Conditioning work —Now available with cellophane caps and discs telling size and type.

Type K—Orange
Type L -White
Type M—Green

WOLVERINE WROUGHT COPPER FITTINGS

These new Wolverine Fittings are of the straight-line design (ends not expanded). They make stronger, neater joints, more

quickly made, give trouble-free service and longer life A complete range of sizes is available

Write for Catalog B

American Society of Refrigerating Engineers

37 West 39th Street, New York, N. Y.

REFRIGERATING DATA BOOK

THE '39-'40 edition of the *Refrigerating Data Book* achieves an excellence matched by few engineering or scientific books. The compilation of the work of many experts in refrigeration, heat, power and manufacturing, as well as authorities from the field of biology, it contains all the fundamental data on the art of refrigeration and air conditioning. It is presented in the simplest possible style so that parts may be used by those of any degree of technical interest.

This fourth edition of the *Data Book* contains over 600 pages in 50 chapters divided into eight sections separated by contents pages with thumb tabs. Contents outline is shown inside the cover for ready reference. It includes a catalog and list of manufacturers, distributors and engineers. There are several inserted charts.

This volume is specifically devoted to principles and refrigerating machinery. Another volume, appearing two years hence, will be devoted specifically to the many applications of refrigeration. Thenceforth the Data Book will be in two volumes. The price of each is $4 with express charges collect.

A series of pamphlets on the application of commercial refrigeration to such uses as fur storage, vegetable coolers, farmers locker plants and the like will appear in 1939.

MEMBERSHIP ACTIVITIES

IT is the policy of the *A.S.R.E.* to treat in its meetings current subjects touching upon all phases of the art of refrigeration. Membership is in four grades with dues from $7.50 to $17.50. Sections hold meetings in the following cities: Boston, New York, Philadelphia, Detroit, Chicago, Milwaukee, St. Louis and Los Angeles. The Society holds its 35th Annual Meeting January, 1940, in Chicago.

REFRIGERATING ENGINEERING

Refrigerating Engineering in its 37th volume continues to be the periodical source of authoritative information on all phases of the arts and sciences of refrigeration. In its most solid aspects it carries the *Journal* of the *A.S.R.E.*, a living record of the technical advance of practice and research in its field. The larger portion of its pages is, however, devoted to material of wider appeal, with original articles, none the less authoritative, but written in a journalistic style to enable the reader to get the background of each subject and an understanding of it that will stick. *Refrigerating Engineering* also prints news, features, write-ups of interesting personalities, and carries detailed reports of local and national meetings of *The American Society of Refrigerating Engineers.*

Refrigerating Engineering has long been unique in its field not alone for the originality and authority of its contents, but for its coverage of all phases of refrigeration both as to machinery and application problems. The applications of this art are, of course, very numerous, and *Refrigerating Engineering* keeps pace as the scope of refrigeration is widened.

The rate has been reduced to $3 for 1939.

CODES AND STANDARDS

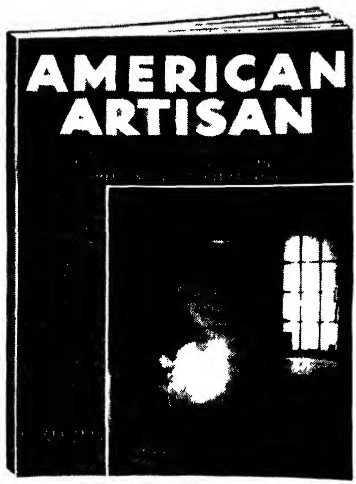
THE newest standards to be published include the Safety Code for Mechanical Refrigeration, a test code for Rating Refrigerant Expansion Valves and the (joint) Code of Minimum Requirements for Comfort Air Conditioning. Others cover the testing and rating of mechanical condensing units and the testing and rating of air conditioning units. The Society participates in fifty or more standardizing projects under current study.

American Artisan

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, **Chicago, Ill.**



AMERICAN ARTISAN, now in its 60th year of publication, covers the field of warm air heating, residential air conditioning, and sheet metal contracting. A special section of each issue has been devoted to air conditioning since 1932, when it first became apparent that air conditioning for homes was to be along the lines of the central, forced warm air heating system

Its readers are warm air heating and sheet metal contractors, dealers, jobbers and manufacturers, and also architects, engineers, and public utility companies who take it for its thorough coverage of air conditioning for the home field.

To answer the industry's need for a dependable guide to equipment purchases, it publishes in each January issue a complete and up-to-the-minute directory of warm air heating, air conditioning and sheet metal products and equipment. This directory lists all products used in the field, their trade names, and the full names and addresses of all manufacturers It is used by readers as a buying reference throughout the year.

Almost from the day interest in residential air conditioning began to develop, the advantages of the warm air type of heating system, with its duct distribution of air, were plain to see. It was adapted to all air conditioning factors, either through a self-contained central unit or through a central furnace to which could be added step-by-step or as a whole, fan, washer, humidifier, filters, controls, cooling, and automatic firing.

Today, as a result of this ready adaptability as well as economy, tens of thousands of homes have winter air conditioning—supplied through forced warm air heating with air cleaning and humidification. Cooling apparatus can be attached to these systems readily whenever complete, year-'round air conditioning is desired.

This trend in residential air conditioning has placed a premium on air handling knowledge, and has brought to the fore the one man experienced in "treating" air at a central place and getting it properly distributed—the warm air heating and sheet metal contractor. The warm air heating industry has, furthermore, undertaken and made notable progress toward the solution of the many new engineering problems involved All this has helped to put warm air heating in the center of residential air conditioning.

In aiding to develop this trend and assist in the solution of new problems, AMERICAN ARTISAN has provided a service to its field which has made it the recognized authority on residential air conditioning practice

To manufacturers whose products are used in residential air conditioning, AMERICAN ARTISAN offers full coverage of the leading buying factors. Such manufacturers will be interested in the market study called "What's What and Who's Who in Air Conditioning" For further information about this study, write to the address above

AMERICAN ARTISAN is published monthly. It is a member of the A. B. C. and A. B. P.

Subscription rates—$2.00 per year, $3 00 for two years in U. S., Canada, Mexico, Central and South America. Foreign $4.00 per year

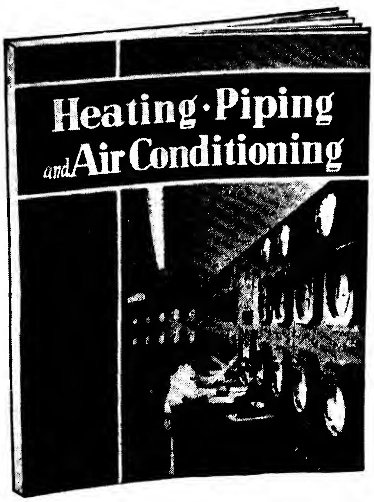
Advertising rates furnished upon request

Heating, Piping and Air Conditioning

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, **Chicago, Ill.**



HEATING, PIPING AND AIR CONDITIONING is the publication which carries in each issue the official JOURNAL OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in addition to its own regular editorial section.

Its field is that of industry and large buildings. Editorially, it gives specialized attention to the design, installation, operation, and maintenance of heating, piping, and air conditioning systems in such plants and buildings.

In addition, there is published in each January issue a complete Directory of Commercial and Industrial Heating, Piping and Air Conditioning Equipment, which lists all products used in the field, their trade names, and the full names and addresses of all manufacturers. This directory has been established as the industry's buying and specifying guide, and is consulted by readers throughout the year, whenever equipment purchases are up for consideration.

H. P. & A. C. is read by consulting engineers and architects . . . contractors . . . and engineers in charge of heating, piping, and air conditioning in industrial plants, large commercial and public buildings, federal, state, and city governments, school boards and public utilities. Among its subscribers are numbered all members of the A.S.H.V.E., who represent about 30 per cent of its total circulation.

Such a coverage means, for the advertiser, consideration at all points in the selling of a heating, piping, or air conditioning product . . . consideration in the selection of a product during the preparation of plans and specifications; consideration in the actual purchase of a product for installation; consideration in the year 'round buying of a product for operating and maintenance requirements.

It has been evident for some time that the air conditioning field is made up of two distinct markets: (1) Industrial and Commercial; (2) Residential.

These two markets are different in equipment used; different in engineering problems involved, different in engineering, distributing, and consuming personnel . . . require, therefore, different selling jobs.

To sell the industrial and large building field for air conditioning, the manufacturer must win acceptance from the engineers who design, specify, install, operate, and select the system to meet the particular requirements of the plant or building. The system may be central, unit, or "split," but it is these engineers who are the influencing or purchasing factors.

It is to such groups that HEATING, PIPING AND AIR CONDITIONING editorially caters—exclusively in the industrial and large building field. Without waste, the manufacturer of air conditioning products and accessory equipment, such as motors, drives, controls, etc., can reach through its pages those from whom he is seeking the necessary engineering acceptance.

These facts are clearly shown in a recently prepared market study titled "What's What and Who's Who in Air Conditioning" which will be presented to interested manufacturers upon request.

HEATING, PIPING AND AIR CONDITIONING is a member of the A. B. C. and A. B. P.

Subscription rates—$2.00 per year; $3.00 for two years in U. S., Canada, Mexico, Central and South America. Foreign, $4.00 per year.

Advertising rates furnished upon request.

Domestic Engineering Publications

1900 Prairie Avenue — Chicago, Illinois

DOMESTIC ENGINEERING

More than a half century devoted to serving the plumbing and heating industry . . . serving it faithfully and serving it well . . . that's the record of *Domestic Engineering*. This record provides the sound foundation upon which *Domestic Engineering* has achieved and maintained its commanding position of leadership over this long period . . . a leadership based upon *all* points by which a business publication may properly be judged.

The reader audience of *Domestic Engineering* is comprised of the top group of merchandisers who are responsible for the major portion of the business in the huge plumbing and heating market. These are the men who look upon *Domestic Engineering* as the outstanding publication in their industry, who depend upon it to keep them informed on all matters pertaining to their business. They have utmost confidence in *Domestic Engineering* and this confidence in turn is carried over to the advertising pages.

Complete data concerning *Domestic Engineering* and the industry it serves is available in "Selling the Plumbing and Heating Market." Write for your copy.

PLUMBING AND HEATING NEWS

Further expanding its service to the plumbing and heating field, *Domestic Engineering* is supplemented by PLUMBING AND HEATING NEWS, which reaches every known factor in the industry in mid-month.

In tabloid form PLUMBING AND HEATING NEWS has filled a void in the industry by bringing up-to-the-minute news of current developments and new products. This news content has made it the newspaper of the industry. Because of its universal circulation in the industry, it has aided in welding into one vast selling force, the factors that sell this huge market.

At an exceptionally low per-line-per-reader rate PLUMBING AND HEATING NEWS enables the manufacturer to blanket the entire industry with his sales message. Write for complete details, rates, etc.

Domestic Engineering Publications

1900 Prairie Avenue — **Chicago, Illinois**

AUTOMATIC HEAT AND AIR CONDITIONING

The development of the automatic heat and air conditioning industry to its present position in American business has been possible only through concentrated selling efforts and the creation of favorable public acceptance by the key men of the industry.

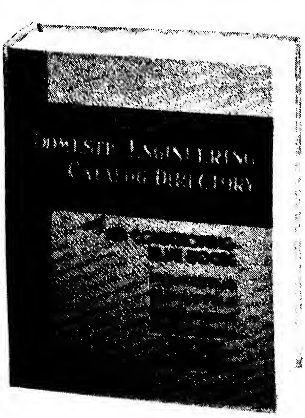
AUTOMATIC HEAT AND AIR CONDITIONING, the business paper of the industry, has played an important part in this rapid growth of its industry. Its many merchandising helps and the wealth of technical information contained in it each month enables the specialty sales organizations to do a better selling job.

This high grade editorial material and the fact that it reaches the *entire industry* makes AUTOMATIC HEAT AND AIR CONDITIONING an effective and economical means for the advertiser to place his sales story before all influencing factors of his sales. To further its value to the industry, this publication sponsors an extensive campaign to create, through newspapers and periodicals, a wide acceptance of the advantages of automatic heat and air conditioning equipment.

The complete marketing facilities of the MARKETING AND RESEARCH BUREAU are also available to aid sales and advertising executives in obtaining full information on markets and in building sales forces.

DOMESTIC ENGINEERING CATALOG DIRECTORY

Now enlarged and extended to incorporate automatic heating and air conditioning and related products, DOMESTIC ENGINEERING CATALOG DIRECTORY has taken another forward step in keeping abreast of the industry. This new edition is specifically designed for buyers, specifiers, specifying engineers, wholesalers, general contractors, air conditioning consulting engineers . . . those whose duty it is to purchase and specify plumbing, heating and air conditioning equipment.



Wholesalers, engineers and contractor-dealers, particularly, will appreciate the wealth of quick-reference purchasing and specifying data the 1939 CATALOG DIRECTORY makes available to them. As a method of presenting their catalog material, manufacturers, too, will welcome the consolidation of all of this buying and specifying data in a single volume which is placed in the hands of the top notch plumbing, heating and air conditioning buyers and specifying engineers.

Due to the trend of distribution of air conditioning equipment, DOMESTIC ENGINEERING CATALOG DIRECTORY as now extended and enlarged, fulfills an opportune desire of the trade . . . and the industry is ideally situated to accept and use this all-purpose, comprehensive buying and specifying guide. An interesting 8 page folder presenting the CATALOG DIRECTORY acceptance and preference among wholesalers will be sent upon request.

Air Conditioning & Oil Heat

232 Madison Ave. **Lex. 2-4566** **New York, N. Y.**

Chicago
903 Merchandise Mart
Delaware 9389

San Francisco
DON HARWAY & Co
155 Montgomery St
Exbrook 6029

Los Angeles
DON HARWAY & Co
318 W Ninth St
Tucker 9706

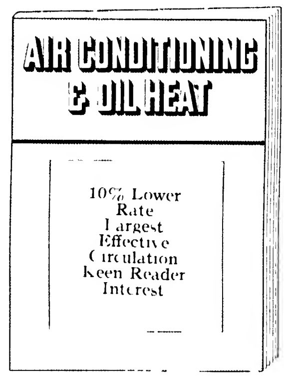
Baltimore
Candler Bldg
Plaza 7065

PRIOR to 1935, this publication was known as "OIL HEAT" and its editorial content consisted entirely of articles about the manufacture, sale and installation of oil burners The title was expanded to its present form in 1935, and inspired a change in the oil burner field The oil burner manufacturers and dealers are a naturally progressive group, else they would not be in the oil burner business at all They understood the modern merchandising and technical problems presented by the sale of air conditioning equipment, and they entered vigorously into this new activity.

The result is that today (January 4, 1939) a total of 44 2 per cent of the oil burner manufactures are selling (many of them also manufacturing) air conditioning equipment And 6698, or 54 4 per cent of the 12,298 oil burner dealers now are selling some form of air conditioning

Editorial content includes articles on "Air Conditioning Schools," "Winter Air Conditioning Equipment – Sale, Installation and Service;" "Summer Air Conditioning—Application, Sale, Dealer Problems;" "Oil Burner Service; Dealer Problems; Selling and Installing;" News items about all forms of heating, air conditioning and firing equipment. A complete coverage of the industry's activities each month. *Subscription price $2 per year.*

Surveys and publications available 16-page booklet on "Answers to your Questions About Winter Air Conditioning," 16 page booklet entitled "What the Air Conditioning-Oil Burner Dealers are Thinking and Doing About Sheet Metal Shops, Ducts, Registers, Blowers, Etc ," January, 1939, Annual Forecast & Statistical Issue —a report on 1938 and intelligent opinions about 1939, Directory of Oil Burner and Air Conditioning Manufacturers, giving equipment specifications, officers' names, trade names, etc.



"The paper with the bright orange cover"

Circulation of this paper
(Publisher's Statement)

Air Conditioning & Power Oil Burner Mfrs	423
Additional Mfgr Executives	183
Air Conditioning & Oil Burner Accessory Mfrs	145
Mfgs Branch Offices & Field men	148
Air Conditioning Depts of Public Utilities	107
Combination Power Oil Burner and Air Conditioning Dealers	6698
Additional Power Oil Burner Dealers (not yet handling air conditioning)	5600
Additional Air Conditioning Dealers (not yet handling oil burners)	1040
Wholesalers and Distributors of Air Conditioning, Oil Burner and Heating Supplies & Accessories	1,172
Advertisers, Agencies, Unclassified	1300
Total	17,116

"Our two previous ads in AIR CONDITIONING & OIL HEAT secured us approximately $30,000 of new business." Dewey-Shepard, P. 53, May, 1938, issue.

"We have been pleasantly surprised with the tremendous response to our ads in the August and September issues of AIR CONDITIONING & OIL HEAT" Eds-Wil Burner Corp , Chicago, Sept 9, 1938.

AIR CONDITIONING & OIL HEAT also has a Mailing Service, under which, for very low cost, literature may be mailed to all of the groups indicated in the Circulation Listing, above.

OILHEATING & AIRCONDITIONING *fueloil* JOURNAL

Published Monthly at

420 Madison Avenue

New York

MARKET: The oilheating market is a closely knit 4-way market—oilburners, heating, airconditioning, and fueloil. The modern and progressive oilheating dealer sells all four—a good oilburner, using good fueloil, firing a good heating or airconditioning system.

From 1919 to 1930, the only oilheating product sold by burner dealers was the conversion burner. In 1930 the sale of conversion burners represented 77.3 per cent of the dealers' gross income.

By the end of 1938, the average oilheating dealer got only 28.7 per cent of his income from conversion burners. But, beginning in 1932, he had added three other major oilheating lines—heating, fueloil and winter airconditioning. 1938 gross dollar volume of the average dealer was divided:

Conversion burner units	28.7 per cent
Heating equipment, including boiler-burner units	26.2 per cent
Fueloil	28.5 per cent
Winter airconditioning, including furnace-burner units	16.6 per cent

In 1938, 42 per cent or 57,141 conversion oilburners were sold with new cast iron or steel boilers. In addition, dealers sold 10,934 boiler-burner units. Total boiler sales by oilheating dealers increased 11 percent over 1937. These dealers did a winter airconditioning dollar volume in 1938 of $21,324,383.

SERVICES FOR ADVERTISERS

Key Market Studies.

Merchandising News.

Specific Products Reports.

Unit Sale Brand Preference Studies.

Booklets, reprints of special articles.

OILHEATING & AIRCONDITIONING: FUELOIL JOURNAL covers this integrated 4-way market.

It is the oldest paper in the field—established 1922. Editorially, it has consistently fostered every progressive development in the field and it has encouraged the trend to the complete oilheating dealer.

Every issue is carefully balanced editorially to cover the dealers' need for usable information on all four sides of his business.

Heating equipment manufacturers have long known FUELOIL JOURNAL as a powerful sales aid. Its reader interest is unique among trade papers.

CIRCULATION: Like its editorial content, the circulation of FUELOIL JOURNAL is carefully controlled to give complete coverage of this great 4-way market. A detailed breakdown from the latest circulation statement (June 30, 1938) shows:

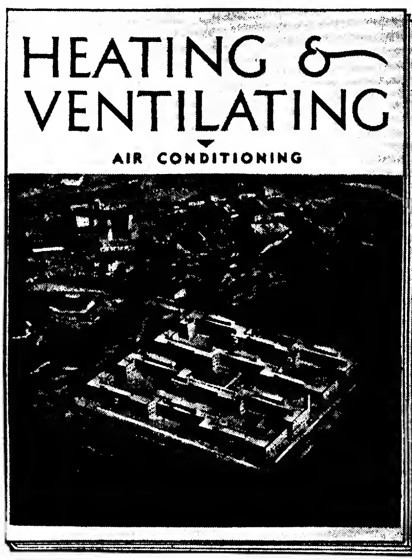
Power oilheating and airconditioning dealers and distributors	10,764	
Key heating contractors, plumbing and heating contractors, and engineers	1,866	
Fueloil distributors, selling fueloil and range oil, and their branches	3,107	
Accessory and heating supply distributors	1,075	
Total dealers and distributors		16,812
Power oilheating and airconditioning manufacturers and their executives,	587	
Accessory manufacturers	428	
Total manufacturers		1,015
Total dealers and manufacturers,		17,827
Per cent of total circulation		97.27
Other miscellaneous		501
Grand total		18,328

FUELOIL JOURNAL circulation covers the automatic heating field at the minimum rate per thousand copies. It will pay you well to get full details. Write, wire or telephone.

HEATING & VENTILATING

AIR CONDITIONING

THE INDUSTRIAL PRESS . . . *Publisher*
140-148 Lafayette St. **New York, N. Y.**



HEATING & VENTILATING reaches the "key men" of the industry—the engineers, contractors and manufacturers who have the final word in the specification, installation, production and maintenance of the mechanical equipment utilized in the heating, ventilating and air conditioning fields.

An editorial program of outstanding alertness and authority is directed by qualified heating and ventilating engineers. Special sections and timely feature articles are included from time to time in line with the forward-looking policy that has characterized the publication since its inception in 1904. News, trends, developments, personalities—every side of this important industry is faithfully and authoritatively reported in this outstanding publication.

There is a regular section devoted to new equipment, profusely illustrated and comprehensively reported. Degree-days and unit fuel consumption for various large cities in the country has been a regular monthly feature of the publication for over eight years. The weather in large cities in typical localities of the country is accurately charted. Two pages of reference data appear in every issue. Reports of meetings, the activities of manufacturers, abstracts of current papers, books and pamphlets, and editorials on the planning, installation and operation of heating, ventilating and air conditioning systems in public buildings, offices, factories, schools, hospitals and homes are other contents of continuous interest.

Air conditioning, now coming into its own, has had a champion in HEATING & VENTILATING since 1904 when, in its very first issue, an article on this then infant industry appeared. Since that time, for more than thirty years, HEATING & VENTILATING has consistently published the news and developments of air conditioning up to its present high state of perfection and its pages have carried an impressive total of editorial lineage on this subject.

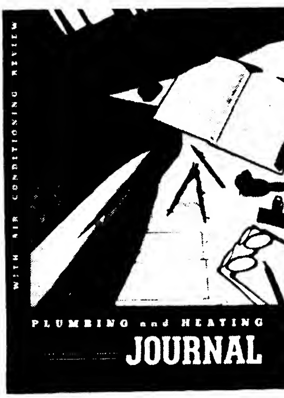
Subscriptions to HEATING & VENTILATING are $2.00 a year. Advertising rate cards, sample copies and market data will be gladly submitted on receipt of application.

Plumbing and Heating Journal

Published by

THE ANGUS CO., INC.

515 Madison Ave., **New York City**



PLUMBING and Heating Journal is edited to furnish a well-rounded, efficient service to the men engaged in the plumbing, heating, ventilating and air conditioning fields. To this end, it covers both the technical and business phases of their work, as well as many minor but exceedingly important ones.

It gives free technical service through a staff of practical engineers; expert merchandising assistance, and its technical and business articles are by men of recognized competence.

Thousands of readers come to THE JOURNAL each year for solutions to their technical problems and while some of the questions and answers are published in the Readers' Technical Service section in each issue of the magazine, the vast majority of them—having to do with practically every phase of heating, ventilating and air conditioning as well as plumbing—are answered by mail, because most of the requests for help are urgent and a delay in answering would, in some cases, entail actual monetary loss to the contractor.

The Readers' Technical Service Department of THE JOURNAL is staffed by editors who have spent their lives in the business; men who were successful plumbing, heating, ventilating and air conditioning engineers before they wrote a line for publication, and who now devote their entire time to keeping abreast of the field's technical developments and using their knowledge and experience for the benefit of JOURNAL subscribers.

The technical service rendered its readers by THE JOURNAL is closely paralled by what it strives to do for them in a business way, for it also publishes many authoritative articles on and answers questions concerning the various ramifications of business management and prints as part of every issue, a special section devoted to selling.

One associate editor spends his entire time in the field writing articles on the business management problems of JOURNAL readers, and the practical solution of those problems.

Supplementing the business and technical articles is a large amount of exclusive, staff-gathered news that high-lights the background of the trade's activities.

This news background is vital. It completes the industrial picture for the reader. It keeps him in intimate touch with what the various important associations and his fellow members of the craft are doing throughout the nation and it charts the trends that are likely to have a very definite influence on the future operation of his business.

THE JOURNAL editorial department draws its news from over a hundred trained correspondents located at strategic points throughout the country—by far the largest group of exclusively editorial workers used by any paper in the industry.

It is this combination of the technical, business and news aspects of the industry that enables THE JOURNAL to achieve a finely balanced magazine that gives the reader the type of information he wants and needs, in brief, compact, time-saving form.

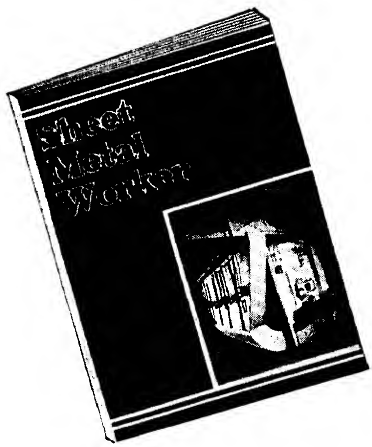
THE JOURNAL is a member of the A. B. C., and costs $2.00 per year by subscription.

Sheet Metal Worker

Published by Edwin A. Scott Publishing Company

45 West 45th Street

New York



THE January 1939 issue of SHEET METAL WORKER will be its Sixty-Fifth Anniversary and Directory Number. It is the oldest publication in its field and is of vital importance to men interested in sheet metal work—air conditioning—warm-air heating and ventilation. Founded and published to 1909 by David Williams Company; 1909 to 1920 by United Publishers Corp.; since 1920 by the present publisher, the Edwin A. Scott Publishing Co.

SHEET METAL WORKER is today a monthly merchandising, business and technical journal basic to the use of sheet metal. It serves the various unified merchandising and installing branches of the industry, consuming sheet metal for the erection, maintenance and operating equipment of homes and buildings, including central air conditioning equipment, warm-air heating, ventilating, dust and refuse removal, and systems for handling material by air; kitchen and restaurant work; a wide variety of interior and exterior work for commercial, industrial, institutional, and residential buildings.

Subscribers are mainly merchandising contractors purchasing practically all products and equipment which they fabricate, erect or install. Manufacturers, jobbers and distributors also subscribe.

The market has three main divisions.

(1) Equipment for resale in connection with erection or installation work.

(2) Materials for fabrication.

(3) Shop equipment and supplies.

CIRCULATION

SHEET METAL WORKER is a member of the Audit Bureau of Circulations and the Associated Business Papers. It has a uniform distribution, with the greater part of its circulation centered in states showing the greatest industrial activity. Readers of SHEET METAL WORKER are made up of warm-air heating, air conditioning and sheet metal contractors and dealers. Also wholesalers, manufacturers, branch offices and salesmen. For further details send for ABC statement.

EDITORIAL

SHEET METAL WORKER has been outstanding in the editorial service it has rendered the trade and is noted for the practical usefulness of its articles and the timeliness of its editorials. Its editor is a noted author in this field and the author of several well-known books.

SHEET METAL WORKER also publishes books on heating, ventilating, sheet metal work, air conditioning, etc.

The Annual Issue published in January, contains a comprehensive and valuable Directory Section.

ADVERTISING

SHEET METAL WORKER has an enviable record of long term advertising and is proud of its long list of regular advertisers.

Because of its intimate contact with this field, SHEET METAL WORKER is well qualified to cooperate with manufacturers in their sales and advertising programs.

Subscription rates—$2.00 per year, U.S., Mexico and Canada; Foreign, $3.00.

Advertising rates on request

Buffalo Pumps, Inc.

450 Broadway, **Buffalo, N. Y.**

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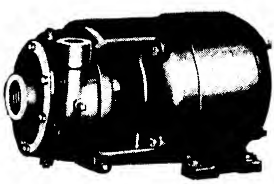
TOLEDO, OHIO, 1922 Linwood Ave., C. M. Eyster
WASHINGTON, D. C., 820 Woodward Bldg., G. S. Frankel

COMPLETE LINE MANUFACTURED IN CANADA BY CANADA PUMPS, LTD., KITCHENER, ONT.

PRODUCTS—A complete line of Single and Multi-stage Centrifugal Pumps, Steam Pumps and Special Pumps for use in all types of heating and air conditioning installations.

Buffalo Single Suction Closed-Coupled Pumps



This pump is close-coupled to electric motor, eliminating the necessity for bearings. The impeller is overhung on the motor shaft, providing a compact, easily-serviced unit. Permanent alignment is assured and the pump mounted in this manner requires very little space.

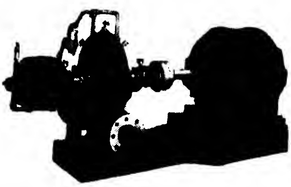
Buffalo Close-Coupled Pumps are suitable for handling hot water with low submergence on suction, or for operating with suction lift as high as 25 ft.

These pumps are also available in special alloys.

Buffalo Double Suction Single Stage Centrifugal Pumps

These pumps embody all of the accepted modern features of centrifugal pump design. Built for capacities from 10 to 50,000 U.S. gal per minute. Recommended for almost any service where clear water is handled. High efficiency and absolute reliability are assured.

Buffalo Self-Priming Single and Double Suction Centrifugal Pumps



Buffalo Single and Double Suction Pumps can now be had with positive self-priming device built with the pump. This primer is built under license from the Nash Engineering Company, and fully covered by patents.

Self-priming pumps have these advantages: (1) All working parts are above the liquid to be pumped. (2) There is complete access to all parts of installation. (3) Rotors are balanced—vibrationless. (4) Buffalo Self-Priming Pumps are very quiet—no long shafts to vibrate and fewer bearings. (5) Constant positive prime obtained without foot valves.

Buffalo Automatic Sump Pumps

Buffalo Sump Pumps are self-contained and have unusually high efficiencies thus permitting the use of small motors. Ball Bearing thrust and enclosed shaft especially adapt these pumps for their service.

Chicago Pump Company

2330 Wolfram Street **BRUnswick 4110** **Chicago**

PRODUCTS—Return Line Vacuum Heating and Boiler Feed Pumps, Condensation, House, Booster, Fire Pumps, Circulating, Brine, Sewage, Bilge, Sludge, Pneumatic and Tankless Water Supply Systems and Automatic Alternator for Duplex Sets of Pumps.

"CONDO-VAC"

Return Line Vacuum Heating and Boiler Feed Pump

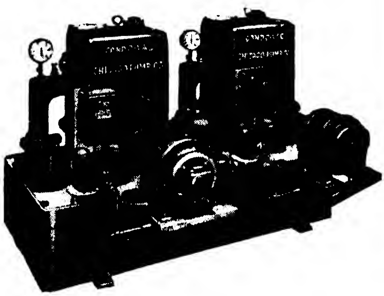


Fig. 2102—Duplex "Condo-Vacs" with Duplex Double Automatic Control

No vacuum on stuffing boxes, ample clearance in rotating member. It costs less to operate a "Condo-Vac." "Condo-Vac" reduces corrosion in piping and boiler to minimum—because pump does not take in air from atmosphere and entirely eliminates all air coming back from system. "Condo-Vac" is quiet, has a low inlet, entirely automatic, fool-proof, easy to maintain. *Ask for bulletin 270.*

Close-Coupled Pumps

Boiler Feed, Circulating, Tank Filling, Water Supply

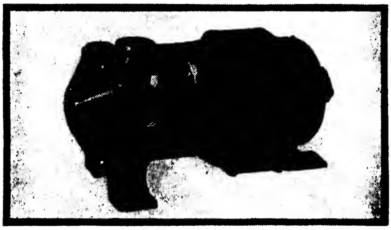


Fig. 2130—Close-Coupled, side suction pump. Capacities range from 3 to 600 Gpm against heads up to 189 ft. Motors from 1/6 to 20 Hp. Discharge 1 to 3 in. Both closed and open type impellers.

"Sure-Return" Condensation Pump

for Low and Medium Pressure, and Systems up to 35,000 Sq Ft Radiation

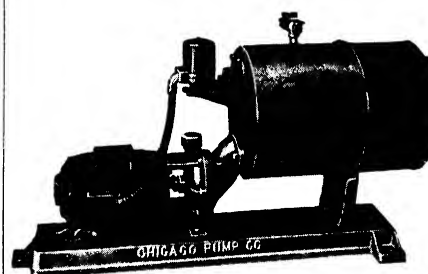


Fig. 1946

"Sure Return" Condensation Pumps and Receivers are built for systems up to 35,000 sq ft of direct radiation and for low and medium pressures. Built in either single or duplex units. Duplex units are alternated in their operation by the Automatic Alternator. *Complete data in Bulletin 250.*

Vertical Condensation Pumps

for Low and Medium Pressure for Systems from 500 to 100,000 Sq Ft Radiation

Fig. 1940 Vertical Condensation Pump

The vertical condensation pump is designed to receive returns from lowest radiation. The receiver is placed underground—an ordinary hole sufficing if necessary — and requires very little floor space. Unit is shipped complete, easy to install, assembled so as to prevent steam leaks. Special bearings will stand up under hot water for several years. A special float mechanism is guaranteed not to leak or stick in stuffing box. *Complete data and description in Bulletins 245 and 255.*

Decatur Pump Co.

Decatur, Ill.

BURKS SUPER TURBINE PUMPS AND WATER SYSTEMS

BURKS SELF PRIMING PUMPS
"Only one moving part"

Burks Heavy Duty Self Priming Super Turbine Pumps

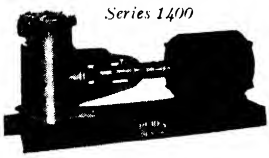
Pressures to 100 lb per square inch. Capacities to 1,700 gal per hour. Only one moving part, the bronze impeller. A general utility pump suitable for many applications, including domestic water systems, commercial building installations, booster and hot water service. Ideal for returning condensate to heating boilers.

Ask for Bulletin No. 115-A.

Series 1100-M

Will handle air along with water, hence will not air or steam bind. Suction lift ability unsurpassed. A pump for service where powerful, economical and efficient performance are first considerations.

Series 1400



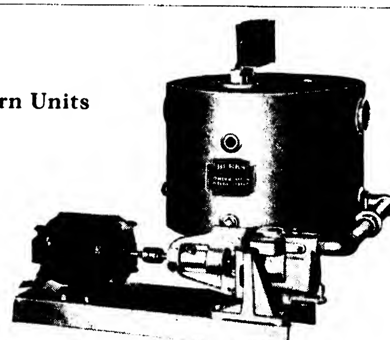
Burks Self Priming Centrifugal Pumps

High efficiency, semi-open-impeller type, positive self-priming Centrifugal Pumps having many outstanding features. Capacities to 24,000 gal per hour.

Ask for Bulletin No. 115-A.

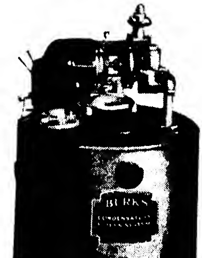
Burks Super Turbine Condensation Return Units

Furnished with receiver tanks constructed of copper bearing steel. Automatic float switch governs the operation of pump motor.



Designed to operate against pressures up to 100 lb per square inch. Balanced Hydraulic Load. Will not steam bind.

Traditional Burks quality insures trouble free performance and maximum efficiency.

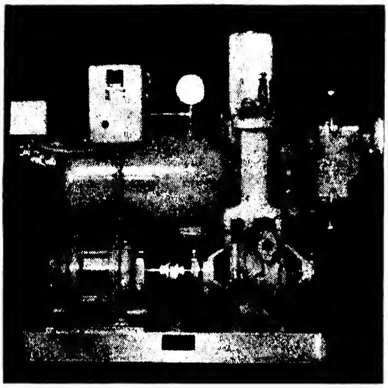


Ask for Bulletin No. 104-C covering complete line return units.

The Nash Engineering Company

South Norwalk, Conn., U. S. A.

Sales and Service Offices in all Principal Cities



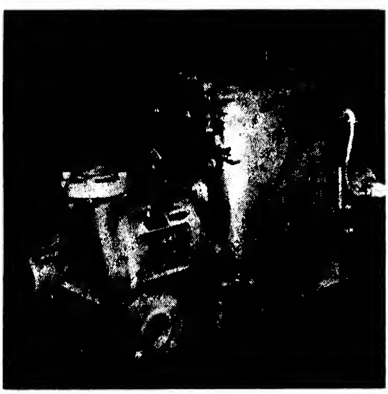
Jennings Return Line Vacuum Heating Pumps

Standard with the heating industry for over sixteen years. They remove air and condensation from the return lines of vacuum steam heating systems, discharging the air to atmosphere and returning the water to the boiler.

Two independent units are combined in a single casing—an air unit and a water unit. Impellers of both are mounted on the same shaft. The pump is bronze fitted throughout.

Supplied either direct connected to standard electric motors, for belt drive, or for steam turbine drive. For continuous or automatic operation against pressures up to 40 lb. Supplied standard in capacities up to 300,000 sq ft E.D.R.

Complete data in Bulletin No. 264 on request.

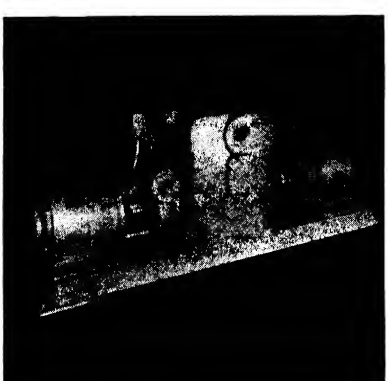


Jennings Vapor Turbine Vacuum Heating Pumps

The Jennings Vapor Turbine Heating Pump combines all of the advantages of the standard return line heating pumps with a new type of drive, a specially designed low pressure turbine which operates directly on steam from the heating mains on any system, requiring a differential of only 5 in. of mercury, and returns that steam to the heating system with practically no heat loss.

This pump affords the safety and economy which goes with a continuous condensation return and steady vacuum, and at no cost for electric current. Furnished standard in capacities up to 65,000 sq ft E.D.R. Over 65,000 sq ft and up to 150,000 sq ft, information will be furnished upon request.

Complete data in Bulletin No. 290 on request.



Condensation Pump and Receiver

Removes condensation from radiators in return line steam heating systems and pumps condensation back to the boiler.

They are sturdy and compact in construction, and combine receiving tank, pump and driving motor in a single assembly. Bronze fitted throughout, with Tobin bronze shaft. Impeller is of special design adapted to handling hot water with highest efficiency.

Jennings Condensation Pumps are furnished in standard sizes with capacities ranging from 1½ to 225 gpm of water. For serving from 1,000 up to 150,000 sq ft of equivalent direct radiation.

Complete data in Bulletin No. 241 on request.

The Nash Engineering Company

South Norwalk, Conn., U. S. A.

Sales and Service Offices in all Principal Cities

Centrifugal Pump

Made in standard and suction (self-priming) types. For circulating hot and cold water; boosting city water pressure; handling water in air washing and conditioning; handling ash sluicing water, etc.

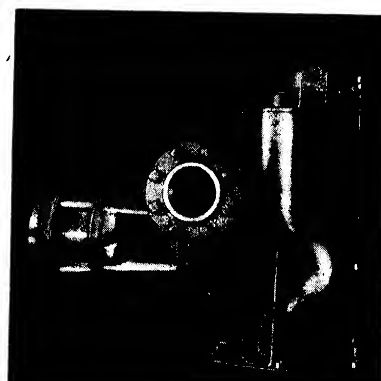
Compact—motor armature and pump impeller are mounted on the same shaft. Simplified—no bearings in pump casing, one stuffing box. Accessible—impeller removable without disturbing piping or shaft alignment.

Self-priming types will handle air or gas continuously with liquid being pumped, and can be operated intermittently without foot valve.

Supplied in 1¼, 1½, 2, 3, 4, 6, and 8 in. sizes with capacity up to 2000 gpm. Heads up to 300 ft.

Complete data in Bulletin No. 155 on request.

Suction Sump and Sewage Pumps



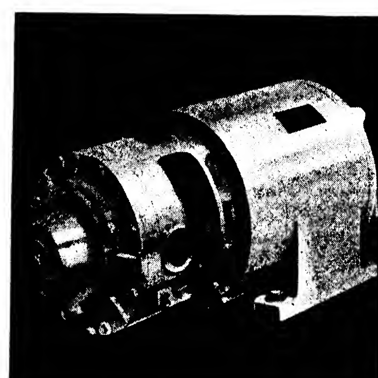
Jennings Suction Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids. The Suction Sewage Pumps are equipped with a non-clog type impeller for liquids containing solids. Suction piping only is submerged. Centrifugal impeller and vacuum priming rotor are both mounted on same shaft that carries rotor of the driving motor, forming a single moving element and rotating without metallic contact.

These pumps will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit. This affords perfect accessibility for inspection or cleaning.

Capacities to meet all requirements.

Complete data in Bulletins 159, 161 and 275 on request.

Air Compressor and Vacuum Pump



The Nash Air Compressor operates on a unique and different principle. The one moving part rotates in casing without metallic contact. There are no valves, pistons, or sliding metal vanes. There is nothing to wear, and no internal lubrication. Nash Compressors deliver absolutely clean air.

Unit illustrated is built integral with electric motor. Compact, may be installed anywhere. Ideal general service compressor. Suitable for priming pumps on water systems, handling CO_2 gas, agitation of liquids, as blood sucking pumps in hospitals, etc.

Pressure 75 lb or vacuum 27 in. of mercury. Equipment furnished for any capacity. Special equipment for higher vacuums and pressures.

Complete data in Bulletins Nos. 252, 255, 258 and 282 on request.

Pomona Pump Co.

Manufacturing Plants

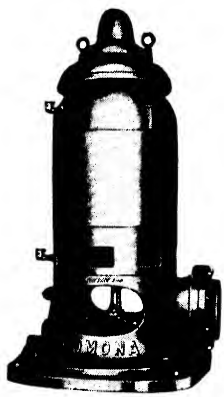
ST. LOUIS, MISSOURI — POMONA, CALIFORNIA

Branch Offices

NEW YORK — CHICAGO — ATLANTA — HOUSTON
LOS ANGELES — SAN FRANCISCO

PRODUCTS

Pomona Deep Well Turbine Pumps; Niagara Low and Medium Lift Pumps



Applications: Pomona Turbine Pumps are used for air-cooling and washing service. Being water lubricated, they operate most successfully in conjunction with water softeners for building water supply. Pomona Niagara Pumps are used for flood protection, condenser service and any other purpose where large volumes of water must be moved rapidly.

Sizes: Pomona Deep Well Turbine Pumps are made in all sizes for wells of any diameter from 4 in. to 32 in. or larger and for any depth to 1000 feet or over.

Capacities: 15 gpm to 10,000 gpm, 3 to 1000 hp.

Drive: Normally by unidrive integrally built electric motor, for any current characteristic; 3-phase induction; 3-phase slip-ring; 3-phase synchronous; single-phase repulsion induction; single-phase capacitor; 2-phase. Voltages 110 and 220 on single-phase; 220, 440, 550, 2200 and 4400 on 3-phase. Direct connected steam turbine drive can be furnished. Simplified construction makes automatic control easily accomplished.

Column Pipe: Welded copper-bearing steel made to Pomona specifications and standardized in interchangeable 10-foot lengths. Seamless steel couplings. Column pipe ends fit bronze bearing retainer, insuring straightness and correct alignment with line shaft.

Drive Shaft: High tensile, cold rolled steel with stainless metal sleeves at all bearing points. Open line shaft.

Underground Bearings: Patented resilient revolvable rubber bearings held in precision bronze bearing retainers. Low coefficient of friction. Minimum wear. Water lubricated. Self flushing.

Impellers: Semi-open bronze, adjustable for capacity from the top of pump.

Bowls: Special Pomaloy cast iron made in the Company's own foundry under laboratory control. Especially designed for resistance to graphitization, abrasion and other forms of corrosion.

Packing Gland: Located at the lower part of the pump head, *above ground*, fully accessible.

Non-reversing Ratchet: Located above the motor at the top of the drive shaft. Positively prevents damage to line shaft or couplings in case of accidental reversal of rotation from any cause.

NIAGARA PUMPS

For low or medium lifts. Capacities from 900 gpm to 100,000 gpm. Made in two types: propeller and Mixflow. Propeller type is for heads up to 15 feet. The Mixflow type is for heads from 15 to 70 feet or higher.

Pomona Pump Co. has distributors and service men in important cities throughout the world. Write us for the name of your nearest distributor.

Anemostat Corporation of America

10 East 39th Street, **New York City, N. Y.**

THE ANEMOSTAT HIGH VELOCITY AIR DIFFUSER

The Anemostat High Velocity Air Diffuser is a ceiling outlet consisting of a series of circular diverging metal cones opening outward from a central circular neck which may be attached directly to the main or branch duct.

The Anemostat assures draftless distribution of air at any duct velocity. Various standard sizes from 2 in. to 38 in. neck diameter will distribute volumes of air between 10 cfm and 25,000 cfm and will handle any velocities between 300 fpm and 4000 fpm. When introducing large quantities of air into a room air motion results. The series of cones which form the Anemostat discharge the air in definite proportions in all directions in a series of planes. This diffusion together with the aspiration (suction) effect causes prompt equalization of temperature and therefore, humidity throughout the room, horizontally, and definitely prevents air pockets and dissipates the evaporation aura around the human body.

The air-mixing effect of the Anemostat causes the predetermined room temperature to be established at a point well above the breathing level, which permits the use of higher temperature differentials. This in turn results in smaller volumes of air to be conditioned and therefore in smaller plants, reduced operating expenses and smaller ducts, while the high velocities which may be employed because of the draftless diffusion, result in further reduction of duct sizes and simplification of duct layouts.

At the different velocities recommended for rooms used for different purposes the increase in decibel ratings through the use of Anemostats is negligible.

The Anemo-lite which is an Anemostat combined with a built in lighting unit is an ideal solution to the combined problem of Air Distribution and Lighting. (See Figure 1).

Pendent lighting fixtures may be hung directly from the center cone of the Anemostat if desired. (See Figure 2).

Particularly suitable for theatres and auditoriums is the Anemostat combined with the indirect lighting unit. (See Figure 3). With this combination unusual and effective results are easily obtained.

Complete technical information on the Anemostat combined with lighting fixtures is available upon request.

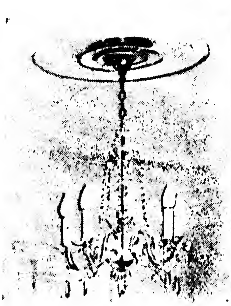


Fig. 2

Fig. 1

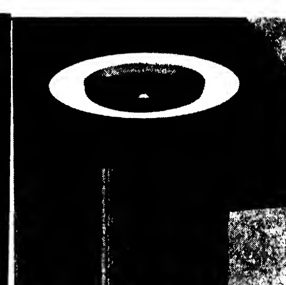


Fig. 3

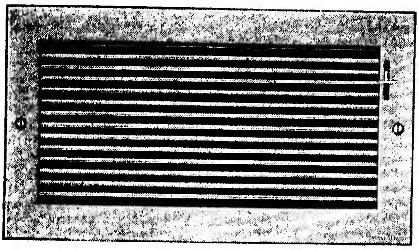
"No Air Conditioning System is better than its Air Distribution"

The Auer Register Co.

3608 Payne Avenue, **Cleveland, Ohio**

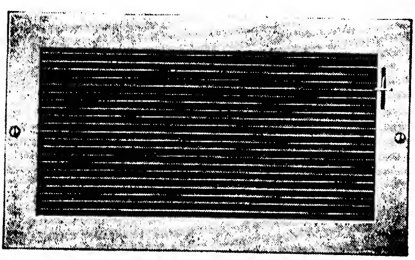
Manufacturers of Registers and Grilles for Gravity and Air Conditioning Systems; Wrought Metal Grilles for Concealing and Protecting Radiation

AIR CONDITIONING REGISTERS AND GRILLES



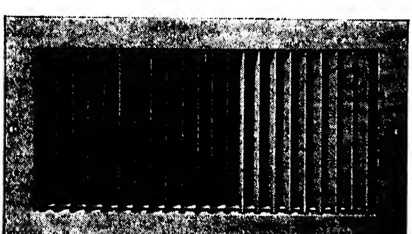
Fin-Flex No. 5030 Register with Band Iron Frame

Flexible fins 3/8 in. on center offer satisfactory one-time adjustment. Adjusting tool furnished with every order. Horizontal fins, furnished as standard. Vertical fins furnished if specified. Same design furnished also without valve, as a return.



Fin-Flo No. 9030 Register with Band Iron Frame

Horizontal Fin-Flo for upward and downward deflection is standard. Vertical Fin-Flo for two-way side deflection also furnished. Same design furnished also without valve, as a return.



Dura-Flo No. 8132 Register—No Frame

Adjustable bars 1/2 in. on center.
Also furnished with horizontal bars (adjustable). Small, convenient adjusting tool furnished with each order. Same design furnished also without valves, as a return.

The Auer line of registers and grilles for heating and air conditioning systems is modern and complete, offering a wide choice of styles for every purpose. Only a few representative models are shown on these pages.

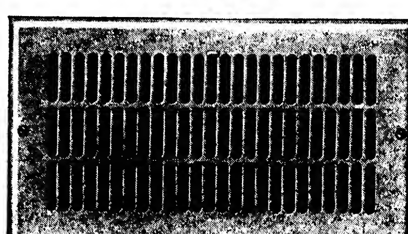
Fin-Flex Registers and Grilles are made with either vertical or horizontal fins which are easily adjusted at time of installation for single or multiple air current in any direction. Ample free air capacity for modern forced air systems.

Dura-Flo Registers and Grilles are also furnished with blades adjustable for any desired air flow. Fin-Flo Registers and Grilles are furnished with either vertical or horizontal fins for any specified air flow, and are not adjustable.

DuraBilt Floor Registers and Cold Air Faces are assembled with steel cross-bar construction, all cross joints locked and mortised. These should be specified wherever extra strength is required. They come in medium or narrow mesh.

The Auer Classic face has wide popularity for air conditioning and heating uses. It is an unusually attractive face, appropriate to most interior decorative schemes.

All Auer models are designed with due regard for air capacity, and supplied in all required sizes and finishes. Complete Catalog 39, showing all types for air conditioning and gravity heating, furnished on request.



Classic No. 2030 Register (with Valve) Band Frame

Hendrick Manufacturing Company

Hendrick Perforated Metal Grilles

48 Dundaff Street, Carbondale, Pa.

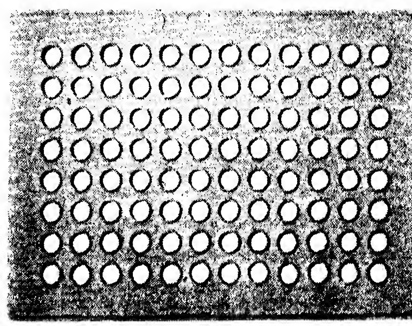
SALES OFFICES IN PRINCIPAL CITIES—CONSULT TELEPHONE DIRECTORY

PRODUCTS—Hendrick Perforated Metal Grilles; also Mitco Open Steel Flooring, Mitco Armorgrids and Mitco Shur-Site Treads.

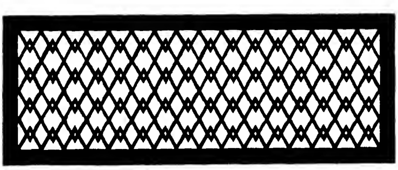
Hendrick Perforated Metal Grilles: To engineers, architects, contractors, building owners, Hendrick offers a large variety of grille designs. In addition to those patterns which have become standard, the Hendrick line includes many exclusive designs.

All Hendrick Grilles are characterized by clean-cut perforations and fine finish. They are given a special flattening operation which makes for easy and pleasing installation.

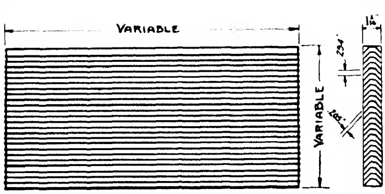
Hendrick Grilles are available in aluminum, brass, bronze, copper, Monel, stainless steel, steel and other commercially-rolled metals. They are supplied unpainted or with prime coat, with lacquer or duco finish in any color, with natural polish or with any standard electroplate finish. Furnished from 16 gauge to 5/16 in. thick, up to 90 in. wide and almost any length, dependent only on rolling mill limits. They come with invisible access doors, angle frames, hinged grilles, etc.



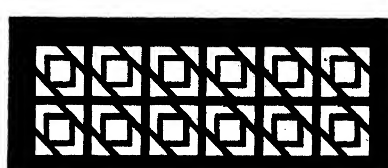
Hendrick Nozzle Grille; Recommended for air conditioning systems requiring grilles for high velocities; particularly efficient in minimizing the danger of noise from air passing through the grille. Fabricated from aluminum, bronze, stainless steel or steel, in gauges not exceeding .078 in. thick and in sizes not exceeding 48 in. x 120 in. 5/8 in. diameter hole is the most popular perforation but this nozzle grille can be furnished in many other sizes.



M. No. 9; 67% Open Area.



Door type, Fixed Louvre Grille; installed in a door this grille permits air circulation but prevents vision from any angle through the grille. Regularly furnished in No. 18 U. S. Gauge Steel with prime coat, enamel or electroplate finished. On special order can be furnished in aluminum, solid bronze or stainless steel .05 in. thick only.



M. No. 7 (Design Patent No. 91,266) 47% Open Area.

Send for a copy of new, 192 page handbook, "Hendrick Grilles."

Hart & Cooley Manufacturing Co.

Established 1901

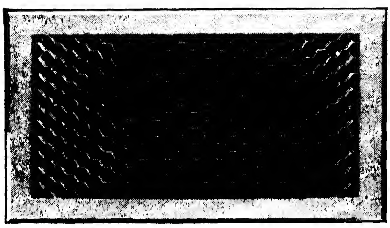
Air Conditioning Registers and Grilles - Warm Air Registers
Damper Regulators - Furnace Regulators - Pulleys - Chain

Engineering Office and Factory

Holland, Mich.

Chicago Office: 61 West Kinzie Street

Seven Complete Lines of Air Conditioning Registers and Grilles in Five Distinct Price Groups.



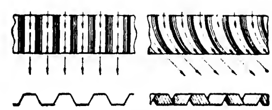
No. 90 DESIGN
Dual Control Directional Flow

THE ACE OF AIR CONDITIONING GRILLES

No. 90 Design—Made up of a number of thin strips which are shaped into a series of grooves—straight or curved—as shown. The assembled strips form an exceptionally attractive grille with openings of ½ in. and a depth of 1 in. A wide range of deflections is available. The tubular shape of the openings insures *positive control* of air flow with all deflections. Price Group E.

CHARACTERISTICS OF No. 90 DESIGN

1. Dual Control of Air Flow—The air is controlled in two planes; horizontally as well as sideways. Thus, on high sidewall installations, the air is prevented from striking the ceiling, thus avoiding discoloration and loss of directional flow.



Straight *Curved*

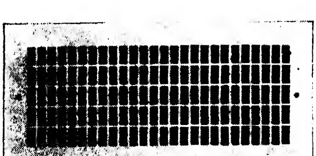
2. Resistance—Curved shape of tubes eliminates turbulence; hence, resistance is remarkably low.

3. Noise—The absence of turbulence likewise eliminates the greatest cause of noise in the grille. See H & C Catalog No. 37AC for Acoustica Ratings.

4. Concealment of Duct—The 1 in. depth of the grille, plus the tubular shape of the openings, results in exceptional concealment of the duct.

5. Free Area—The free area of No. 90 Grille is exceptionally large. See Page 39 of H & C Catalog No. 37AC, or Page 74 of Catalog No. 37 for Actual Free Areas.

6. Types of Directional Flow—Practically any deflections or combination of deflections are available. See Page 9 of H & C Catalog No. 37AC, or Page 65 of Catalog No. 37 for standard types of deflections.



No. 68 DESIGN
Perforated Plain Lattice

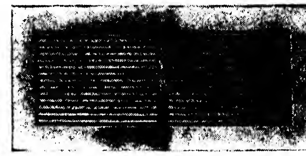
No. 68 Design—A very inexpensive grille combining ample free area, attractive appearance and effective concealment of the duct. Openings are ⅜ inches by ⅞ inches with ⅛ inch frets. Price Group AA.

No. 72 DESIGN
Non-Adjustable Vertical Bar Open Mesh

No. 72 Design—Combines rigid bar type construction, neat appearance, maximum free area with low cost. Depth of bars is ⅜ in. Price Group A.

No. 84 DESIGN
Adjustable Vertical Bar Close Mesh

Nos. 84 and 85 Designs—Adjustable deflection grilles with bars ⅜ in. in depth and spaced ⅜ in. apart—bars are connected in 2 in. sections, allowing sectional adjustment. The unique construction assures positive directional control of air flow with quick adjustability to any combination of deflections desired. Depth of bars and close spacing combine attractive appearance and concealment of the duct. Price group D.

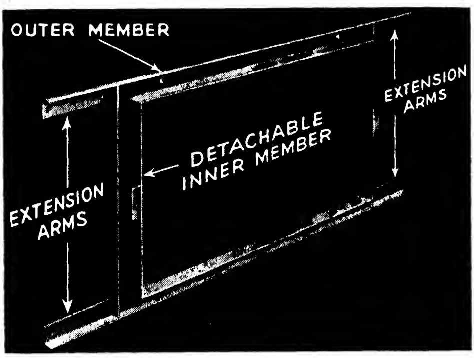


No. 85 DESIGN
Adjustable Horizontal Bar Close Mesh

Nos. 77 and 78 Designs—Fixed air flow. Similar in appearance to Nos. 84 and 85 Designs. Price Group B.

REGISTER AND GRILLE FRAMES

Any of the grille designs shown are available with any of the installation frames listed below.

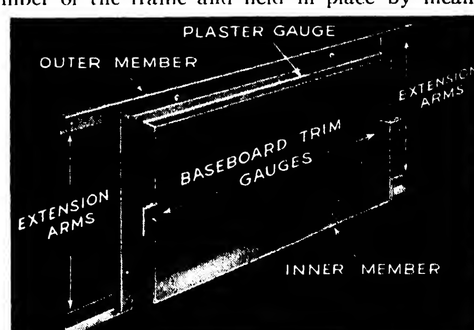


No. 3 SIDEWALL STUD FRAME (Left)

Installations when made as recommended with this frame are permanently streakproof. The frame provides a positive plaster lock, preventing the plaster from pulling away from the frame. The register face overlaps the frame ¼ in. on all sizes. The *Sponge Rubber Gasket* which is furnished attached to the register face, forms a seal between the wall and the register face, preventing air leakage at this point. The frame has rigid supporting arms of sufficient length on all sizes to fasten directly to the studs. The stackhead is bent over the outer member of the frame and held in place by means of the inner member.

No. 8 BASEBOARD STUD FRAME (Right)

Extension arms fasten directly to studs. Stackhead is attached to frame either by forming over inner member or by clamping it between the outer and inner members. May be used with any H & C one-piece Baseboard Register, all of which are furnished with Sponge Rubber Gasket as standard to prevent streaking. Frame insures a solid foundation for register and protects stackhead. Inexpensive, easy to install.



Schedule of Class Numbers for Use in Specifying H&C Registers, Grilles, and Intakes

Type of Frame	Design No.						
	68	72	77	78	84	85	90
Grille only, or Return Air Intake, Flat	680	720	770	780	840	850	900
Register without installation frame	681	721	771	781	841	851	901
Sidewall Register with No. 2 Band Iron Frame	682	722	772	782	842	852	902
Sidewall Register with No. 3 Stud Frame	683	723	773	783	843	853	903
Baseboard Register with Integral Frame	684	724	774	784	844	854	904
Baseboard Register with No. 5 Stack Frame	685	725	775	785	845	855	905
Intake with ⅞-in. projection	687	727	777	787	847	857	907
Baseboard Register with No. 8 Stud Frame	688	728	778	788	848	858	908

Complete Separate Catalogs on Air Conditioning Registers and Grilles or Warm Air Registers Available on Request.

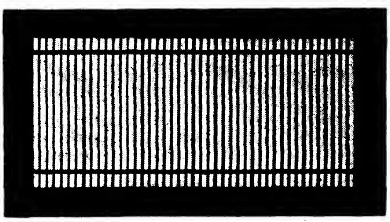
Tuttle & Bailey, Inc.

New Britain, Conn.

Branch Offices: Boston, New York, Chicago, Philadelphia

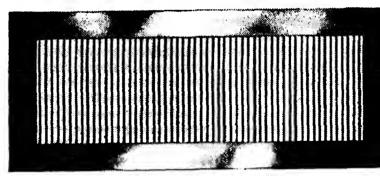
Air Conditioning
Grilles, Registers and Intakes
Air Control Devices

Ornamental Grilles
Cast or Wrought Metals
Convection Heaters



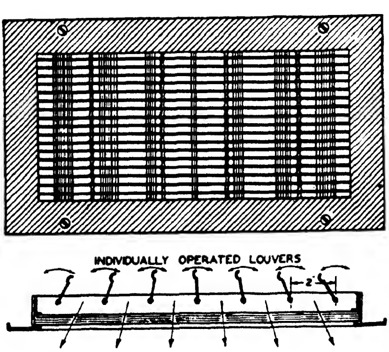
THE AIRLINE GRILLE

Neat and attractive. Provides fixed air deflection. Very moderately priced. Large effective area. Vertical or horizontal bars.



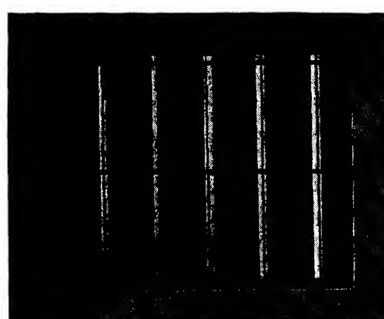
THE FLEXAIR GRILLE

Solid bar construction. Strong and sturdy. Provides adjustable air deflection. Easily painted to match trim. Vertical or horizontal bars.



COMBINATION VERTICAL AND HORIZONTAL DEFLECTION

A series of individually operated louvers is placed directly behind the face of the grille. Deflector louvers run in a direction opposite to that of the fan bars.



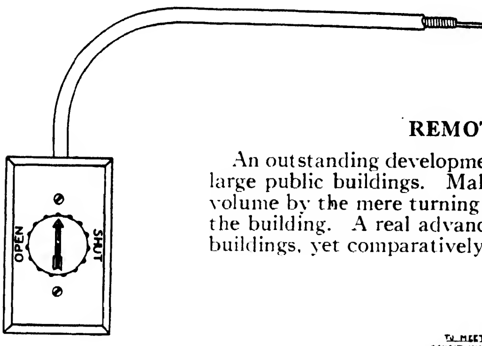
McKNIGHT REGISTERS

A scientifically designed register for commercial air conditioning work. Provides positive control of air volume at the outlet. Volume control louvers are operated by means of a special key furnished with each register.

For complete information on Tuttle & Bailey's entire line of air conditioning products, write for copy of latest Catalog No. 39.

Tuttle & Bailey, Inc.

New Britain, Conn.

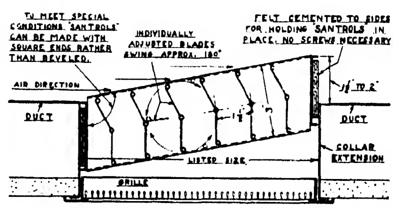


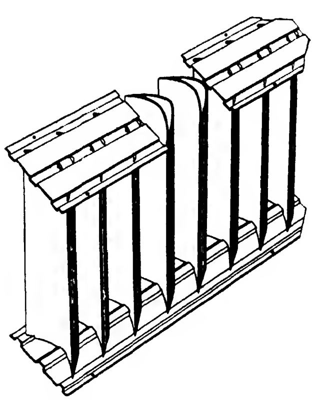
REMOTE CONTROL

An outstanding development. Ideal for hotels, office buildings, large public buildings. Makes possible individual control of air volume by the mere turning of a knob in every room throughout the building. A real advance in air conditioning for commercial buildings, yet comparatively inexpensive to install.

SANTROLS

A device which provides positive control of air volume throughout an entire duct system, and insures even distribution of air over each outlet face. An inexpensive unit that accomplishes much.





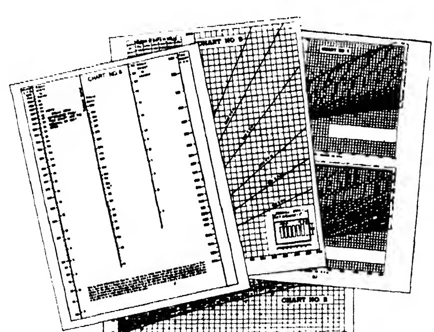
DUCTURNS

Ducturns are composed of scientifically designed turning blades. When installed in a duct system they eliminate the necessity for long radius turns and allow the use of right angle elbows throughout. Ducturns greatly simplify the layout of duct work and furnish a much more attractive and finished installation.

SIMPLIFIED SELECTION SYSTEM AND ENGINEERING DATA

A new and authentic, yet simplified, system for selecting proper sizes of grilles is now ready for distribution. The information given is the result of actual laboratory tests, made under conditions closely resembling actual job installations, and compiled only after many months of experimenting.

The data are distributed to Engineers without charge as an assistance to them in determining proper sizes and constructions of grilles to meet specified conditions.



The Independent Register Co.

ESTABLISHED 1898

3747 East 93rd Street, **Cleveland, Ohio**

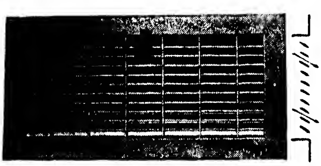
INDEPENDENT "Fabrikated"

Reg. U. S. Pat. Office

AIR CONDITIONING REGISTERS AND GRILLES

No. 311-A—ADJUSTABLE DIRECTED AIR FLOW

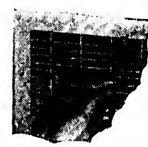
With Horizontal Grille Bars



No. 311-A—Air Flow Downward Adjustable from straight to 45 deg.

With "INDEPENDENT" Adjustable Directed Air Flow Registers and Grilles the Engineer is in complete control of the direction of air flow.

The directional adjustment can be made at the time of installation—or after the system is operating, to meet unforseen or changed conditions.



Patented

The method of adjustment is simple as shown, and many directions and combinations can be developed to suit the need.

Each interior grille bar is adjusted individually.

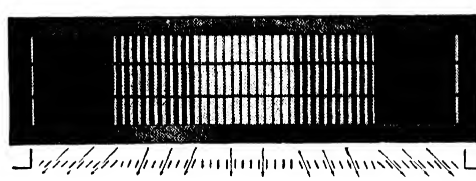
No. 321-A—ADJUSTABLE DIRECTED AIR FLOW

With Vertical Grille Bars

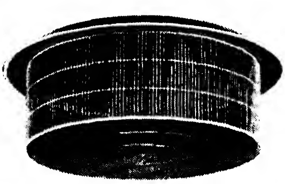
Patented

Standard registers are furnished with single valves;—they are also made with "Multiple Valves" operating in unison—also with rear "Deflecting Vanes" which can be adjusted individually—with which the Engineer is given a dual control of the air flow—being able to secure right and left together with up and down deflection at the same time.

Grille Bars set for straight, right and left deflection. The grille bars are set in a firm tension, yet easily adjusted, with the tool sent with each order.

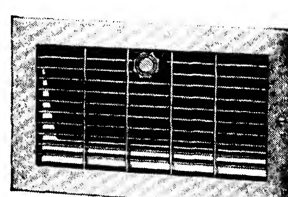


No. 321-A—Showing a combination of Adjustments



Ceiling Outlet No. 1360-R

Ceiling Outlets are furnished in several sizes—both Round and Square styles.



No. 311-A—Knob Control

Ceiling Outlets offer one of the most flexible units for ventilation and distribution of cooled or heated air. The ceiling outlet grilles are of perforated metal.

The Nos. 311-A or 321-A Registers can be furnished with either Lever, Knob, Key, Chain or Pull Rod Control.

You should have the Independent Register Catalogues—Yours for the Asking.

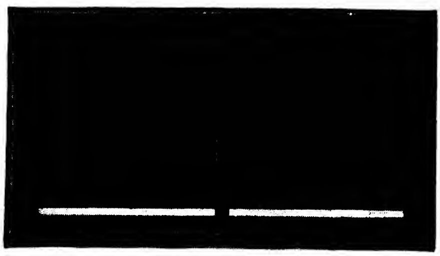
United States Register Company

General Offices: **Battle Creek, Mich., U.S.A.**

Branches: Minneapolis, Minn., Kansas City, Mo., Albany, N. Y., New York, N. Y., San Francisco, Calif.

Air Conditioning Registers, Vents and Grilles

Style 145

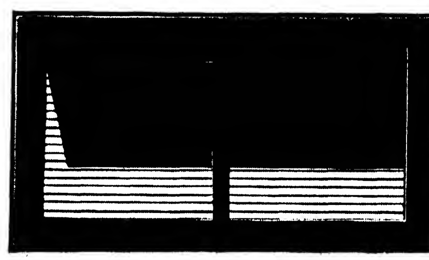


Style 145—U. S. Adjustable Bar Air Conditioning Registers specially designed for Summer Cooling and Cold-weather Warming. Bars adjust from 60 deg Up-Flow to 60 deg Down-Flow by a lever operating Pin that is removable after Adjustment. Does not MAR or INJURE Register finish.

Can be furnished with Box Band Frames or Studding Frames, and INSET PANELS.

Also furnished in Vertical Adjustable Bar Styles for Right or Left Adjusted Setting of Diffusions.

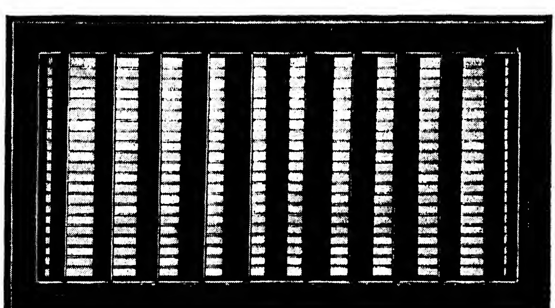
Style 153



Style 153—New "Louvre-Type" Style of Air Conditioning Register with Narrow Bar Spacing conducive to non-vision through Grille. Bars are ¼ in. deep. Can be furnished in any Directional Flow, Horizontal or Vertical Styles. For complete Reference Refer to 1939 Edition General Catalog No. 27.

Also furnished with all styles of setting frames, and with INSET PANELS.

Style 177 V. V. I. or 145 V. V. I.



Letters *V. V. I.* used as suffix to an Air Conditioning Vent or Grille Style Number means this Grille is equipped with Vertical Deflector Back Blades that can be set individually in any desired Angle and Locked in that fixed position with Set Screw at end of each Blade.

This setting may be changed by adjusting Deflector Back Blades after Grille is installed.

Horizontal Back Blades are designated by Letters *H. V. I.*

Letters *V. V. L.* used as suffix to an Air Conditioning Register Style Number means that this Register is equipped with Vertical Group Operated Valves that swing to complete Closing Position and to any desired practical angle of Deflection.

Can be furnished with Knob-Operators, Chain-Operators with Pulley, also Pole Operating.

The Style Number indicates the Style of Register Face and the suffixing Letters, for Example, *V. V. L.* means vertical Lever Operated Valves.

Letters *H. V. L.* means Horizontal Valves.

Further Data may be had in Latest General Catalog.

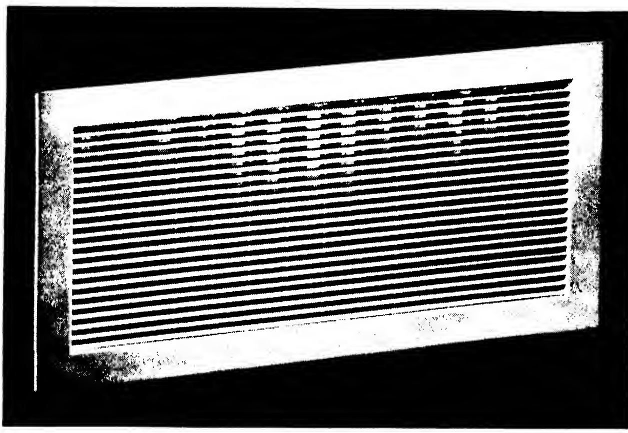
Waterloo Register Company

Waterloo, Iowa **Seattle, Wash.**

Established 1902

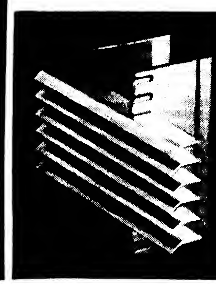
Representatives in Principal Cities

AIR MASTER GRILLES



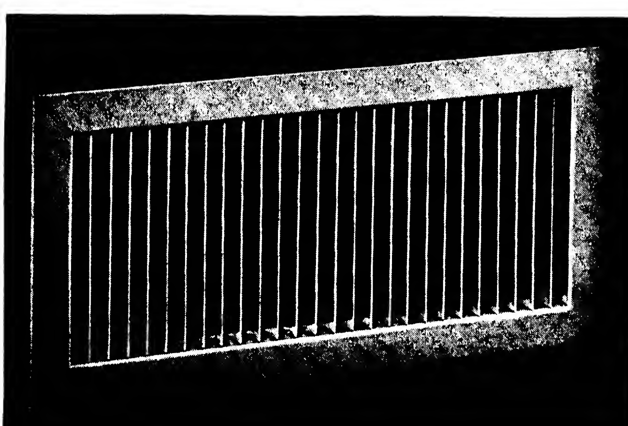
1-A Supply Grille

Fixed fine mesh diffusion plus individually adjustable rear deflectors.



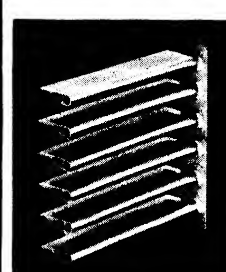
Supply grilles are matched with return grilles for both styles 1-A and 2-A. All supply designs feature maximum adjustability and flexibility coupled with efficiency and rugged strength.

Many other grilles and air control devices for air conditioning applications are illustrated in Waterloo Catalog No. 18. Engineering and Performance data are published in a separate bulletin. Both are available from main office or representatives.



2-A Supply Grille

Individually adjustable streamline bars.



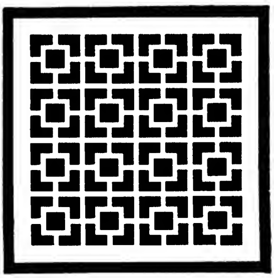
Wickwire Spencer Steel Company

41 E. 42nd St., New York, N. Y.

BUFFALO WORCESTER LOS ANGELES CHICAGO SAN FRANCISCO

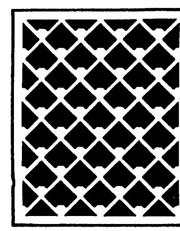
WISSCO STAMPED GRILLES
WICKWIRE SPENCER PERFORATED METALS

Wissco Grilles are manufactured in a great variety of designs—50 different styles from standard dyes, and many special designs and curved shapes to conform with customer specifications. Grille perforations are designed so that uninterrupted vertical and horizontal members give adequate rigidity and strength of grille structure with effective concealment and large free air opening. The new "Lacecane" design provides free air openings of 50 per cent of grille area—in other designs free air openings range upward to 70 per cent of grille area.



Design 820

In Wickwire Spencer plants, modern machinery produces grilles as heavy as $\frac{5}{16}$ in. in thickness, and in any size or shape up to 60 in. x 156 in., in one piece. Larger sizes in two or more pieces suitable joined so that the joint is virtually invisible on the front of the grille.



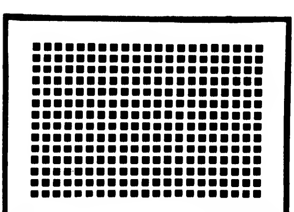
Design 850

MATERIALS

"Wissco" Bronze, an alloy sheet metal of high tensile strength, is specially recommended for Wissco Grilles. It compares favorably in cost with electroplated steel, but for durability equals commercial or naval bronzes. Finished in any manner to harmonize with surroundings. Wissco Grilles are also furnished in stainless and regular steels, brass, copper, bronze, nickel, monel, zinc or aluminum.

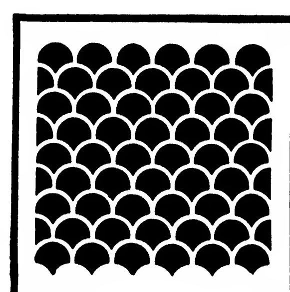
Sheet Metals, because of their strength, permit use of thinner lighter materials – Wissco grilles are perforated in sheet steels ranging from $\frac{1}{4}$ in. thick to 16 U. S. gauge, and in bronzes and aluminum $\frac{5}{16}$ in. to 16 B. & S. gauge.

FINISH



Plain Lattice

In the Wickwire Spencer finishing plant any type of finish may be supplied — electroplated, japanned, buffed, or painted; special finishes matched or supplied as desired. When a paint finish is desired it is recommended that grilles be shipped from the manufactory with shop coat, only finishing coat to be applied where installed.



Grille Design—117

SPECIAL FEATURES

Invisible doors, hinged grilles, angle frames, or other special features supplied as required. Other sheet metal designs and specialties produced as desired.

A copy of the New Catalog on Wissco Grilles, giving detailed analysis of Grille layouts and specifications, will be sent upon receipt of your written request.

The American Rolling Mill Company

Executive Offices, **Middletown, Ohio**

Atlanta, Ga,	1437 Citizens and Southern National Bank Bldg	Kansas City, Mo	7100 Roberts St
Boston, Mass	201 Devonshire St	Middletown, Ohio	703 Curtis St
Buffalo, N Y	504 Seventeen Court St Bldg	Minneapolis, Minn	171-27th Ave, S E
Chicago, Ill	310 S Michigan Bldg	New Orleans, La	3501 S Carrollton Ave
Cleveland, Ohio	1516 B F Keith Bldg	New York, N Y	50 Church St
Dallas, Texas	1111 Santa Fe Bldg	Philadelphia, Pa	1808 Lincoln-Liberty Bldg
Detroit, Mich	5-261 General Motors Bldg	Pittsburgh, Pa	1632 Oliver Bldg
Indianapolis, Ind	Circle Tower	San Francisco, Calif	465 Tenth St
		St Louis, Mo	1725 Ambassador Bldg

Choose the Correct Armco Grade

These grades of Armco sheet metal are recommended for the air conditioning applications shown For detailed information get in touch with the nearest district office or write direct to The American Rolling Mill Company, Middletown, Ohio

Armco Ingot Iron

(Galvanized)

Ducts
Washer Chambers
Plenum Chambers
Steam Line Casings
Furnace Casings
Spray Towers
Drip Pans
Housings
Machine Guards
Unit Conditioners (Industrial)
Roof Ventilators
Eliminator Blades

Armco Paintgrip

(Galvanized)

Recommended for all applications listed above that need immediate painting Send for complete information.

Hot Rolled

(Sheets and Strip)

Fan Blades
Blower Casings
Fuel Oil Tanks
Unit Conditioners
Stoker Hoppers

Armco Zincgrip

A special galvanized sheet that can be severely formed without peeling or flaking of the zinc coating.

Cold Rolled

(Sheets and Strip)

Furnace Casings
Room Unit Casings

Plates

(Armco Ingot Iron)

Smoke Stacks
Coal Hoppers
Breeching
Unfired Pressure Vessels
Low-fired Boilers
Tanks

Armco H. T. -50

A low alloy, high tensile steel possessing great strength. Used with proper design it results in weight reduction of framework, tanks and similar items Under atmospheric service conditions it has four to six times the resistance of regular steel

Stainless Steel

(Sheets, Strip and Plate)

Combustion Chambers
Heat Flues and Tubes
Furnace Casing Trim
Grilles
Corrosion Resistance
Fan and Blower Blades
Heat Resistance without destructive scaling up to 1600° F. or higher

Other Armco Products

The grades for these applications are only a few that Armco makes Others include copper-bearing sheets and plates and open-hearth steel, either galvanized or uncoated.

Bethlehem Steel Company

General Offices: **Bethlehem, Pa.**

BETHLEHEM STEEL COMPANY, GENERAL OFFICES: BETHLEHEM, PA. DISTRICT OFFICES: ALBANY, ATLANTA, BALTIMORE, BOSTON, BUFFALO, CHICAGO, CINCINNATI, CLEVELAND, COLUMBUS, DALLAS, DETROIT, HONOLULU, HOUSTON, INDIANAPOLIS, JOHNSTOWN, PA., KANSAS CITY, MO., LOS ANGELES, LOUISVILLE, KY., MILWAUKEE, NASHVILLE, NEW HAVEN, NEW YORK, PHILADELPHIA, PITTSBURGH, PORTLAND, ORE., ST. LOUIS, ST. PAUL, SALT LAKE CITY, SAN ANTONIO, SAN FRANCISCO, SAVANNAH, SEATTLE, SYRACUSE, TOLEDO, TULSA, WASHINGTON, WILKES-BARRE, YORK. EXPORT DISTRIBUTOR: BETHLEHEM STEEL EXPORT CORPORATION, NEW YORK.

SOME FACTS ON CORROSION-RESISTANCE OF STEEL and advantages of copper-bearing Beth-Cu-Loy

Beginning some twenty years ago, the *American Society for Testing Materials* started a number of tests to determine the corrosion-resistance in atmosphere of various irons and steels. The charts at the right summarize the results to date; compare the life, in atmosphere, of the four most generally used materials. These tests are worth remembering when buying or specifying iron or steel—they show one material, copper-bearing steel, to be definitely superior.

Beth-Cu-Loy, Bethlehem's copper-bearing steel, is of the identical composition as that shown by the lower bar of each chart. It is available in the form of sheets, pipe, and plates. It costs only 3 to 5 per cent more than ordinary steel- the tests show that it has from 2 to 2½ times the resistance to rust. It costs considerably less than open-hearth or copper-bearing iron.

BETHLEHEM MAKES:

Sheet Steel—all types, hot-rolled (black), cold-rolled, and galvanized—available in Beth-Cu-Loy.

Steel Pipe—all sizes and weights, butt-welded and lap-welded—available in Beth-Cu-Loy.

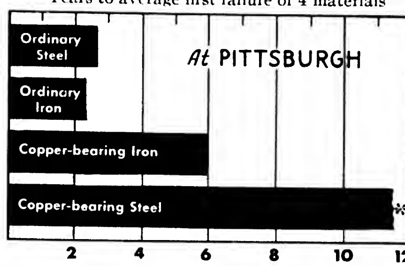
Ammonoduct—A new kind of pipe that has an outstanding advantage in its unusual ductility. It can be bent cold, without need for annealing, without danger of fracturing. Recommended for ammonia piping and for heater coils, water legs in furnaces and similar uses where pipe must be bent. Available in Beth-Cu-Loy.

Boiler Tubes—Charcoal iron and steel.

Plates—all sizes; flanged and dished heads. Available in Beth-Cu-Loy.

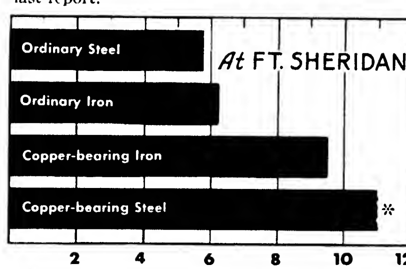
Literature and further information on any of these products can be secured from the nearest district office or from the general office in Bethlehem, Pa.

A.S.T.M. Test of 22-gage sheets
Years to average first failure of 4 materials



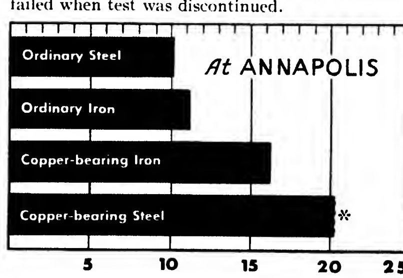
Sheets exposed April 21, 1926; tests still under way (see Proceedings of *A.S.T.M.*—Committee A-5, Vol. 38).

*No failures in copper-bearing steel sheets at last report.



Sheets exposed April 19, 1917; test discontinued April 16, 1928 (see Proceedings of *A.S.T.M.*—Committee A-5, Vol. 28).

*Only 10 of 61 copper-bearing steel sheets had failed when test was discontinued.



Sheets exposed October 17, 1916; tests still under way (see Proceedings of *A.S.T.M.*—Committee A-5, Vol. 38).

*Only 9 of 78 copper-bearing steel sheets had failed at last report.

A new booklet, "Beth-Cu-Loy Sheets," gives the story of these tests. A copy is yours for the asking.

Carnegie-Illinois Steel Corporation

General Offices: **Pittsburgh and Chicago**

District Offices

BIRMINGHAM	CLEVELAND	INDIANAPOLIS	PITTSBURGH
BOSTON	DENVER	MILWAUKEE	ST. LOUIS
CHICAGO	DETROIT	NEW YORK	ST PAUL
CINCINNATI	HOUSTON	PHILADELPHIA	WASHINGTON

COLUMBIA STEEL COMPANY, SAN FRANCISCO, Pacific Coast Distributors

UNITED STATES STEEL PRODUCTS COMPANY, NEW YORK, Export Distributors

RUST-RESISTING U·S·S COPPER STEEL GALVANIZED SHEETS

Copper steel is an alloy made by adding copper to molten steel, thereby increasing the resistance of steel to rust Metallurgists, railroad construction engineers and independent research laboratories have tested U S S Copper Steel and discovered that it lasts two to three times as long as plain steel when subjected to atmospheric corrosion

When you specify U·S S Copper Steel Sheets for air-conditioning ducts or equipment, you get the advantage of 100 per cent to 200 per cent longer life for about 5 per cent more cost Duct work is so costly to replace that Copper Steel should be considered for any worthwhile job.

U·S S Copper Steel Sheets are saving thousands of dollars yearly in the ceaseless fight against rust They give maximum protection per dollar of cost and are being used in ever-increasing volume by heating, ventilating and air-conditioning engineers, architects and contractors

Gauges of Steel Sheets Used for Duct Construction

HEATING AND VENTILATING				Planing Mill and Other Exhaust Systems	
Round Ducts		Rectangular Ducts			
Diam , Inches	Gauge	Width, Inches	Gauge	Diam , Inches	Gauge
6 to 19	26	4 to 18	26	Up to 8	24
20 to 29	24	19 to 30	24	9 to 14	22
30 to 39	22	31 to 60	22	15 to 20	20
40 to 49	20	61 to 118	20	21 to 30	18
50 and above	18	118 and above	18		

In the above table rectangular ducts are to have cross breaks for the gauges shown, otherwise two gauges heavier should be used. One inch standing seams should be used on widths up to 48 in , 1½ in. seams on widths over 48 in., and for widths over 60 in , the seams should in addition be provided with reinforcing bars or angles.

(This material is reprinted by permission from "Fan Engineering," Buffalo Forge Co).

U·S·S Black and Galvanized Sheets

Two principal types of black sheets are used by air-conditioning engineers. They are U·S S Hot Rolled and U·S·S Hot Rolled Annealed. These steel sheets may be had in a number of different finishes, suitable for all sorts of forming operations and for painting.

Remember that every kind of U S S Black Sheet is available in Copper-Steel Black Sheets for blowers, refrigerator cabinets, dust collectors, tube collectors, fans, ducts and a thousand other specialty products.

Five types of galvanized sheets made by U·S·S Subsidiaries are used extensively in heating and ventilating work. The best types for your particular products or installations may be ascertained by consulting U S·S engineers Write the nearest branch office.

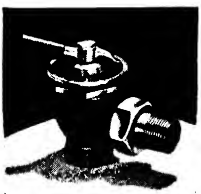
Barnes & Jones
INCORPORATED

Boston, Mass.

New York Office: 101 Park Avenue

Barnes and Jones Vapor and Vacuum Systems of Steam Heating; Modulation Valves, Packless Quick Opening Supply Valves; Metering Orifice Supply Valves; Thermostatic Radiator Traps and Cage Units, that provide instant trap repair; Thermostatic Traps for medium and high pressures; Condensators (Boiler Return Traps); Float and Thermostatic Traps; Strainers, Damper Regulators; Gages; Proportionator Systems with Zone Control.

Modulation Valves Type K



With non-tarnishable indicating dial, non-rising stem, renewable disc seat. Tail piece extra-heavy to prevent breakage—extra-long to facilitate connection to radiator. Three models: lever handle, wheel handle and lock shield.

Size	½ In.	¾ In.	1 In.	1¼ In.
Cap. Sq Ft Rad.	30	60	100	180

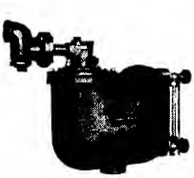
(8 oz pressure).

Packless Quick Opening Valve Type F

Non-rising stem and renewable disc seat. Large unobstructed passages to prevent trouble from dirt or scale. Furnished with wheel handle only.

Condensators

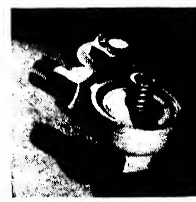
For returning water of condensation to boiler from open return line systems independently of boiler pressure, without change in operating conditions, air binding, or admitting steam to return side. Simple and rugged in construction; positive in operation. All working parts are of best bronze metal.



No. 32 Condensator

No.	Capacity in Sq Ft
31	700
32	1,600
33	3,500
34	6,000
35	10,000
36	16,000
37	32,000

Thermostatic Radiator Traps



The Cage Unit, complete operating unit in itself, carries its own seat of special alloy. Calibrated under actual working pressure at the factory and permanently locked in adjustment. Unit easily and quickly replaced without special tools; lift out old unit and insert a new one. Available in sizes to fit almost any make of trap.

Symbol	120	12	124	134	13	14
Inlet Tapping	½"	½"	½"	¾"	¾"	1"
Outlet Tapping	½"	¾"	¾"	¾"	¾"	1"
Capacity, Sq Ft C. I. Rad.	200	200	400	400	700	1200

Capacities based on 1½ lb pressure differential.

Float and Thermostatic Traps

Drip Type

For use on Unit Heaters, also drips from supply mains and risers and on returns from water heaters and indirect stacks. Float controlled valve governs discharge of water; thermostatically controlled valve allows passage of all of air but prevents passage of steam. Made with ¾ in., 1 in., and 1¼ in., tappings. Capacities 200 lb to 1200 lb of water per hour at 2 lb pressure differential.

Heavy Duty Type

Combination float and thermostatic traps with air and water capacity large enough to take care of the load from the largest vent stacks, dry kiln coils, hot water heaters and other units condensing large quantities of steam at low pressures. Made in 1½ in. and 2 in. sizes. Capacities to 5,000 lb of water per hour. 2 lb pressure differential.

Armstrong Machine Works

851 Maple Street **Three Rivers, Mich.**

Exclusive Manufacturers of Armstrong Inverted-Bucket Steam Traps

ARMSTRONG REPRESENTATIVES

Atlanta, Ga., J M Tull Metal & Supply Co, Inc, 285 Marietta St, N W.

Baltimore, Md., Milby & McKinney, 116 Light St

Birmingham, Ala., Southeastern Products Co, 1401 Lomb Ave

Boston, Mass., Files Steam Specialty Co, 261 Franklin St.

Buffalo, N. Y., Herr Steam Specialty Co, 360 Warwick Ave

Charleston, W. Va., Baldwin Supply Co, 518 Capitol St

Chicago, Ill., Barrett-Christie Co, 108-112 N Clinton St

Dallas, Texas, Geo B Allan & Co, North Texas Bldg.

Denver, Colo., Hendrie-Bolthoff Mfg & Supply Co, 1637-17th St

Des Moines, Iowa, E B Carr, 307 Securities Bldg

Detroit, Mich., A F Squier, 2910 National Ave

Duluth, Minn., John E Smith, 1720 W Superior St

Erie, Pa., Coblentz Equipment Co, 1119 Peach St

Evansville, Ind., Evansville Supply Co, Esco Bldg

Fond du Lac, Wis., A N Goff, 94 Eighth St

Honolulu, T. H., The von Hamm-Young Co, Ltd

Indianapolis, Ind., Indianapolis Belting & Supply Co, 34 S Capitol Ave

Kansas City, Mo., Hughes Machinery Co, 342 Mfrs Exchange Bldg

Knoxville, Tenn., Leinart Engineering Co, 311 W Cumberland Ave

Los Angeles, Calif., Guy L Warden, 114 West 17th St

Louisville, Ky, Graft-Pelle Co, 309 W Main St

Memphis, Tenn., The Power Equipment Co, 1352 Madison Ave

Milwaukee, Wis., Hamacher & Williams, 2540 W. Wells St

Minneapolis, Minn., Albert C Price Co, 257 Fourth Ave, South

Montreal, Quebec, Preston, Phipps Inc, 955 St James St W

New Orleans, La., Louisiana Steam Equipment Co, 109 Tchoupitoulas St

New York, N. Y., Advance Engineering Co, 69 Dey St

Philadelphia, Pa., Brogan & Co, 810 Race St

Phoenix, Ariz., John W Ladlow, Box 1784

Pittsburgh, Pa., R S Eastman Co, 222 First Ave

Portland, Ore., Heating and Ventilating Equipment Co, 927 S W Oak St

Richmond, Va., A T Shepherd, 411-12 Tenth St Bldg

St. Louis, Mo., O'Brien Equipment Co, 2726 Locust Blvd

Salt Lake City, Utah, Lee, Pace & Turpin, 144 S Fifth West St

San Francisco, Calif., Refrigerating & Power Specialties Co, 380 Brannon St

Seattle, Wash., Heating & Ventilating Equipment, Inc, 500 First Ave, South

South Bend, Ind., Smith-Monroe Co, 1912 S Main St

Syracuse, N. Y., The Hopton Co, 321 Denison Bldg.

Tampa, Fla, G W Neale, 404 Morgan St

Toronto, Ont., Arthur S Leitch Co, Ltd, 1123 Bay St

Vancouver, B. C, General Equipment, Ltd 319 W Pender St

Winnipeg, Man., Kipp-Kelly, Ltd, 68 Higgins Ave

Wooster, Ohio, Steam Economies Co, 1011 Beall Ave

Armstrong offers two types of traps for heating, air conditioning, and steam distribution service

Standard Inverted Bucket Traps, the type originated by Armstrong, are *non-airbinding* and *self-scrubbing* They are used for low, medium, and high pressure duty where relatively little air must be handled along with the condensate Their free-floating lever design makes it possible to open very large discharge orifices compared with the size and weight of the trap itself

Armstrong Blast Traps are used where large amounts of air must be vented quickly when steam is first turned on The Armstrong Blast Trap (illustrated on next page) consists of a standard inverted bucket trap with a large auxiliary air vent controlled by a piece of rustless thermostatic bi-metal. When the trap is cold, the auxiliary vent is wide open, allowing air and condensate to blow right through the trap When steam reaches the trap, the auxiliary vent closes, allowing the bucket to float and close the main valve after the condensate has been discharged

The Armstrong Blast Trap has several advantages over the conventional float and thermostatic trap

1 The Armstrong Blast Trap has but a *single orifice* to be maintained tight against the full pressure differential. The pressure differential across the auxiliary air vent amounts to only 1 in to 4 in of water

2. *Positive action* The discharge valve in an Armstrong Blast Trap is either wide

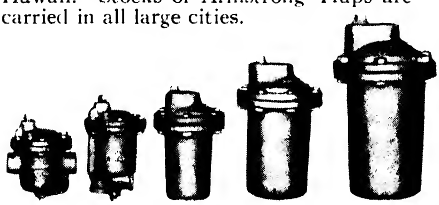
open or tight shut. Fast opening and fast closing prevent wire-drawing.

3. *Handles dirt.* There are no dead spots in an Armstrong Trap in which dirt can settle and interfere with the operation of the trap. No blow-down valves are necessary and ordinarily no strainers are required ahead of Armstrong Traps.

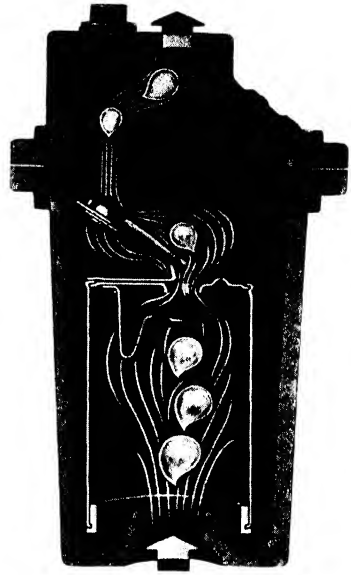
4. The wearing parts in all Armstrong Traps are identical in design, material, and precision workmanship with parts used in Armstrong Forged Steel Traps for pressures up to 1500 lb gage and total temperatures of 850 F. At ordinary heating pressures, Armstrong Traps last many years without any maintenance expense whatever.

Armstrong Steam Trap Book. This 32 page book gives complete information on all sizes and types of Armstrong Traps. It also contains 15 pages of data on the subject of trap selection, installation, and maintenance. A free copy will be mailed no request.

Service Organization. Satisfactory selection and operation of all Armstrong Traps is assured by 43 district representatives in the United States, Canada, and Hawaii. Stocks of Armstrong Traps are carried in all large cities.



200 201 211 212 213



THE ARMSTRONG BLAST TRAP

The above picture shows how the Armstrong Blast Trap works when steam is first turned on. Note that the auxiliary vent in the top of the bucket is wide open. Air and water are free to rush straight through until steam comes in and causes the bi-metal strip to bend upward closing the auxiliary vent with the small flat disc. Thereafter it works the same as the standard trap.

Sizes, Capacities and List Prices of Armstrong Traps

Trap Size	No. 200 and 201	No. 211	No. 212	No. 213	No. 214	No. 215	No. 216
Pipe Connections	½″*	½″	½″ or ¾″	½″ or ¾″	1″	1″ or 1¼″	1½″ or 2″
List Price (Regular)	$7.00	$9.25	$15.00	$20.75	$29.00	$38.00	$55.00
List Price (Blast Trap)	$8.50	$10.75	$17.00	$22.75	$31.50	$40.50	$60.00
Telegraph Code (Regular)	Acacia / Acanthus	Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)	Acacette / Acanette	Aspette	Birette	Walette	Hemlette	Larette	Tamrette
Height	4 11/16″	6⅜″	8″	10¼″	12½″	14″	16¾″
Diameter	4¼″	4⅛″	5″	6⅜″	7½″	8½″	10 9/16″
Weight	4 Lb	5½ Lb	10½ Lb	19 Lb	32 Lb	47 Lb	76 Lb
Maximum Pressure	125	200	200	250	250	250	250
Lb. pressure 5	450	840	1560	3000	4600	7600	14,300
10	560	1000	1900	3500	5600	9100	17,200
15	640	1080	2060	3900	6300	10,000	19,000
20	690	890	1800	3100	5900	8500	18,200
30	500	970	2050	3600	6600	9800	17,600
50	600	840	1840	3750	6200	8900	18,200
70	660	940	2030	3700	6100	9200	18,500
100	650	880	1840	3400	6300	10,400	18,500
125	660	960	2040	3880	6700	10,900	20,400
150	...	820	1530	3500	5900	9500	18,600
200	...	900	1680	3200	5400	9500	17,400
250	...	...	....	3500	5700	10,200	19,000

Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information, see the Capacity Chart in the Armstrong Steam Trap Book.

*If ¾ in. connections are desired, order No. 202 for straight way or No. 203 for angle.

The Beaton & Cadwell Mfg. Company

Main Office and Factory: **New Britain, Conn.**

CADWELL No. 45 UNIT FOR HOT WATER HEATING SYSTEMS

A complete unit—nothing else to buy.

A departure from the conventional type of equipment as used with tank in basement systems

To keep the expansion pressure in the system within practical working limits even under sudden firing methods, as in oil burners

Providing means for elastic pressure distribution within the system.

Keeping the system filled to any desired pressure, against increase of pressure above relief setting

All of this is achieved in a novel manner

The combined pressure governing and relief feature—which is new·

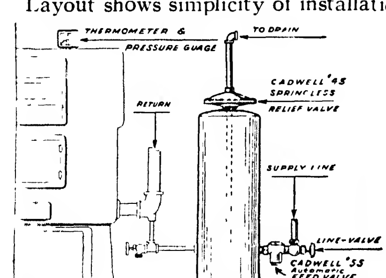
1. Has no springs
2. Is governed entirely by physical laws.
3. Cannot increase setting at any time
4. It tests itself automatically as long as there is any water in system
5. It is responsive to the slightest difference of pressure within the system
6. Absolutely guaranteed to protect the boiler, against increase of pressure above relief setting.

The filling arrangement is automatic and can be varied in pressure from 5 to 20 lb by adjusting screw

A large strainer prevents foreign matter from entering system The strainer, valve disc and seat can be cleaned at any time without losing water out of system

Valve and tank supplied in Maroon color May be used in connection with any circulator system

Layout shows simplicity of installation



CADWELL PRESSURE AND TEMPERATURE RELIEF VALVE—Self Closing

Cadwell No. 25 Pressure and Temperature Relief valve Opening and closing governed entirely by internal pressure—not by heavy springs—no sticking valve seats, only slight drop in pressure required to close valve, leak proof. Set to maximum temperature of 210 F, or on special order, supplied with temperature relief manually adjustable to lower temperatures.

Cadwell No 25 Relief Valve

Valve readily tested by pressing cap on top of valve—same operation also clears valve seat of sediment accumulation Construction of valve with split base allows valve seat to be inspected, cleaned or renewed without changing setting

Cadwell No. 35 Pressure and Vacuum Relief Valve is similar to valve No 25 except it does not have the temperature relief feature Standard No. 35 valve is set and sealed to 150 lb maximum water pressure—other pressures if required If desired, this valve can be supplied with pressure relief manually adjustable to lower pressures Valve tested for pressure and seat cleared of sediment by pressing cap on top of valve

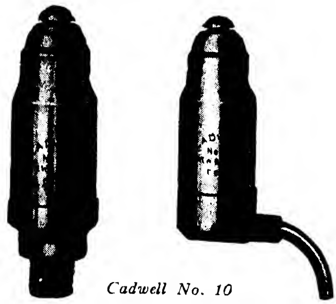
Cadwell No 35 Pressure and Vacuum Relief Valve

Cadwell No. 35-F Pressure, Vacuum and Temperature Relief Valve is similar to No 35 except it has a fusible-plug temperature relief—not self-closing, fusible plug must be renewed after each temperature relief action

Cadwell relief and pressure valves 25, 35 and 35 F have A G A. approval

Complete catalog sent upon request.

CADWELL THERMOSTATIC AIR VALVES FOR ONE PIPE STEAM AND VACUUM SYSTEMS



Cadwell No. 10

Bottom Outlet 1/8 and 1/4-in. and 3/4 x 3/8-in. sizes *Angle Type*

Cadwell No. 10 positive action for radiators, non-vacuum, our best all metal syphon air valve now available in either the non-adjustable or adjustable type.

Cadwell 10 S.S. same as No. 10 except straight shank for venting return lines, etc.

Cadwell No. 50

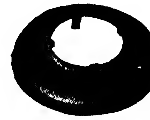
Bottom Outlet 1/4-in. and 3/4 x 3/8-in. sizes *Angle Type*

No. 50 Diaphragm operated vacuum air valve, closes efficiently against the escape of steam or vacuum. Large closing surface of diaphragm, through nickel silver pin forces check off seat with every close of main valve by steam or vacuum, positively preventing sticking of valve. Equalizes the system—port can be increased in size to take care of distant or sluggish radiators.

Nickel silver pins are used in all Cadwell air valves to prevent corrosion.

"PERFECTION" FLOOR AND CEILING PLATES

No. 1



No. 3A

No. 1—Sectional floor and ceiling plate. Cast iron or Brass—1 in. flange—sizes 1/8 in. to 12 in.

No. 2—Same as No. 1 with set screw instead of springs.

No. 3—Hinged floor and ceiling plate. Cast iron or Brass—1 in. flange—sizes 3/8 in. to 4 in.

No. 3A—Same as No. 3, with set screw—sizes 1/4 in. to 4 in.

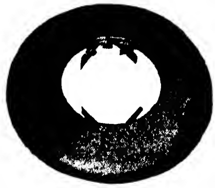
No. 3 S.—Solid floor and ceiling plate, cast iron or brass. 1 in. Flange with or without set screw—sizes 1/8 in. to 8 in.

No. 6—Sectional floor and ceiling plate cast iron or brass—1 in. flange—with set screw. 1 1/4 in. high—size 1/2 in. to 4 in.

No. 6A—Solid floor plate cast iron or brass—size 1/4 to 5 in.—with or without set screw.

No. 7—Same as No. 1—But with 1 1/2 in. flange. Size 1/2 in. to 4 in.

No. 9—Same as No. 1 but with 2 in. flange—size 1/2 in. to 4 in.



No. 10 Hinged

No. 6A

No. 10—The original No. 10—Perfection—1 in. flange—size 1/4 in. to 6 in. Can be furnished in grained oak finish.

No. 11—Same as No. 10 but with set screw.

Note:—No 10 type Copper Service Tube Plates—1/4 in. to 3 in. tube sizes.

All plates can be furnished in plain, nickel or chromium plated finishes, as specified.

Complete Catalog Upon Request

Cochrane Corporation

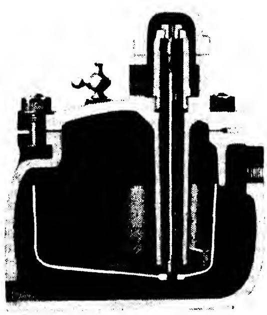
3130 North 17th Street, **Philadelphia, Pa.**

Branch Offices in 40 Principal Cities

COCHRANE HEAVY-DUTY STEAM TRAPS

A high pressure unit for condensate drainage of steam lines, separators, coils, evaporators, etc., and for conditions involving relatively high drainage rates. Recommended for pressures up to 400 lb. Simple construction. No levers, constricted passages or stuffing boxes to become clogged with sediment or scale. All parts are readily accessible. Action is quick and positive, avoiding wire drawing and erosion. Discharge capacities may be conveniently altered by easy change of valve seat.

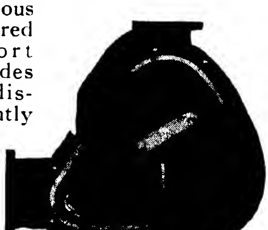
Write for publication No. 2850.



Bucket Trap

MULTIPORT DRAINERS

Of the multiport type, they afford unusual capacity for removing condensate or drips from purifiers, separators, jackets, radiators, pressure heating or drying coils, etc. Eliminating condensate delivers maximum heat from steam production at lower cost. Tremendous capacity assured by large port areas. Provides continuous discharge. Instantly responsive. Compact and light in weight. For pressures up to 150 lb.



Multiport Drainer

COCHRANE MULTIPORT RELIEF VALVES

For back pressure, atmospheric relief flow or check valve service on air, gas steam or water lines to give positive protection against explosions arising from stuck, jammed or overweighted valves. Differ in the usual construction in that a number of small disks are used instead of one large disk. For full description write for publication No. 2710.

Multiport Back Pressure Valve

ALL-SERVICE SEPARATORS

Steam is rarely or never generated dry or clean and, in that state, corrodes turbine blades, engine and pump cylinders, valves, pistons, etc. Exhaust steam contains oil and entrained solids which should be removed to eliminate friction and wear. Cochrane Separators purify steam by separating out oil, slugs of water and condensate. Complete removal of entrainment is accomplished by vertical baffle ribs which guide it into a direct unrestricted fall, and a baffle area which extends far beyond the flow from the inlet pipe. Ports at the sides of the baffle prevent the purified steam from passing over the drip area and coming into contact with the entrainment. The steam flow is uninterrupted and pressure loss is minimized.

All Service Separator

For information on other Steam Specialties write for individual publications.

GRINNELL COMPANY INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence, R. I.**

National Distributors of Thermoflex Traps and Heating Specialties

For data on other Grinnell Products, see pages 1000-1002

Thermoflex Specialties

The heart of all Thermoflex Traps is the Hydron Bellows.

The Hydron Bellows is formed under hydraulic pressure. This powerful internal pressure locates any weakness of any nature in the tubing. Such hydraulic pressure is many times more severe than any pressure the Trap will ever be called upon to control. Every Thermoflex Trap, therefore, is practically indestructible.

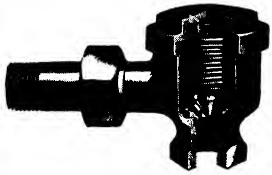
Thermoflex Traps have an exceptionally large orifice. This large orifice combined with high lift, insures fast action and freedom from clogging.

We supply Thermoflex Traps guaranteed for steam pressures of 25 lb, to 50 lb and to 125 lb. Complete information and details of typical installations will be gladly sent on your request. Ask for Catalogue on Thermoflex Heating Specialties.

Valves, Traps, Gauges, Etc.

The Thermoflex line includes: Radiator Traps, Offset Traps, Blast Traps, Drip Traps, High Pressure Traps, Vent Traps, High-grade Packless Inlet Valves, and the Thermoflex Alternator, Thermoflex Compound Gauge, Thermoflex Damper Regulator.

No. 12
Thermoflex Radiator Trap



The full eight-fold Thermoflex-Hydron Bellows is guaranteed because of the Hydron-forming process. Body is heavy bronze construction throughout, with renewable seat.

Fully nickel-plated with highly polished trimmings. The No. 12 is made in angle and in corner patterns, with ½ in. inlet and ½ in. outlet tappings. The inlet neck is double thick to allow for expansion strains. Guaranteed for steam pressures up to 25 lb.

Thermoflex High Pressure Traps

The No. 100A Thermoflex Trap is guaranteed for steam pressures from 50-125 lb. Must not be used where the steam temperature exceeds 400 F.

For use with all types of process work, Laundry Machinery, Kitchen Equipment, Hospital Sterilizers, Vulcanizers, Dry Kilns, Unit Heaters, Street Steam Service, etc., in fact any place that a trap is desired for service at the above pressures.

Small, compact and inexpensive.

Extra heavy body. Renewable nickel steel seat and disc. Bellows made from special bronze tubing and encased in brass sleeve to prevent distortion due to pressure.

Regularly furnished without unions, plain nickel finish. Can be furnished with unions, polished nickel or chromium plated at extra cost.

No. 4 Thermoflex Drip Traps

Used for dripping mains, risers, coils and unit heaters. Semi-steel body, bronze cap and inserted renewable bronze seat, angle pattern only, without unions. Can be used for any general purpose where a finished, nickel-plated trap is not necessary, and at a lower cost. Guaranteed for steam pressures up to 25 lb.

(*See also Pages 993-995*)

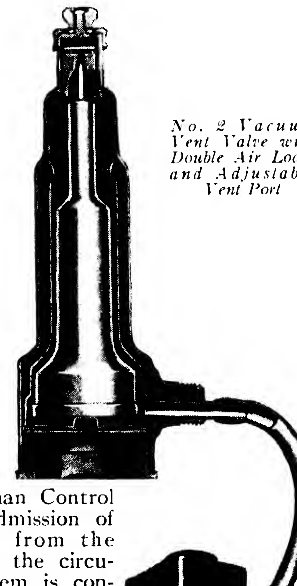
Hoffman Specialty Co., Inc.

Executive Office
500 Fifth Avenue, **New York, N. Y.**
Main Office and Factory: **Waterbury, Conn.**
Sales Representatives in Principal Cities

Manufacturers of Adjustable Port Radiator Venting Valves, Quick Vents an Air Eliminators for One and Two Pipe Steam and Vacuum Systems; Hoffma Supply Valves, Traps and Basement Specialties for Controlled Heat System: Air Conditioner Hoffman-Economy Vacuum and Condensation Pumps, an Hot Water Controlled Heat Equipment.

SIPHON AIR VALVES

The Nos. 1, 70 and 71 are used for venting radiators on One or Two Pipe Steam Systems, and the Nos. 4, 5 and 75 are used in conjunction with these valves for venting steam mains, risers and other quick venting services.



No. 2 Vacuu Vent Valve wi. Double Air Loc. and Adjustab. Vent Port

VACUUM VALVES

The Nos. 2, 77 and 78 Vacuum Air Valves operate on a similar principle as described, but in addition feature the Hoffman Double Air Lock consisting of the vacuum check and vacuum diaphragm. These valves are used on One Pipe Vacuum Systems; and for venting the ends of steam mains or heating risers, where it is also desired to prevent the return of air into the system, the Nos. 6, 16 or 76 Float Vacuum vents are used.

HOT WATER CONTROLLED HEAT EQUIPMENT

The Hoffman Temperature Controller is connected by capillary tubing to the Outdoor Temperature Bulb, and to the Water Temperature Bulb installed in the supply main. Variations in outdoor and circulating water temperatures are instantly transmitted by these two Bulbs to the Temperature Controller which electrically opens or closes the Control Valve.

Temperature Controller

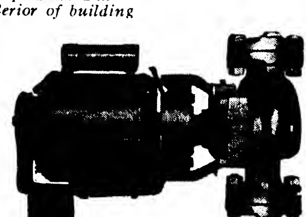
The Hoffman Control Valve. Admission of hot water from the boiler into the circulating system is controlled by this valve. It is opened and closed electrically when actuated by demands for more or less heat from the Hoffman Temperature Controller.

Contro Valve

Avai able sizes corr spo with Hof man C culat

Outdoor Temperature Bulb located on exterior of building

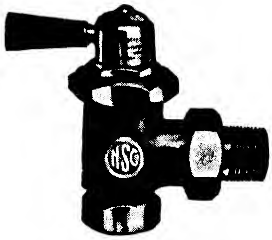
Water Temperature Bulb



Hoffman Circulator

The Hoffman Circulator is a centrifuga pump of large capacity, low power con· sumption and furnished in all standarc sizes. It is installed in the return mair and operates continuously except wher outdoor temperature rises above 65 deg

HOFFMAN CONTROLLED HEAT



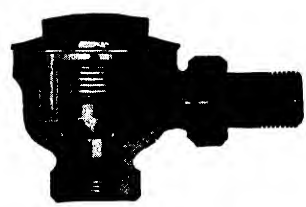
No. 7 Modulating Valve

A Hoffman Controlled Heat System consists of the No. 7 Adjustable Orifice Modulating Valve on the supply end of the radiator, the No. 8-A Thermostatic Trap on the return end and either a Hoffman Differential Loop (for coal-fired installations operating at pressures up to 8 oz), or a Boiler Return Trap where higher pressures are encountered, for returning the condensate to the boiler.

SUPPLY VALVES

Besides the No. 7 Adjustable Orifice Modulating Valve the Nos. 37 and 47 series (not illustrated) represent a complete line of Packless Supply Valves that meet the exacting requirements of architects and engineers.

THERMOSTATIC TRAPS



No. 8A—½ in.

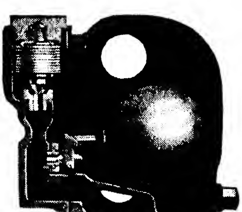
The line of Bellows Type Thermostatic Traps, with hydraulically formed and tested bellows, consists of the Nos. 17-A, 18-A, 8-A and 9-A, and are principally used for low pressure steam or vapor systems. These traps have nominal capacities from 200 sq ft up to 700 sq ft of radiation. The Nos. 8-A and 9-A have renewable elements, which combine the thermostat, valve pin and renewable seat into a single unit.

The No. 10-A Hoffman Trap, which is equipped with waterhammer proof bellows, 1 in. connection, has a nominal capacity of 2800 sq ft.

The Nos. 8 and 9 Traps have a thermostatic element consisting of three chambers each having a top and bottom diaphragm. These chambers are all joined together and the complete thermostatic member is housed in a cage and is not attached to the valve body or cap. This allows the thermostatic element and valve pin to be easily removed and replaced without adjustment. These traps range in sizes from ⅜ in. to 1 in., are medium pressure traps, and are recommended where pressures up to 50 lb are encountered.

The Nos. 20-A and 21-A High Pressure Traps are equipped with waterhammer proof bellows and integral strainer, for use on pressures up to 100 lb. Available in ⅜ in. to 1 in. connection.

DRIP AND HEAVY DUTY TRAPS



No. 50 Series Trap

Where large amounts of condensation are encountered, it is recommended to use one of the float and thermostatic traps, which are available with or without the thermostatic element. These traps are available in large capacities and are mainly used for venting and dripping risers, steam mains, unit heaters, blast coils, etc.

VACUUM AND CONDENSATION PUMPS

The Hoffman-Economy line of Vacuum and Condensation Pumps offers a dependable method of economically returning the condensation from larger heating systems to the boiler. These pumps are made in single and duplex units, for varying capacities and pressures.

HOFFMAN SALES AND SERVICE

Hoffman Products are sold and stocked by leading wholesalers of heating and plumbing supplies everywhere. Hoffman representatives are available to assist in selection of suitable equipment for various services.

ILLINOIS ENGINEERING COMPANY

General Offices and Factory:
Chicago

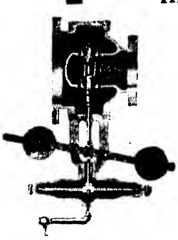
ILLINOIS
HEATING SYSTEMS

Branches and Representatives in Principal Cities

Illinois Motorized Valves (on and off)

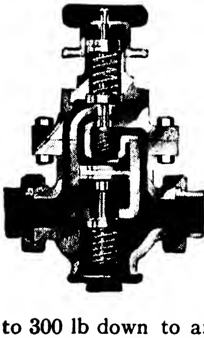
Prevents overheating and fuel waste in large buildings or groups of buildings heated from one central power plant. Buildings may be zoned as to occupancy, time, location, exposure and so on. In many installations this valve has paid for itself in one heating season.

Illinois Reducing Valve



In general use on vacuum or low pressure heating systems. Will reduce to 4 oz pressure from an initial pressure of 150 lb. The large diaphragm insures sensitive operation. Made in both straightway and expanded outlet bodies in sizes from ¾ in. to 12 in.

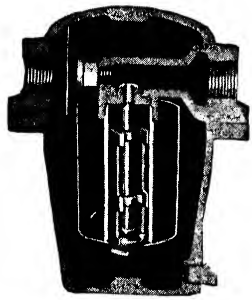
Master Type Pressure Regulator



For exacting requirements, such as tire and rubber vulcanizing, chemical process pressure control, and wherever high pressure steam must be accurately reduced in varying amount to any steady lower pressure not less than 10 lb.

It will reduce initial pressure up to 300 lb down to any lower pressure not less than 10 lb, and does not build up pressure on a closed or dead end line. Made of bronze with Monel metal valves and seats.

Illinois Steam Trap



Series 30

Valve and stem are separate from the bucket and operated only by the bucket at the extreme top and bottom of travel—result—*valve is always either full open or tight closed*. No wire drawing or cutting of valve and seat which are of Monel metal.

Eclipse Spring Controlled Regulating Valve

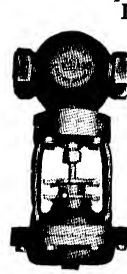


Fig. 121

Furnished in either single seated or double seated type as the service conditions require, for the control of steam, air or gas. Controlling spring is completely enclosed, protecting it from dirt and rust. Valves are furnished with the proper size diaphragm and the proper length spring to give satisfactory service under all operating conditions.

Steam and Oil Separators

Vertical Standard Separators

Eclipse steam separators are made in both horizontal and vertical type, and also the special receiver separators for standard or extra heavy pressures.

Eclipse oil separators are furnished in the horizontal type and have a removable baffle plate to facilitate cleaning of baffle and keeping the separator's efficiency at the highest point.

Write for Bulletins

ILLINOIS ENGINEERING COMPANY

General Offices and Factory:
Chicago

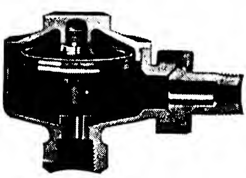
Branches and Representatives in Principal Cities

Illinois Selective Pressure Control Systems


Illinois Selective Controller

An entirely new and unique method of Steam Circulation Control . . . Heating Systems that set new standards in comfort, economy, simplicity and convenience of operation. Each system is individually engineered to meet the exact requirements. Recorded fuel savings, without sacrifice of comfort, warrant your investigation. Ask for Bulletin 16.

Illinois Thermo Radiator Traps


Series G

Illinois Thermo Radiator Traps for vacuum, vapor and low pressure heating systems. Has cone type valve. Flushes thoroughly and seats perfectly at all times. Valve and seat are of Nitralloy. The duplex diaphragm is of special phosphor bronze. Scientific design and rugged construction assure flexibility and long life. These diaphragms have withstood over three million strokes on a breakdown test.

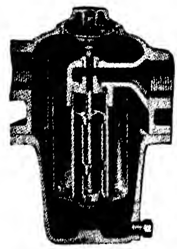
Made in three sizes ½ in., ¾ in. and 1 in. and in a variety of patterns.

Special thermostatic traps can be furnished for working pressures up to 125 lb.

Illinois Modulating Supply Valve

Quick-opening, packless. Steam tight on 50 lb pressure. Large diameter of thread spool and machine cut threads make valve operation easy. Furnished in a complete line of sizes and patterns.

Illinois Combination F & T Traps


Series 7G

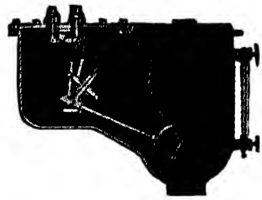
Unsurpassed for draining ventilating units, unit heaters, and for dripping mains and risers—wherever it is desirable quickly to vent air from the main as well as handle the water of condensation in quantity, whether hot or cold.

Illinois Flow Control Valves


Type I

In large installations where process steam or zoning requirements do not permit the variable control of combustion, and in housing projects or on Central Station service, Illinois Flow Control Valves are used. They are of the full floating type, giving complete regulation of steam flow. Furnished for manual, pneumatic or electric operation. Ask for Bulletin 517.

Illinois Return Trap



Automatically returns the condensation to the boiler, regardless of pressure on the boiler up to 8 lb, at the same time discharging the air. Insures positive and complete circulation, and prevents cracked boiler sections.

Trap is self-contained, with no external working parts to be misadjusted, tampered with or injured. No stuffing boxes or packed joints, which insures continuous tightness against air or water leakage.

Write for Bulletins

William S. Haines & Company

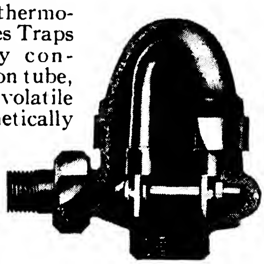
12th and Buttonwood Sts., **Philadelphia, Pa.**

Manufacturers of

EQUIPMENT FOR VAPOR AND VACUUM HEATING SYSTEMS

PRODUCTS—Haines Vento Radiator Traps, Medium Pressure and Blast Type Traps, Combined Float and Thermostatic Traps, Air Eliminators, High Pressure Thermostatic Traps, Boiler Return Traps, Packless Radiator Valves and Modulating Supply Valves.

HAINES RADIATOR TRAPS



The operating thermostat in all Haines Traps is a specially constructed Bourdon tube, charged with a volatile fluid and hermetically sealed. It is the expansion and contraction of the fluid, under varying temperatures, that furnishes the operating power. The thermostatic element is outboard the valve seat closing the valve against the flow of steam.

HAINES MODULATING VALVES

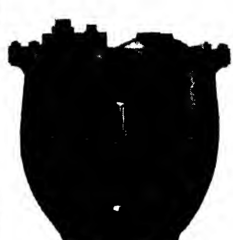
The seat and carrying member construction assures positive leak proof performance. Less than a full turn of the handle completely opens or closes the valve. This valve is packless and made in sizes from ½ in. to 2 in. Can be furnished with wheel or lever handle or lockshield.

HAINES F & T TRAPS

This trap is designed for handling large quantities of condensation such as occur at main line drip points, unit heaters, hot water generators, etc. This trap cannot become air bound as it has a thermostatically controlled bypass. It is light enough to be supported in the pipe line.

HAINES BOILER RETURN TRAPS

For vapor and atmospheric heating systems. Prevents cracked boilers. Assures positive circulation by venting the air and returning the water of condensation to the boiler irrespective of boiler pressure. Weighted valve mechanism prevents wiring drawing of valves. This trap has no stuffing boxes or packed joints to leak air or water.



All Haines material is ruggedly constructed to assure long life and accurately designed for economical operation.

Each device is individually tested, factory adjusted and guaranteed.

Kieley & Mueller, Inc.

Established 1879

Engineering Specialties for Pressure and Flow Control

40 West 13th Street, **New York, N. Y.**

Factory: NEWARK, N. J.

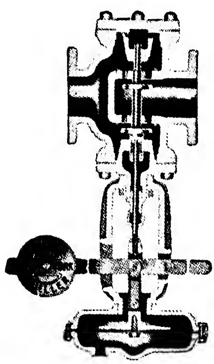
Agents in All Principal Cities

PRODUCTS—Valves: Altitude, Stop and Check, Pressure Regulating, Float, Pilot Reducing, Back Pressure, Tank Control.

Liquid Level Controllers, Water Feeders, Pump Governors, Steam Traps, Y-Type Strainers.

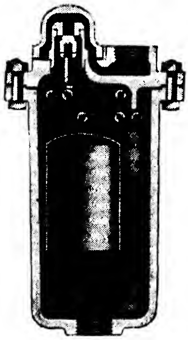
Also Damper Regulators, Hot Water Temperature Controllers, Oil Separators, Steam Separators, Return Traps, Water Columns, etc.

Catalogs sent upon request



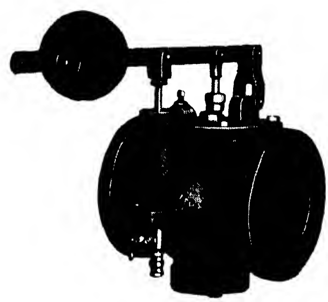
Pressure Regulating Valve

Spring and lever weighted valves for all services and for initial pressures up to 250 lb and reduced pressures from 0 to three-quarters of the initial pressure. Single or double seated in sizes 3/8 to 16 in. Suitable for steam, water, air, oil and gas. Controlled by a small feeler pipe connected from diaphragm to low pressure side.



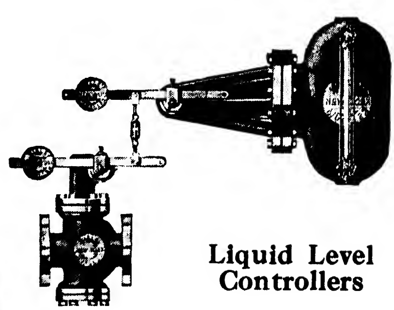
Steam Traps

Large capacity, small sized inverted bucket traps; quick-acting, self-cleaning and non-air binding. Sizes 3/8 to 2 in. Pressures up to 250 lb. Body and cover, semi - steel. Valve and seat, stainless steel. Removable cap allows inside inspection or replacement of valve parts without disturbing pipe connections. (All parts are interchangeable).



Back Pressure and Atmospheric Relief Valve

For use where plant is operated either condensing or non-condensing. Outside air dash pot insures noiseless operation. Maintains exhaust line back pressure from 0 lb to 25 lb. Made horizontal or vertical lever and weight or spring operated.



Liquid Level Controllers

For the accurate control of liquids in tanks or other vessels; suitable for use in industrial plants, gasoline plants, refineries, etc. Direct connected or remote control; ball bearing spindle and easy-to-pack stuffing box; rotary or sliding valve. Write for special bulletin C-3.

Milwaukee Valve Company

Milwaukee — Wisconsin

Manufacturers of the Complete Line of Vapor and Vacuum MILVACO Heating Specialties — APPROVED Bronze Valves

Each unit is scientifically designed and constructed by precision methods, and carefully inspected and pre-tested to insure uniformity, higher quality and finer workmanship. MILVACO representatives, located in principal cities, render intelligent, courteous service.

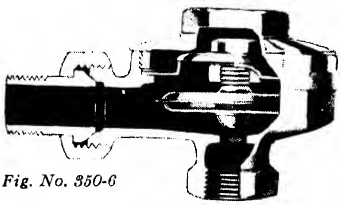


Fig. No. 350-6

Diaphragm Thermostatic Trap. Positive seating accuracy; sensitive operation. 15 lb pressure to 25 in. vacuum. Nickel finish rough body and polished cover, tailpiece and nut. Sizes ½ in. and ¾ in. Cap. 200-400 sq ft E.D.R.

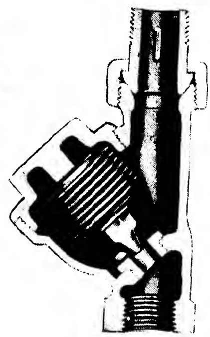


Fig. No. 25-2ST

Straight Through Bellows Type Trap. Size ½ in. Finish same as No. 25-4. Cap. 200 sq ft.

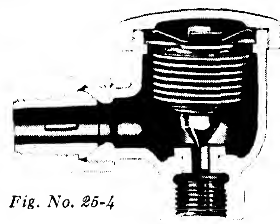


Fig. No. 25-4

Bellows Type Thermostatic Trap. Phosphor bronze bellows, hydraulically formed. MILVACO cage construction employed. 25 in. vacuum to 25 lb pressure. Nickel rough body with polished trimmings. Sizes ½ in. and ¾ in. Cap. 400 sq ft.

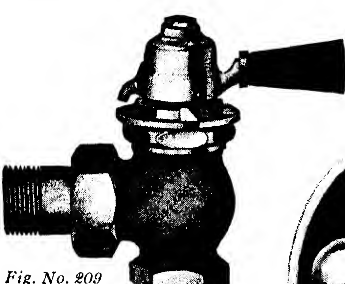


Fig. No. 209

Packless Radiator Angle Valve. Lever handle, graduated. Regularly furnished with rough nickel plated body, finished nickel plated trimmings. Designed for low pressure and vacuum heating systems. Sizes ½ in. to 2 in. incl.

MILVACO Radiator Valves and Thermostatic Traps are also available in chrome finish.

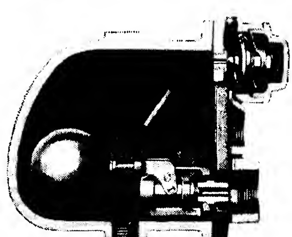


Fig. No. 13 Series

Float and Thermostatic Trap. Self-aligning float valve with reversible monel metal seat. Diaphragm thermal element. Baffle plate prevents condensate from flowing directly on ball float. Sizes ½ in. to 2½ in. incl. Cap. 200 lb to 8000 lb per hr. at 2 lb press. differential.

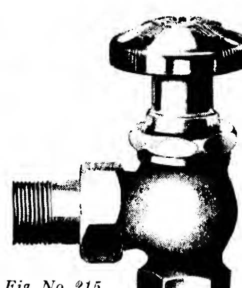


Fig. No. 215

Packless Radiator Angle Valve. Finish same as No. 209. Also available in lock and shield pattern. MILVACO Radiator Valves cannot bind or stick in operation. Sizes ½ in. to 2 in. incl.

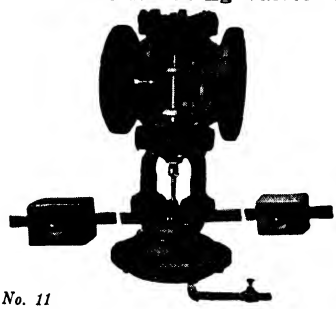
For complete detailed information send for condensed catalog of Milvaco Heating Specialties.

Mueller Steam Specialty Co., Inc.

349-351 West 26th Street, New York City

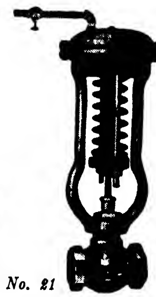
Steam, Water, Air, Oil and Gas Specialties for Heating and Power Plants

Pressure Reducing Valves—Straight Pattern and With Increased Outlet



No. 11

No. 17



No. 21

No. 11—For Vacuum, Vapor and Low Pressure Heating Systems. Initial Pressures, up to 200 lb; Reduced Pressures, 0 to 10 lb.

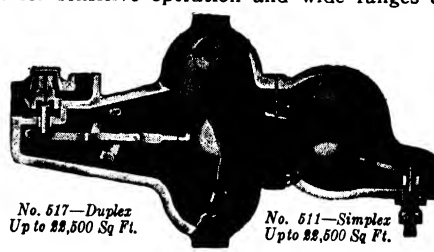
No. 17 and 21—For automatic control of reduced pressures on dead-end service, requiring a tight closing valve, such as tank heaters, kitchen utensils, sterilizing apparatus, laundry equipment, kettles, cookers, driers, etc. Initial Pressures up to 200 lb. Reduced Pressures 0 to 150 lb.

Constructed with full globe bodies. Center guide eliminates the wings on discs, and increases efficiency, assures minimum noise and prolongs the life of the seats and discs. Lever and weight operates on a steel roller bolt, assuring a most sensitive valve. Spring type furnished with special long springs for sensitive operation and wide ranges of reduced pressures.

Automatic Water Feeders

With a powerful leverage to control the water line in steam boilers, etc. They supply make-up water to compensate for evaporation, leaks, steam utilized in process work and condensation wasted. Where condensation held up in the system eventually returns in large quantities, our Duplex type protects the boiler against flooding. All working parts of non-corrosive metal, are accessible without breaking pipe connections. Provided with an integral strainer. For steam pressures up to 100 lb, water pressures up to 120 lb.

Equipped with low water and pressure Mercoid Tube Switches for all services.



No. 517—Duplex Up to 22,500 Sq Ft.

No. 511—Simplex Up to 22,500 Sq Ft.

Steam Traps

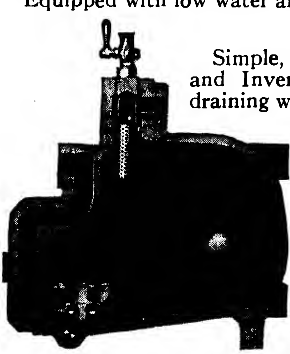
Simple, Sturdy and Compact Ball Float and Inverted Bucket Steam Traps for draining water of condensation from steam apparatus and steam mains.

Powerful leverage enables them to take care of large quantities of condensation.

Ball Float Steam Traps equipped with integral strainer, water gages, air cocks, blow-off and integral by-pass valve, when desired.

All working parts are accessible without disturbing any pipes.

Valves are sealed with several inches of water, making the escape of steam impossible.



Ball Float
No. 219—Up to 30 lb.
No. 221—Up to 150 lb.
Sizes ½ to 3 in.

Inverted Bucket
No. 211—For Pressures Up to 250 lb.
Sizes ½ to 2 in.

Catalogue and Bulletins covering our Complete Line gladly furnished on application.

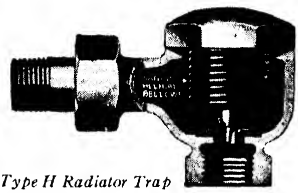
Sarco Company, Inc.

183 Madison Ave., **New York, N. Y.**

Branches in Principal Cities

SARCO CANADA LIMITED, FEDERAL BLDG., TORONTO, ONT.

PRODUCTS—A complete line of Specialities for Vapor, Vacuum and Gravity Steam Heating Systems and Control combined with a competent Engineering Service to architects and heating engineers to assist them in providing modern heating.

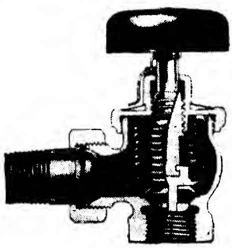


Type H Radiator Trap

SARCO RADIATOR TRAPS

Sarco Heating Systems are "prestige Systems." The traps and valves *are* the system as far as maintenance and cost is concerned.

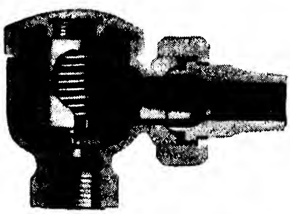
Sarco Type H Traps—Are available in angle, straightway, and corner patterns. The Sarco Thermostatic Bellows—made by special machinery, has not been duplicated or even imitated with success. It works efficiently, repeatedly and persistently. It has worked that way for a quarter of a century. Sizes ½ in. to 1 in.



Bellows-Packless Valve

SARCO RADIATOR VALVES

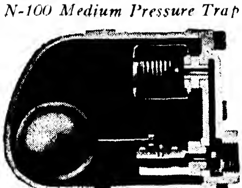
Sarco Packless Valves—Used for one and two pipe heating systems and are truly packless. Steam leaks are impossible. Furnished with round or lever handles or lock shield in angle straightway, or corner patterns. Sizes ½ in. to 1½ in.



N-100 Medium Pressure Trap

SARCO N-100 TRAP

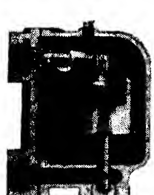
For high pressure radiators and heating coils in stationary and marine service, and for hospital and kitchen equipment. Has full length protecting shield and stainless steel valve head and seat. Sizes ⅜ in. to 1 in.



Float-Thermostatic Trap

SARCO FLOAT-THERMOSTATIC TRAPS

For dripping ends of mains and risers, and for stack or blast heaters, large unit heaters and hot water generators. Automatic thermostatic air vents built in. Available in six sizes with connections ¾ to 2 in.



Inverted Bucket Trap

SARCO INVERTED BUCKET TRAPS

Are recommended for high pressure unit heaters and sometimes preferred for kitchen and laundry equipment. Strainers are built right into these sturdy traps. Seats and valves are stainless steel and renewable. Thermostatic air vents can be furnished on the larger sizes. Available in sizes ½ in. to 2 in. for pressures up to 900 lb. Catalog HV-165.

See Sarco Catalog HV-45 for all heating specialities

SARCO ALTERNATING RECEIVER

A complete line of boiler return traps for vapor systems.

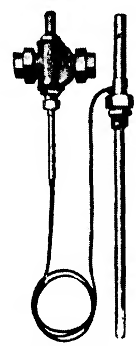
Returns water of condensation to boiler automatically, thereby assuring positive return of water under all pressure conditions.

Made in six sizes for from 1500 to 25,000 sq ft of radiation. Catalog HV-45.

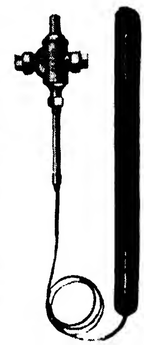
Alternating Receiver

SARCO AIR ELIMINATORS

For venting air from vapor systems at one central point in the basement. Available in two sizes: No. 6 for systems up to 2500 sq ft and No. 12 for 15,000 sq ft. Both are equipped with float valves to stop water escaping through the vent and with check valves to prevent ingress of air when system is under vacuum.



Type TR-21 Standard for hot water storage tanks, fan units, etc.



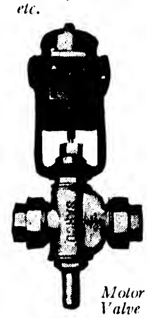
Type KR-14 Designed for room control and air conditioning.

SARCO SELF-CONTAINED TEMPERATURE REGULATORS

Sarco Temperature Regulators are simple, self-operated valves—the only self-contained units that use the irresistible force of liquid expansion. No stuffing boxes to leak, no auxiliary "power" required; all moving parts are *inside* the equipment. Here again—a type and size for every purpose—for steam, gas, oil, water or brine for temperatures ranging from 0 to 400 F. Catalog HV-52.

SARCO ELECTRIC CONTROLS

Comprise room thermostats, aquastats, and limit controls for all heating and air conditioning needs; also motor valves for steam, water, brine, or freon. The Sarco Graduator System provides simple mechanical control of building heating, direct by the weather. Catalogs HV-150 and HV-128.



Motor Valve

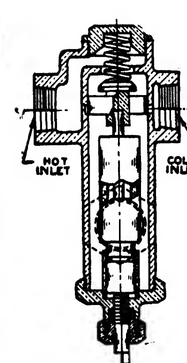
Room Thermostat

SARCO WATER BLENDER AND TEMPERING VALVES

For mixing hot and cold water to deliver automatically water at any desired temperature. Two models are available, type IB for showers, wash basins, etc., and type DB, a tempering valve for use with submerged heating coils or tankless heaters. Catalogs HV-140 and 839.

See Sarco Catalog HV-45 for all heating specialities

Water Blender



Tempering Valve

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating

-since 1888 Webster Systems of Steam Heating

Main Office and Factory:
Camden, New Jersey

Representatives in over 60 cities—
Consult Your Local Phone Directory

Webster Nesbitt UNIT HEATERS

PRODUCTS AND SERVICES

Improved Webster Systems of Steam Heating including Vacuum and Type "R" (vapor).

Webster Central Control Systems including HYLO and MODERATOR.

Modernization of Obsolete and Faulty Heating Systems.

Webster System Equipment including Light-Weight Concealed Radiation (Gravity Convection Heaters), Webster-Nesbitt Unit Heaters, Radiator Supply Valves, Metering Orifices, Thermostatic Traps, Drip Traps, Heavy Duty Traps, Dirt Strainers, Dirt Pockets, Boiler Return Traps, Vent Traps, Damper Regulators, Boiler Protectors, Lift Fittings, Expansion Joints, Separating Tanks, Steam and Oil Separators, Steam Vacuum Pump Governors, Air Separating and Receiving Tanks, Gages, Water Accumulators.

Webster Series "78" and Series "79" Traps for use at process pressures (10 to 125 lb per sq in.)

IMPROVED WEBSTER SYSTEMS

The Improved Webster Systems are low pressure, two-pipe systems of steam circulation ***with the addition*** of accurately-sized metering orifices at radiator supply connections and, when required, intermediate metering orifices at points in branch mains. Metering orifices effect even distribution of steam to all parts of the heating system and permit the successful application of a centralized control. Webster Valves are used at supply of radiators Webster Thermostatic Traps prevent flow of steam into return mains when radiators are filled. Webster Drip Traps and Dirt Strainers are used where needed on steam mains. Improved Webster Systems are available for vacuum, open return or "vapor" operation. The Type "R" System corresponds to the so-called Vapor type. Fig. 1 illustrates a typical arrangement of Boiler Return Trap, Vent Trap, etc., when low pressure boiler is the source of steam.

CENTRAL CONTROL SYSTEMS

These are patented systems for varying the amount of steam to all radiators according to outside temperature. They provide *continuous heat delivery* with effective *fractional filling of radiators.* The Hylo Systems may be provided for manual control, or if desired, may be semi-automatic by incorporation of inside thermostat or thermostat and schedule clock. The Moderator Systems employ an automatic Outdoor Thermostat supplemented by a manual Variator.

The latter is used for quick heating-up, night load, and unusual weather or occupancy conditions. Use of Webster Central Control Systems results in (1) increased comfort because over-heating and underheating are minimized and (2) lower fuel or steam costs.

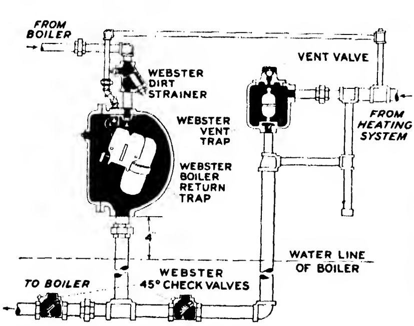


Fig. 1. Conventional arrangement of piping around Webster Basement Equipment for the Webster Type "R" System

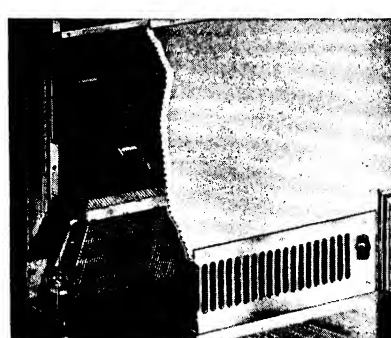


Fig 2. Webster System Radiation

WEBSTER SYSTEM RADIATION

Concealed, non-ferrous type for use exclusively with Improved Webster Systems. Is unique in that it combines *in a single unit,* a light-weight heating element of high efficiency with an orificed radiator supply valve, a radiator trap and supply and return piping connections. Metal enclosures for installation within the wall and exposed metal cabinets are available. Webster System Radiation and enclosures are so designed that the entire heating element can be quickly removed without damage to plaster or paint. Space requirements reduced to a minimum and installation greatly simplified.

RADIATOR SUPPLY VALVES

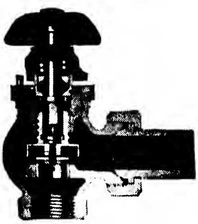


Fig. 3. Webster Three-Point Valve

"Three-Point" The finest Webster Valve. Has sleeve orifice in seat opening. Can be used as *open-shut* operator or "three-point" providing *shut-normal-excess. Excess* permits 140% of normal heat with Webster Central Controls. Especially suitable for hospitals, hotels, homes where extra flexibility is desired. Quick-opening, non-rising stem type. Uses molded ring packing which meets usual "packless" specifications. Available in angle, right corner, left corner and single union straightway models with bakelite wheel handle, or lever, lockshield or extended stem fixtures. Sizes ¾ and 1 in. only.

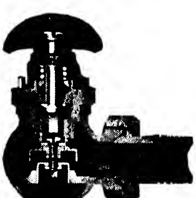


Fig. 4. Webster Type "W" Valve

Type "W"—Same high quality as "Three-Point" Valve except "Three-Point" features are omitted and modulating device included. Successful "modulation" requires that proper pressure be maintained on system. Made in angle model in sizes of ½, ¾, 1 and 1¼ in. Right-corner, left-corner, straightway single union and double union models in selected sizes. Choice of wheel, lever, lockshield, chain wheel or extended handles.

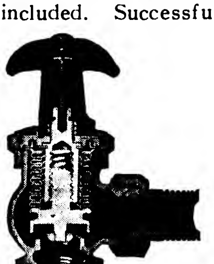


Fig. 5. Webster Packless Valve

Sylphon Packless—A high quality valve incorporating a Sylphon Bellows completely enclosing a non-rising stem and fully meeting "bellows packless" specification. Made in angle model in sizes of ½, ¾, 1, 1¼, 1½ and 2 in. Right-corner, left-corner and straightway single-union models in sizes of ½, ¾, and 1 in. Choice of lever, wheel, lockshield, chain wheel or extended stem handles.

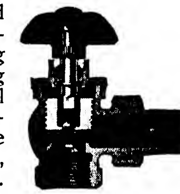


Fig. 6. Webster Type "B" Valve

Type "B"—A good quality valve. Quick-opening. Non-rising stem. Molded ring packing meets usual "packless" specification. Made in angle model in sizes of ½, ¾, 1, 1¼, 1½ and 2 in. Right-corner, left-corner and straightway single-union and double-union models in sizes of ½, ¾ and 1 in. Choice of wheel, lever, lockshield, chain wheel and extended stem handles.

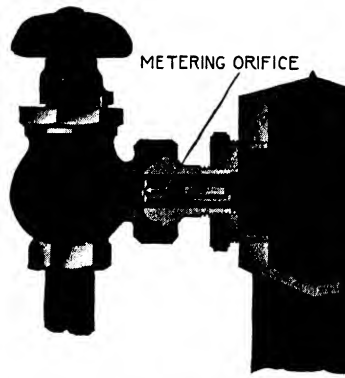


Fig. 7. Metering Orifice Inserted in Union Connection of a Webster Supply Valve

Metering Orifices—Accurately sized and made of heavy gage Monel Metal to resist erosion and corrosion, amply thick to be free from vibration and shaped for quiet operation. Available in a number of types to fit both new and installed radiator supply valves of Webster or other make.

RETURN TRAPS

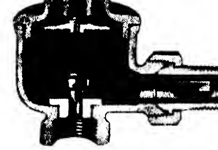


Fig. 8. Webster 502 Sylphon Trap

Sylphon—Perfected thermostatic bellows trap, *fully compensated for pressure.* Stainless steel valve piece and renewable seat. Factory adjusted. Made in angle, right-

corner, left-corner, vertical, and straightway bodies. Sizes: ½, ¾ and 1 in. Normal operating pressures up to 15 lb per sq in. Maximum occasional pressure 25 lb per sq in.

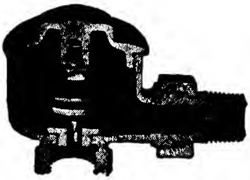


Fig. 9. Webster Size 702-M Trap

Series "7M"—Perfected diaphragm-type thermostatic trap, *fully compensated for pressure.* Uses Monel Metal diaphragm, Stainless Steel valve piece and seat insert. Renewable seat. Factory adjusted. Made in angle, right-corner, left-corner, vertical, and straightway bodies. Sizes: ½, ¾ and 1 in. Normal operating pressures up to 25 lb per sq in.; maximum occasional pressure 50 lb per sq in.

Series 7 with phosphor-bronze diaphragm can be used where normal operating pressures do not exceed 15 lb per sq in.

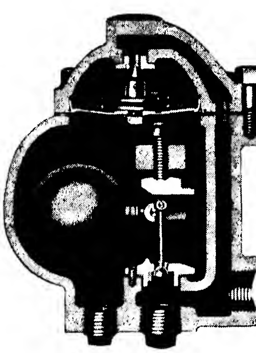


Fig. 10. The Webster Size 0026-T Drip Trap is Rated 700 lb Water per Hour at 2 lb Pressure Difference

Series "26"—A heavy duty trap for drips of mains, blast radiation, unit heaters, hot water generators and similar applications. A rugged float-type trap available with and without thermostatic air vent. Made in five sizes: 200, 700, 1200, 2400 and 5000 lb water per hour at 2 lb pressure difference. Maximum working pressure is 15 lb per sq in.

Fig. 11. Webster Size 782 Trap

Series "78" — thermostatic trap *built for* process steam pressures (10 to 125 lb per sq in.). Monel Metal diaphragm. Stainless Steel valve piece and seat insert. Angle model only. Sizes: ⅜, ½, ¾ and 1 in. Extensively used with laundry, cooking, sterilizing and other process-steam uses.

Series "79"—For use where large volumes of very hot condensate form more quickly than can be discharged by thermostatic traps alone. Float and thermostatic traps designed for normal working pressures between 15 and 125 lb per sq in. Water of condensation is passed through a float-controlled seat opening while air is discharged into the return piping by a thermostatically controlled vent. Compact and light in weight. Can be readily mounted in a pipe line without other support. Available with either ¾ in. or 1 in. inlet and outlet.

Cast iron body, copper asbestos gasket and cover bolted together with steel cap screws. Monel Metal valve piece and stem. Stainless steel seat. Air vent unit is Monel Metal diaphragm with Stainless Steel valve piece and brass seat with Stainless Steel insert.

DIRT STRAINERS AND POCKETS

Placed in return lines of steam heating systems to prevent dirt, rust and scale from impairing tightness of traps.

Fig. 12. Size 34C-1 Webster Boiler Protector with Low Water Electrical Cut-out Switch. Size 34 has no Cut-out Switch

BOILER PROTECTOR

Prevents breakage in low pressure heating boilers when water level becomes inadequate. Automatically supplies raw water to boiler when water level drops to 1 in. above bottom of gage glass.

For maximum boiler pressure of 15 lb per sq in. Maximum cold water main pressure should not exceed 150 lb per sq in.; minimum must not be less than 25 lb per sq in.

Made with ¾ in. connections, with or without electrical cut-out switch.

PROPELLER-FAN TYPE

Webster-Nesbitt Unit Heaters are manfactured by John J. Nesbitt, Inc., Holmesburg, Philadelphia, Pa., and are distributed solely through Warren Webster & Company, Camden, New Jersey.

Webster-Nesbitt Unit Heaters are designed to circulate large volumes of air at comparatively low temperatures. The heated air is mixed thoroughly with the room air to reduce overheating in upper areas and temperature stratification, and to assure quick heating, low fuel costs, and complete comfort for room occupants.

All ratings of Webster-Nesbitt Unit Heaters are based on tests made in accordance with the standard test code of Industrial Unit Heater Association and A.S.H.V.E.

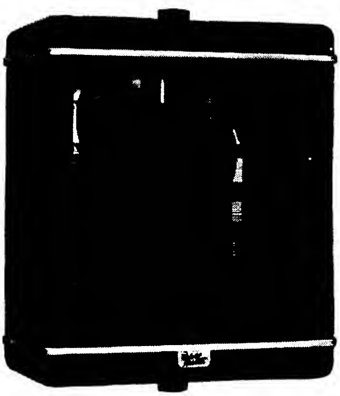


Fig. 13. Webster-Nesbitt Unit Heater, Propeller-fan Type

Seven unit sizes—air capacities, at maximum fan speed, from 495 cfm to 4580 cfm. **Three types of heating element** for each size, affording wide range of final temperatures and permitting flexibility in selection. **Modern casing** of heavy furniture steel, die-formed and welded, rounded corners, black featherweave finish, stainless steel trim. **All-copper heating element** of tube-and-fin construction, guaranteed for working steam pressures up to 150 lbs gauge. **Steam and return headers** of heavy seamless steel tubing. **Freedom for expansion,** because heating assembly is held in casing by snugly fitted angle guides. **Center steam and return connections** permit four smaller units to be suspended direct from steam piping. **Quiet fans** of four-blade type; four smaller units have extra wide overlapping blades for especially quiet service. **Rubber-mounted motors;** and **Adjustable discharge louvres.**

Send for Catalog W-N 100.

Fig. 14. Webster-Nesbitt Unit Heater, Centrifugal-fan Type

GIANT HEATERS

Centrifugal-Fan Type for the economical heating or large areas. Floor-mounted, wall-mounted, horizontal-suspended and inverted types; in sizes and capacities, from 3330 to 16,000 cfm, and from 125,000 to 1,008,000 Btu, with 2 lb steam, 60 deg entering air. With or without Thermadjust Temperature Control Damper which prevents overheating and stratification, and effects fuel savings. Also available with Nesbitt Steam-distributing Tube Heating Surface for installations employing modulating steam valve control. Quality throughout, from the sturdy, efficient copper tube-and-fin radiators to the fans, motors, and durable, attractive casings. Giants of efficiency and endurance.

Send for catalog W-N 104.

Fig. 15. Webster-Nesbitt Series F Unit Heater

SERIES F UNIT HEATERS

Centrifugal fan units for quiet and efficient circulation of heated air in offices, stores, showrooms, restaurants, halls vestibules, etc. Four casing sizes, each with two radiator sizes; capacities (60 deg entering air; 2 lb steam) from 94 sq ft EDR to 423 sq ft EDR; 282 cfm to 1355 cfm. Send for Publication W-N 105.

Wright-Austin Co.

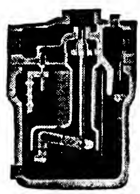
317 West Woodbridge St., **Detroit, Mich.**

PRODUCTS—Steam Traps, Strainers, Air Traps, Steam and Oil Separators, Compressed Air Purifiers, Exhaust Heads, Boiler Feeders, Alarm Water Columns, Water Gauges, Trycocks.

"Airxpel" Bucket Type Steam Traps

Are "double duty" traps, because they automatically discharge both air and condensate.

Union connections make them easy to connect up. Also, furnished with screw connections when desired. They save money for fittings and installation labor, by having straight through horizontal pipe connections.



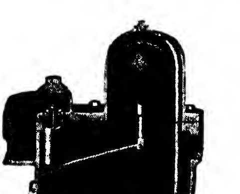
The Cub sizes are made in ½ in., ¾ in., 1 in. Especially suitable for individual unit drainage on heating and process equipment.

Also three "Master" sizes ½ in. to 2 in., for general service.

"Combination" Steam Trap

Float Type with internal thermostatic air bypass and strainer. A modernly designed and very successful trap for vacuum and pressure heating.

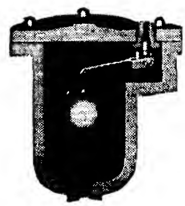
"Victor" Low Pressure Steam Trap



A heavy duty trap for large volumes of condensation at low pressures.

"Emergency" Float Type Steam Trap

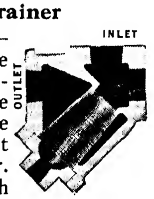
Three valve trap with large capacity at high pressures. An exceptionally reliable trap for use in inaccessible places.



Air Relief Trap

For relieving air from forced circulation hot water heating systems, water supply lines, closed tanks, receivers, pumps, etc.

"Tuway" Strainer

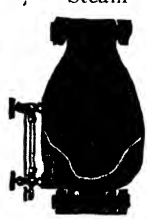


May be used two ways—as a straight-way or angle strainer, in either horizontal or vertical pipe line, because it has the choice of two inlets at right angles to one another.

For cleaning, flush through blow-off connection, or remove screen by unscrewing bottom plug.

Separators—Steam and Oil

Type "A" Vertical Steam



Type "S" Horizontal Oil

We make separators of every type and all sizes for all pressures.

Exhaust Head

Designed to eliminate noise and spray. Three types to select from—the "Cyclone" Heavy Duty, and Standard Galvanized Steel—also, the cast iron type, to remedy all conditions. Sizes 1 in. to 48 in.

Send for descriptive Bulletins on any of the items listed on this page.

Yarnall-Waring Company

Manufacturers of YARWAY **Steam Specialties**

7600 Queen Street, **Philadelphia, Pa.**

YARWAY IMPULSE STEAM TRAPS

Construction—The Yarway Impulse Steam Trap is unique in that there is only one moving part, the simple valve F. This trap is made of bar stock throughout, no castings used. Body and bonnet of cold rolled steel, cadmium plated; cap of tobin bronze, valve and seat of heat treated stainless steel. For pressures 400 to 600 lb, bonnet and cap are stainless steel.

Operation—When handling condensate, pressure required to lift valve F is greater than reduced pressure in control chamber K; therefore, valve F opens, allowing free discharge of condensate. As remaining condensate approaches steam temperature, flashing takes place, flow through center orifice is choked and pressure builds up in control chamber K, closing valve F.

Advantages

Light Weight—Yarway traps need no support—½ in. trap weighs only 1⅜ lb. 2 in. trap weighs 8⅝ lb.

Small Size—They practically eliminate radiation losses—can be installed in cramped quarters—½ in. trap measures 2¼ in. long—2 in. trap, 4¾ in. long.

Will not air bind.

Require no priming.

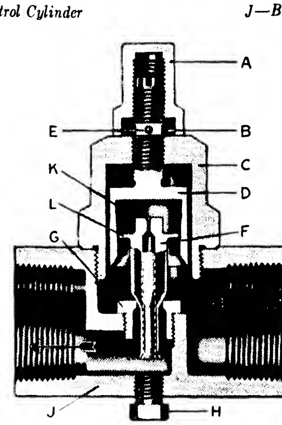
Insure quick heating.

Operate on exclusive Impulse principle (*U. S. Patents No. 2,051,732 and 2,127,649.*)

Low Price—Often cheaper than repairing old traps.

Factory set to operate at all pressures up to 400 lb (or 600 lb) without change of valve seat.

A—Cap Nut *B—Lock Nut* *C—Bonnet* *D—Control Cylinder* *E—Lock Pin* *F—Valve* *K—Control Chamber* *L—Control Disc* *G—Valve Seat* *H—Test Plug* *J—Body*



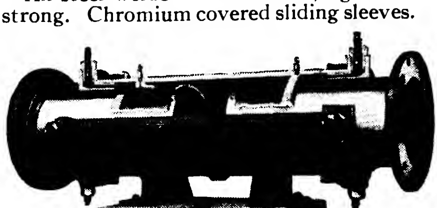
Prices, Weights and Dimensions

Size In.	Trap No.			Prices	Wt Lb
	400 lb 450 F	600 lb 550 F	600 lb 750 F	Trap Complete	
½	60	70	120	$15.00	1⅜
¾	61	71	121	22.00	2
1	63	73	123	31.00	2¾
1¼	64	74	124	48.00	4
1½	66	76	126	68.00	6
2	67	77	127	90.00	8⅝

For further information send for descriptive bulletin T-1734.

YARWAY GUN-PAKT EXPANSION JOINTS

All-steel welded construction; light but strong. Chromium covered sliding sleeves.



Cylinder guide and stuffing box integral, assuring perfect alignment. Internal limit stops. Gun-pakt and Gland-pakt types; Gun-pakt (illustrated) fitted with screw guns which permit insertion of plastic packing while joint is under pressure. Sizes 2 in. to 24 in., single end or double end, flanged or welding ends; 150, 300 and 400 lb pressures. For additional details send for bulletin EJ-1906.

The Brownell Company

ESTABLISHED 1855

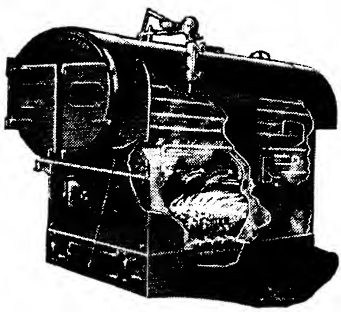
Dayton, Ohio

Manufacturers of

BROWNELL BOILERS AND STOKERS

Representatives in All Principal Cities

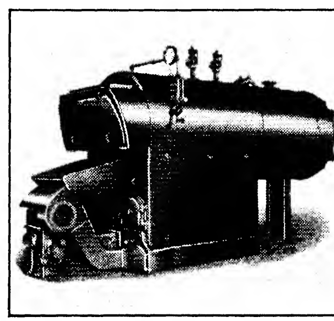
Power Boilers of various Fire Tube Types; Steel Heating Boilers; Underfeed Stokers and Steel Plate Work.



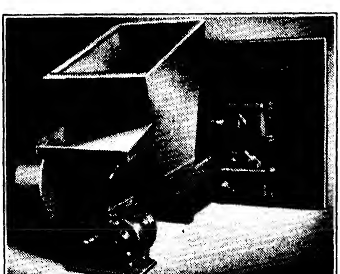
Brownell Standard Welded Smokeless Boiler

Welded Boilers built in Standard and Master types. "Standard" type, Direct Draft or Smokeless, Coal Hand Fired— 500 to 34,000 sq ft. Stoker Fired— 930 to 43,130 sq ft. "Master" type, Coal Hand Fired—500 to 35,520 sq ft. Stoker Fired— 1070 to 45,110 sq ft.

Riveted Double Pass Firebox Boilers built for both high and low pressure. Coal Hand Fired—4000 to 35,000 sq ft. Stoker Fired—4860 to 42,500 sq ft.



Brownell Riveted Double Pass Firebox Boiler, Stoker Fired

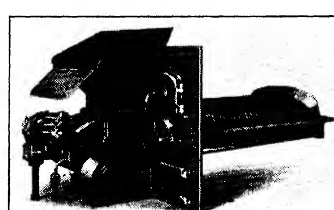


Brownell Type "C" Side Dump Stoker

The Brownell Type "C" Stoker is built in both stationary dead plate and side dump types, in sizes suitable for medium and large industrial and heating plants.

The Brownell Type "R" Heavy Duty Ram Feed Stoker is designed for use in the larger industrial and heating plants. Ruggedness, flexibility, economy and low maintenance costs are features of this stoker.

Brownell Matched Units, Boiler and Stoker combinations, are available in sizes from 500 sq ft to 45,000 sq ft.



Brownell Type "R" Heavy Duty Stoker

In addition to the above, a complete line of Domestic Stokers is built. These Stokers are built in either ratchet or continuous feed.

Send for Descriptive Bulletins

Combustion Engineering Company, Inc.

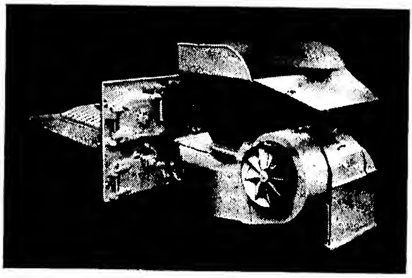
All Types of Fire and Water-Tube Boilers
Mechanical Stokers

Complete Steam Generating Units
Pulverized Fuel Systems

200 Madison Avenue, **New York, N. Y.**

Offices in all principal cities of the United States and Canada

More than 14,000 C-E Stokers installed to date

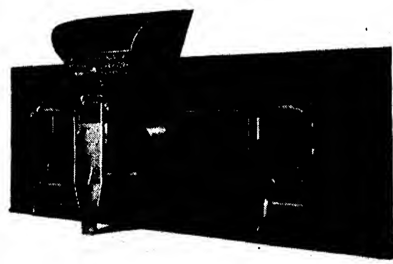


C-E SKELLY STOKER UNIT

A compact, self-contained unit adapted to burn either anthracite or bituminous coal. Alternate fixed and moving grate bars assure lateral distribution of fuel. An integral forced-draft fan, with vortex inlet control, permits positive regulation of air-coal ratio. Automatic control is standard equipment. Approximate application range —20 to 200 rated boiler hp.

C-E SPREADER STOKER

A simple, rugged stoker designed to burn a wide variety of coals. Hopper, feeding and distributing mechanism, variable-speed drive and motor are combined in a compact unit. A series of rotating spreader blades feeds coal into the furnace in criss-crossing streams which assure uniform distribution. Fines are burned in suspension and the rest of the coal is burned on a grate which may be of either the stationary or dumping type. Grate surface is zoned for regulating air admission and to facilitate cleaning. All parts subject to wear are readily accessible for inspection, adjustment or replacement, when necessary. Rate of fuel feed and air supply may be regulated over a wide range and are readily adaptable to automatic control. Applicable to boiler units from about 100 hp up.



TYPE E STOKER

A single-retort, underfeed stoker with an established reputation of many years' standing for dependable service. Designed to burn a variety of bituminous coals under boilers up to about 600 rated hp. Available with either steam, electric or hydraulic drive.

OTHER C-E STOKERS

Type K Stoker—A single-retort, underfeed stoker for burning bituminous coals under boilers in the upper size range of the C-E Skelly Stoker Unit.

C-E Multiple Retort Stoker—For burning bituminous and semi-bituminous coals under boilers up to the largest sizes.

C-E Traveling Grate Stokers—Including both Coxe and Green types. Available with grate surfaces suitable for anthracite, coke breeze, lignite or bituminous coal, as required. Chain grate types are built for either forced- or natural-draft application.

C-E BOILERS

All fire tube and water tube types in sizes ranging from 25 hp up to the largest. Standard and special designs to suit all conditions of fuel, load and space. Included are all types formerly known by the trade names "Heine," "Walsh & Weidner," "Casey-Hedges," "Ladd" and "Nuway".

Detroit Stoker Company

Sales and Engineering Offices
General Motors Bldg., Detroit, Mich.

Main Offices and Works at
Monroe, Mich.

Since 1898

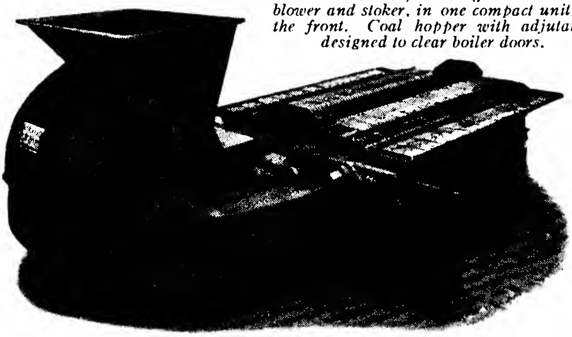
District Offices in
Principal Cities

Built in
Canada at London, Ont.

Detroit Stokers provide the **Economical Method of Burning Coal.** Built in many types and capacities for all heating and power requirements. Features embodied in the various designs represent 40 years' experience in Stoker manufacture exclusively.

Complete Line to Meet Every Need: Detroit LoStoker, Detroit UniStoker, Detroit Single Retort Stoker, Detroit Double Retort Stoker, Detroit Triple Retort Stoker, Detroit Multiple Retort Stoker and Detroit RotoStoker.

Recommendations covering individual requirements submitted upon request. District Offices located in principal cities; or write Detroit Stoker Company, Detroit, Michigan.



Detroit LoStoker, showing motor driven blower and stoker, in one compact unit at the front. Coal hopper with adjutator designed to clear boiler doors.

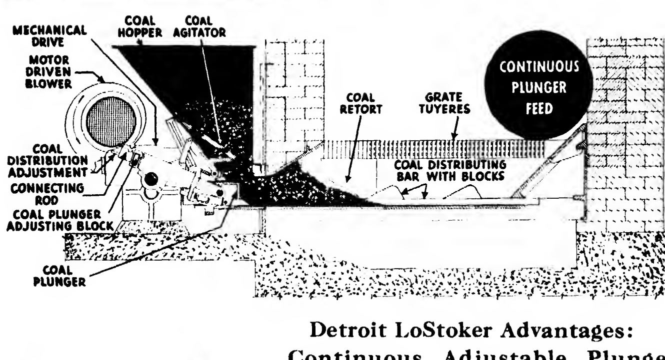
Detroit LoStoker

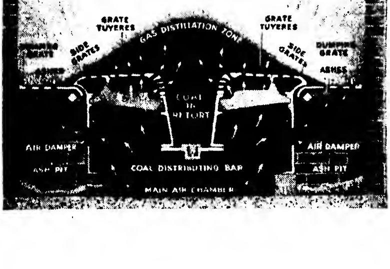
Many grate area sizes and capacities to fit furnaces of all types of boilers. Side cleaning, with provision for admitting air under the dumping grates at each side to burn out the combustible prior to dumping ashes. Compact, easily installed, responsive and automatic. Highly efficient, saves coal. Successfully burns less expensive grades of coal.

Detroit LoStokers are mechanically driven through machine cut worms and gears fully enclosed, running in oil, require little power for operation.

•

Large active fuel bed to fit the furnace and provide sufficient grate area to carry heavy loads and still operate economically with light loads.





Detroit LoStoker Advantages:

Continuous Adjustable Plunger Feed with control of the quantity of coal fed and its distribution.

Heavy Mechanical Drive of simple design, requires little power.

Side Cleaning with dumping grates, ashes removed through doors provided in the Stoker front. No hand cleaning.

Agitator in coal hopper for continuous coal feed, cannot stick or jam with wet coal.

Automatically Controlled. Motor or steam turbine driven, controlled from steam pressure, water temperature or thermostat.

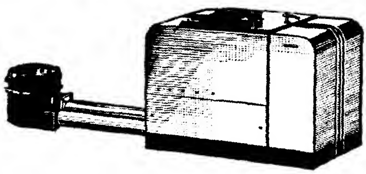
ECON-O-COL STOKER DIVISION

OF COTTA TRANSMISSION CORP.

MAIN OFFICE AND PLANT

ROCKFORD, ILLINOIS

DEALERS IN PRINCIPAL CITIES THROUGHOUT AMERICA



Streamline Domestic Model

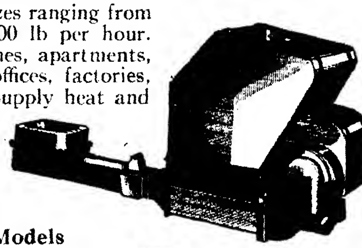
Finished in smart green and black with chrome trim. A high quality product, now giving complete satisfaction in thousands of homes throughout America. Three sizes.

Bin-Feed Models

Two styles: "Transfer" or "Pull-Thru." Install delivery tube above or below floor. Use in new or old homes.

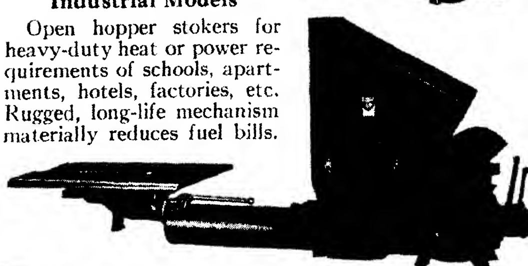
Domestic and Commercial Models

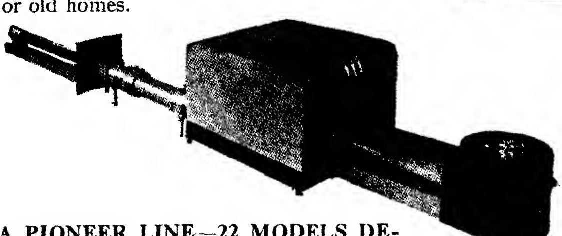
Six sizes ranging from 18 to 200 lb per hour. For homes, apartments, stores, offices, factories, etc. to supply heat and small power requirements.



Industrial Models

Open hopper stokers for heavy-duty heat or power requirements of schools, apartments, hotels, factories, etc. Rugged, long-life mechanism materially reduces fuel bills.





A PIONEER LINE—22 MODELS DELIVERING 10 TO 1250 LB PER HOUR

Developed by Cotta Transmission Corp. —long and favorably known for "precision" transmissions in the heavy automotive and oil industries—Econ-O-Col Stokers are designed and made to give exceptional performance over a long period of years with little or no upkeep costs. The line is complete—a model for every purpose from the heating requirements of a bungalow up to heavy-duty power requirements of a modern factory. Equipped with nationally-known motor and controls. Adaptable to steam, vacuum, water, or warm air systems. A stoker that gives much more for a little more, rather than much less for a little less—Econ-O-Col!

YOU GET ALL THESE FEATURES

Continuous-feed, Free-rolling, Automotive-type Transmission—Made to exacting tolerances from extra-heavy, long-wearing materials. Triple heat-treated gears cut from electric furnace, chrome-nickel steel—diamond point tested. Ball and roller bearings.

Motor—Spring or rubber mounted. Quiet.

Aero-Dynamic Fan—Delivers double required capacity. No whine.

Copper-Bearing Steel—Used in all parts contacting coal to resist corrosion.

Straight Flight Feed Screw—Requires less power, saves electricity.

"Marvel" Air Volume Control—Really works—not just a damper.

Retort—Thick, strong. Keeps coal out of air chamber and scientifically distributes air.

Electric Safety "Shear Pin" Switch—Shuts off both motor and fan when obstruction stops flow of coal.

Write for Special Bulletins Giving Complete Specifications on other Models

Iron Fireman Manufacturing Company

Automatic Coal Burners

Portland, Oregon

Factories: Portland, Ore.; Cleveland, Ohio; Toronto, Canada

Retail Branches or Subsidiaries

Chicago, Ill.; Milwaukee, Wis.; St. Louis, Mo.; New York, N. Y.; Brooklyn, N. Y.; Montreal, Can.

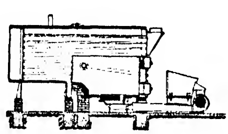
Dealers in Principal Cities and Towns in the United States and Canada

Representation in numerous foreign countries

IRON FIREMAN Automatic Coal Burners

"Forced Underfiring" Principle—Iron Firemen "Forced Underfiring" is based on the scientific principle of feeding fuel to the fire from below, under forced draft. From the conveyor screw coal enters the firebox under the fire and is gradually forced upward into the flame. As the coal approaches the fire, it is gradually heated. The volatile gases are distilled off in the presence of an excess of oxygen and are thoroughly ignited while passing through the incandescent fuel bed. This insures complete combustion. The ash is fused into clinkers which are easily removed.

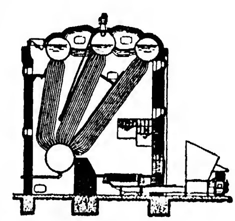
Advantages—Iron Fireman saves money and increases heating plant efficiency in four major ways: (1) Cuts fuel costs; (2) Reduces labor costs; (3) Provides steady, even heat or power; (4) Eliminates the smoke nuisance.



Typical Installation Down Draft Firebox Boiler

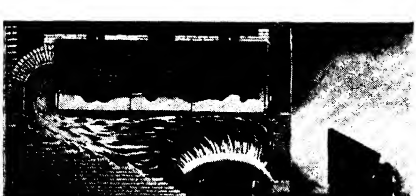
Installation and Sizes—Iron Fireman is made in a range of hopper and bin-feed sizes for commercial heating and power boilers and also for homes. It can be installed quickly in practically any solid fuel boiler or furnace, old or new. Machines are shipped complete from the factory. All parts are standard and interchangeable.

Features of Design and Construction—Construction and operation of the Iron Fireman are characterized by simplicity throughout. Outstanding features of design and construction are: (1) Pressed steel construction. (2) Special patented transmission—three speeds and neutral. Gears run in bath of oil. (3) Electric motor—standard make. (4) V-belt drive. (5) Safety shear pin protects mechanism from damage. (6) Quiet ball bearing fan supplying forced draft to fire. (7) Automatic fire banking damper—conserves fuel and holds fire in proper condition when stoker is idle. (8) Positive pneumatic fume eliminator—an auxiliary air supply that insures positive movement of all gases through the fire. (9) Volumeter that supplies exactly the amount of air needed for perfect combustion regardless of fuel bed conditions or type of coal used. (10) Sectional retort especially designed to allow for heat expansion. (11) Dead plates of heavy iron and ribbed. (12) Sectional, self-cleaning tuyere blocks. (13) Conveyor screw cast of special Iron Fireman alloy steel from one-piece pattern. (14) Automatic electric controls designed for and used exclusively on Iron Fireman.

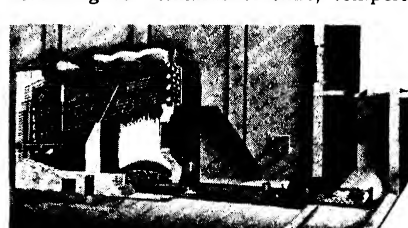


Typical Installation Four Drum Water Tube Boiler

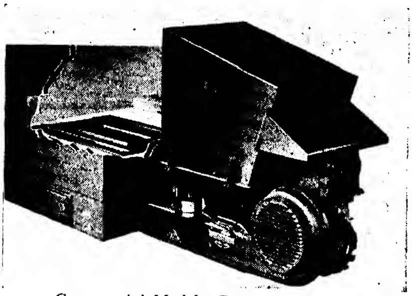
Automatic Controls—Iron Fireman starts and stops at the command of sensitive, accurate automatic controls. Directing controls govern stoker operation according to demands of time, tempera-



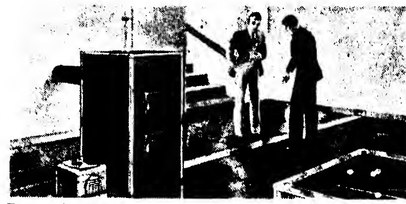
Iron Fireman in Operation in Horizontal Return Tubular Boiler, Low Bridge Wall



Commercial Installation—Coal Flow model that carries coal direct from bunker to fire



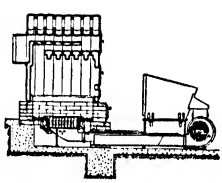
Commercial Model—For Heat or Power



Domestic Installation—Coal Flow model that carries coal direct from bin to fire

ture, or pressure. An example of the efficiency of Iron Fireman directing controls is the "Syncro-Stat" which provides automatic control of day and night temperature. Other directing controls include pressure regulators, hot water and furnace regulators, and the "Timetactor," a device which runs the Iron Fireman during predetermined intervals in order to keep the fire alive during mild weather.

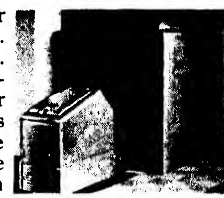
The most important unit of the operating control system is the motor-driven relay switch. This device starts and stops the stoker motor at the command of the Syncro-Stat or other directing controls. In the case of the larger stokers a magnetic operating switch works in conjunction with the relay switch.



Typical Installation Cast Iron Boiler

Iron Fireman for Homes—The Iron Fireman residential model employs "Forced Underfiring" principle the same as larger machines, with simplified operation.

Can be recommended for any steam, hot water, vacuum, or warm air furnace. Quickly installed. Hopper and bin-feed models for both bituminous and anthracite coal. Anthracite models have been tested and approved by The Anthracite Institute.



Typical Installation Domestic Furnace

ENGINEERING SERVICE

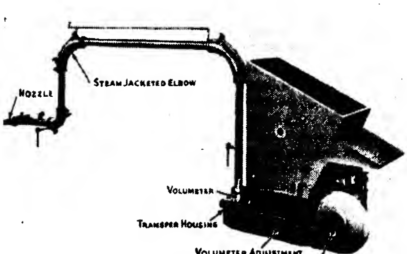
The Iron Fireman organization is nationwide. Trained men—backed by one of the largest manufacturing organizations in the field—are at your service to help you with the experience and practical heating information gained through servicing thousands of boiler rooms and heating plants in all parts of the country.

Any Iron Fireman engineer will gladly call and submit any additional information requested.

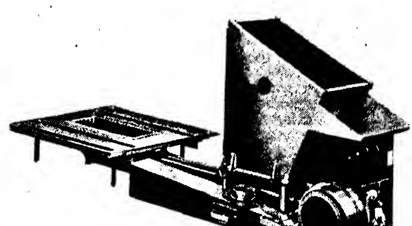
CATALOG AND INFORMATION

Catalogs give full information about the Iron Fireman. Descriptive folders give special data about installation in particular types of industries and in homes.

Secure them by addressing the factory or any Iron Fireman representative.



Air Spreader Model—For High Pressure Boilers



Industrial "Poweram" Model—For Heat or Power

Motorstokor Division
Hershey Machine & Foundry Co.

Factory and Home Office

Manheim, Pa.

MOTOR STOKOR

Installation and Service by factory-trained dealers in all anthracite burning areas.

DEFINITION

A complete stoker, burner, and optional ash-removal system for automatic combustion of buckwheat or rice anthracite. Applicable to coal, gas or oil furnaces or boilers for providing warm air, hot water, or steam. Especially designed for automatically heating buildings and providing year-round hot water.

RANGE OF TYPES

Standard installations in all sizes include direct-from-bin feed with ash removal, direct-from-bin feed with pit collection, hopper feed with ash removal, and hopper feed with pit collection of ashes.

RANGE OF SIZES

Available in 9 models, providing a complete range. The smallest is the new domestic MOTORSTOKOR No. 10 capable of heating average small homes. It feeds up to 20 lb of anthracite per hour, is rated at 640 sq ft of steam and 1020 sq ft of water radiation. The largest are MOTORSTOKORS No. 2 and No. 3, feeding up to 100 lb of coal per hour and rated at 2800 sq ft of steam or 4480 sq ft of water radiation.

ADVANTAGES

Simplicity: Entire mechanism functions intermittently, including coal feed, controlled draft, and ash removal. A portion of the combustion air is fed with the coal, preventing dust and back draft. Worm feed. Concentric dustless ring burner needs no cleaning. Revolving bar breaks all clinkers.

Efficiency: Fixed air mixture for uniform combustion. Floating worm assures uniform, trouble-free coal feed. Flexibly mounted motor operating intermittently and using little current. Quiet, direct-mounted, self-compensating fan.

Ruggedness: Heavy cast parts, with lavish use of chrome-moly, monel, nickel, and special alloys. Oil-submerged gears reduce all operations to very slow wear-free motions. Extraordinary structural and metallurgical protections against corrosion.

Safety: Floating coal screw minimizes stoppage or jams. Air feed through coal prevents back draft and escaping gas. Automatic release-clutch cuts off current when over-loaded. Minneapolis-Honeywell controls.

The new MOTORSTOKOR No. 10 marks its manufacturer's new low in the first cost of completely automatic anthracite equipment.

MOTORSTOKOR 2AF for heavy-duty service in apartments, office buildings, etc. Bin-feed pipe at left. Ash removal system at right.

Schwitzer-Cummins Company

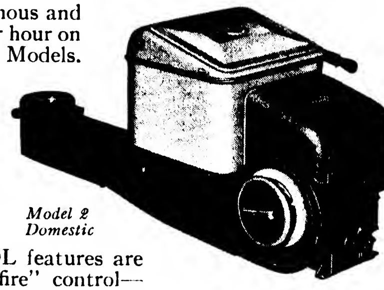
AUTOMATIC COAL STOKERS

Indianapolis, Indiana, U. S. A.

Dealers in principal cities and towns in the United States and Canada and certain foreign countries.

STOKOL

61 STOKOL MODELS for burning bituminous and anthracite coal—capacities range from 12 lb per hour on domestic sizes to 600 lb per hour on commercial Models. Both bituminous and anthracite Models are made in hopper type and bin feed construction. All STOKOLS are "underfeed" type.

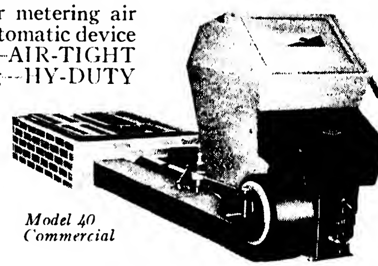


Model 2 Domestic

SUPERIOR FEATURES

Schwitzer-Cummins Company pioneered many basic improvements important to the success of small Stokers—the Hydraulic Transmission, Automatic Air Control, and Universal Bin Feed are outstanding. Superior STOKOL features are STOKOSTAT, hydraulically operated "Holdfire" control—STOKOL AUTOMATIC AIR CONTROL for metering air delivered by the blower—STOKOLARM, an automatic device for signaling notice of obstruction in feed screw—AIR-TIGHT HOPPER with low door for convenient filling—HY-DUTY multi-blade blower fan—AUTOMATIC ASH REMOVER for anthracite models.

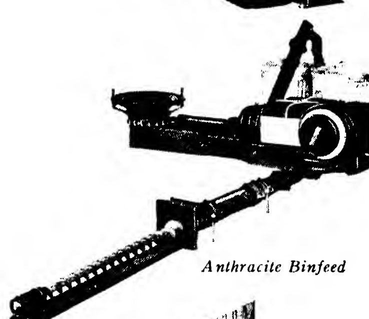
An unusual engineering feature is the STOKOL Hydraulic Transmission—An oil pump located on the fan and drive pulley shaft, draws oil from the reservoir in the bottom of the case and forces it into the hydraulic cylinder. The pressure of oil moves the piston forward and with it a lever which turns a ratchet wheel attached to a main shaft which drives the coal feed screw. Turning a simple valve varies the rate of oil flow from the cylinder, and gives an unlimited number of coal feeds. Oil used to operate the piston is diverted to flood every moving part, assuring perfect lubrication and long life.



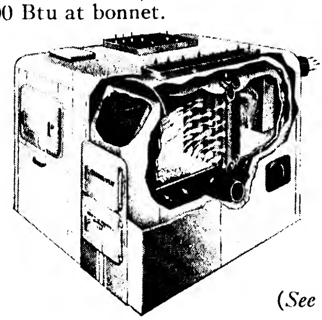
Model 40 Commercial

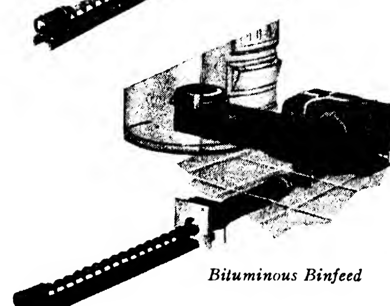
STOKOL-HEAT

Stoker-fired Furnace or Winter Air Conditioner, embodying a welded steel furnace—automatic coal stoker, either hopper type or binfeed—Blower-filter—and Humidifier. Capacities from 75,000 Btu at bonnet to 300,000 Btu at bonnet.



Anthracite Binfeed





Bituminous Binfeed

(*See also Page 894*)

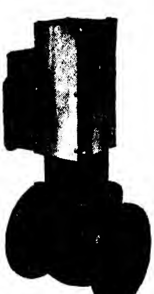
Barber-Colman Company

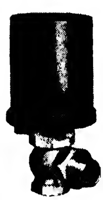
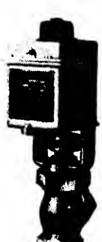
Rockford, Illinois

AUTOMATIC *Electric* CONTROLS








Barber-Colman Controls are all electric. Precision built to insure long, continuous, and dependable service. Easy to install in either new or existing installations. Ready for instant service at all times, even after long shut down periods.

Thermostats. All types—room, duct, immersion and air-stream. For snap-action, floating and proportioning controls.

Hygrostats. Room, and duct types.

Motor-Operated Valves. Packless, packed, single seat, pilot piston, vee-ported, balanced, three-way, four-way, and butterfly. For shut-off, throttling and proportioning service.

Solenoid Valves. For air, oil, water, gas, and refrigerants.

Damper Control Motors. Unidirectional, or reversible, fixed or adjustable speed. For positive and proportioning service.

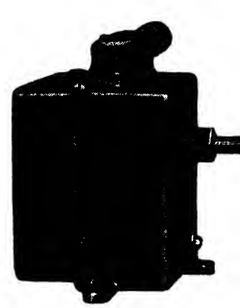
Program Switches. Automatic contact-making mechanisms for multi-compressor control or similar applications.

Micro Controls give accurate proportioning control of modulating valves or dampers by operating them rapidly to a definite position for each different temperature at the thermostat.

Literature is available describing complete automatic control of heating, ventilating, and air conditioning systems. Consult a Barber-Colman representative, or write the factory.

Listed as standard by Underwriters' Laboratories.





Barber-Colman Company

Rockford, Illinois

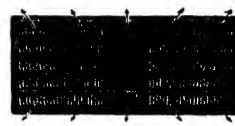
GRILLES **Uni-Flo** **REGISTERS**

Uni-Flo grilles and registers are designed especially for air conditioning installations involving both heating and cooling.

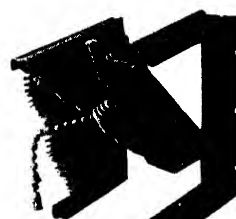
Dimensions and **core arrangement** may be selected to give desired directional flow and throw without increasing the noise level or causing drafts. Various sizes and shapes are available including curved surfaces.

Registers are same construction as grilles, with the addition of spring loaded, positive closing, chain or key-operated dampers.

Finish: Plain metal, gray prime coat, clear lacquer, or any of the following electroplated finishes: Gun-metal, brushed bronze, plain zinc, buffed zinc, brushed zinc, and satin copper.



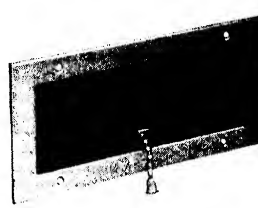
Uni-Flo Grille



Uni-Flo Register

UNI-FIN

Uni-Fin grilles and registers are designed especially for residential warm air installations. Available in standard sizes and prime coat or electroplated finishes.



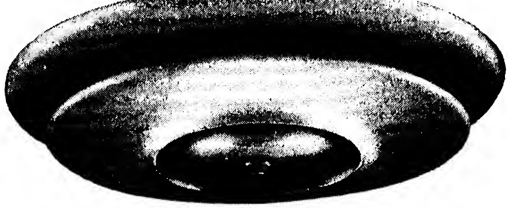
Uni-Fin Register

VENTURI-FLO

Venturi-Flo is a ceiling outlet of modern design, attractive in appearance, available for a wide range of capacities.

Write for descriptive literature on all types of outlets and accessories.

Ceiling Grille



Venturi-Flo

Uni-Flo-Lite

Detroit Lubricator Company

Detroit, Michigan, U. S. A.

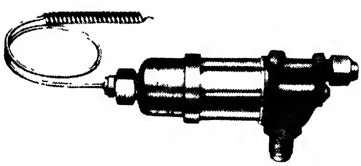
NEW YORK, N Y., 40 West 40th Street

CHICAGO, ILL, 816 S Michigan Avenue   LOS ANGELES, CALIF, 320 Crocker Street

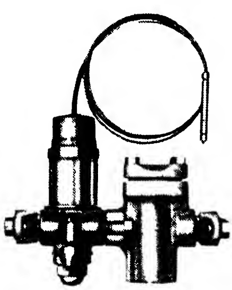
Canadian Representative:
RAILWAY AND ENGINEERING SPECIALTIES LIMITED, Montreal, Toronto, Winnipeg

Division of American Radiator & Standard Sanitary Corporation

Detroit Thermostatic Expansion Valve No. 673



Detroit valves are scientifically designed to keep evaporators completely refrigerated under all conditions. Orifice sizes available from 1/32 in to 7/32 in with capacities up to 3½ tons on Dichlorodifluoromethane or 6 tons on Methyl or Sulphur.



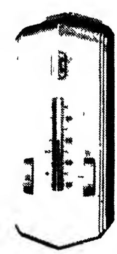
Detroit Thermostatic Expansion Valves Nos. 781-783 and 785

Large capacity valves for air conditioning installations. Capacities up to 20 tons on Dichlorodifluoromethane and 35 tons on Methyl. Line Strainer illustrated available for large valves.

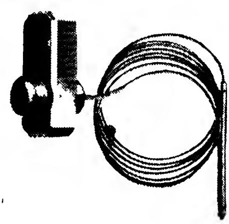
Pressure Control (Model RB-3)

Controls low side pressure. Available with high pressure cut-out to protect against high head pressures. Also available to control temperatures.

Two-Eleven Room Thermostat

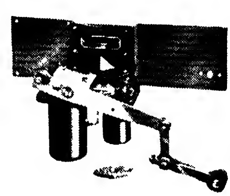


A thermostat neat in appearance and of new design for heating, cooling or in combination for both heating and cooling Supplied with or without adjustable compensator providing uniform control.



Differential Thermostat No. 691

An inexpensive room thermostat for room cooling which modifies indoor temperature in accordance with outdoor temperature to maintain comfort conditions Provides economy of operation and prevents shock due to over-cooling.



Duct Damper Motor No. 431

An inexpensive means of providing individual temperature control for zones or groups of rooms. Is quiet and can be mounted directly on the duct. Furnished with auxiliary switch to control heating equipment. Neat in appearance and easily installed.

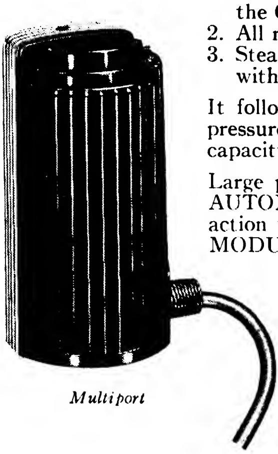
Other Controls

This Company can also supply you with Refrigeration Solenoid Valves, a full line of Boiler and Furnace Limit Controls—Room Thermostats both high and low voltage for both heating and cooling—Fan controls—Humidity and Stoker Controls and Solenoid Valves for the control of water, fuel and gas.

NEW IDEAL FAST-VENTING SYSTEM

For All One Pipe Steam Jobs — New and Old

There are three fundamental requirements for the venting of one pipe steam jobs—particularly when automatically fired.



Multiport

1. Venting must proceed rapidly and be accomplished as early in the ON period of the burner as possible.
2. All radiators must start venting simultaneously.
3. Steam flow to each radiator must be regulated in accordance with its location and capacity.

It follows that most of the venting must be done before boiler pressure increases beyond a few ounces—also that large port large capacity valves are thus necessary.

Large port low pressure venting has been made possible by the AUTOMATIC MODULATOR (patented). This permits full port action up to a few oz. pressure. Beyond that the AUTOMATIC MODULATOR reduces port area so as not to jeopardize subsequent venting at normal pressures.

The No. 300 Arco-Detroit Multiport for radiators, is ideal for automatic heat as its construction permits much faster venting than the average air valve.

The No. 861 Arco-Detroit Hurivent for mains, which has a ½ in. port area, vents with amazing rapidity and cannot be compared with the ordinary vent valve in performance.

Hurivent

The No. 300 Multiport is adjustable over a wide range and its adjustment is proportional to the movement of the adjusting arm. Thus any system can be quickly and accurately balanced, and when desired, the adjustment can be locked to prevent unauthorized tampering. Adjustment in no way interferes with the action of the AUTOMATIC MODULATOR.

The No. 300 Multiport is new, modern, and very attractive—has a lustrous black molded jacket accentuated by brilliant chromium striping.

Of great importance to the installer when necessary to "balance" the system, is the scientifically designed slide valve method of adjustment. The movement of this adjusting lever, which is located at the top of the valve, effects a corresponding adjustment of port area, and this adjustment is in direct proportion to the distance through which the lever moves. Moving the lever over half its arc increases or decreases port area just 50%—moving the lever through a quarter of the arc changes port area 25%.

303
Straight Shank
Multiport

Two straight shank valves designated as the 302 and 303 Multiports are furnished for concealed or "convector" type radiators. They have the same construction as the No. 300, except for the shank and all metal jacket. The No. 302 has a ⅛ in. connection and the No. 303 a ¾ in. connection.

Experience with one pipe steam jobs has shown that such troubles as uneven heating, hard-to-heat rooms, etc., are fundamentally due to inadequate venting. Putting the No. 300 Multiport on each radiator and one or more No. 861 Hurivents on the main, eliminates all this and also improves fuel economy. Whenever a conversion burner or a new boiler is installed, all radiator and vent valves should be replaced to avoid complaints and dissatisfaction.

The Fulton Sylphon Company

Manufacturers of Sylphon Automatic Temperature Controlling Instruments and Packless Expansion Joints

Knoxville, Tenn.

Sales Representatives in Principal Cities

PRODUCTS

Automatic Radiator Valves, Air Conditioning Controls, Temperature Regulators for Storage Water Heaters; Thermostatic Water Mixers; Refrigeration Temperature Regulators; Packless Expansion Joints; Pressure Reducing Valves.

GENERAL INFORMATION

Fulton Sylphon Products depend upon the famous Sylphon Metal Bellows for their trouble-free service and long life

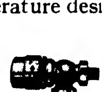
Over thirty-five years ago, THE FULTON SYLPHON COMPANY originated this bellows . . . a precision-built, seamless, jointless "miracle in metal."

By continuous engineering study and intimate contact with heating and refrigeration problems, this company has used this efficient, practically indestructible bellows in the development of a line of temperature control and heating specialties known everywhere for outstanding service and quality.

SPACE HEATING AND AIR CONDITIONING CONTROL

No. 885 Automatic Radiator Valve

For exposed radiation. Small, neat, finely finished, adjustable to room temperature desired Simply replace ordinary radiator valves with these Sylphon Automatic Regulators—no wiring, piping or auxiliary equipment are required. These valves answer the demand for an inexpensive means of providing accurate, dependable space temperature control in rooms, sections or throughout large buildings, new or old. Similar type valves for concealed radiation—get Bulletin HVG-80.



Sylphon No 885 Automatic Radiator Valve

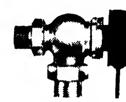
Sylphon Thermostats

Heat accelerated and on-off types, with or without night set-back feature and with or without anticipating feature.

Sylphon Thermostats (4 types)

No. 890 Electric Radiator Control Valve

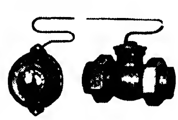
For either exposed or concealed radiation Similar in appearance and action to Sylphon Automatic Valves, but operated by an electric wall thermostat The closing of the thermostat circuit energizes a low voltage electric heater coil surrounding a bulb containing a volatile liquid. This liquid expansion causes pressure on a bellows in the valve head operating the valve. This provides radiator valve control from a remote location, permits regulation of several radiators from a single thermostat, enables a time switch to be installed, if desired, offers effective zone control of large areas at a fraction of the cost of conventional motor-operated valve systems. Bulletin HVG-70



Sylphon No 890 Electric Control Valve

No. 7 Temperature Control

A self-contained, self-powered regulator for controlling unit heaters, wall or ceiling type radiators, heating coils in duct-type heating systems, etc. A sturdy control, quickly installed, holds temperatures within close limits Valve is placed in steam line to one or a battery of heaters, thermostat is mounted on wall or column. Designed for use on regular heating pressures up to 15 lb. Similar regulators, Nos. 7-2 and 7-3 for 50 and 75 lb pressure and temperatures up to 170 F. Bulletin HVG-50.



Sylphon No 7 Temperature Control (Self-operating)

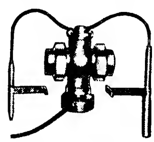
No. 928-C Temperature Regulator

A compact regulator, especially suited to control of duct temperatures. Bulb is a series of copper coils, sensitive to slightest change in air temperature. Convenient adjustment. Three types for 15, 50 and 75 lb steam pressures and temperatures not exceeding 170 F.

Sylphon No 928-C Duct Temperature Regulator

The Fulton Sylphon Company

No. 889-E Unit Ventilator Control



Sylphon No. 889-E Unit Ventilator Control (Electrically operated type)

An electrically and mechanically operated dual valve developed for use in unit ventilators. The electric side of the valve utilizes the same "heat motor" as used in Sylphon Electric Radiator Valves and is operated by a wall thermostat in the room. Valve operates to bring room temperature to desired level during heating up period, and maintains desired temperature during period of "thermal balance." Mechanical side of valve is operated by a thermostat bulb placed in discharge air stream and is known as "minimum air stream thermostat."

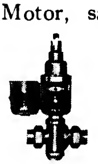
This modulates the valve, when the electric thermostat is "off," to prevent discharged air from falling below a predetermined minimum temperature, eliminating objectionable cold drafts. Suitable for steam pressures up to 15 lb and for any two-pipe steam, vapor or vacuum system.

No. 371 Damper Motor

Sylphon Damper Motors (Self-operating and electric types)

A positive type motor for on-and-off control of dampers. Operation may be controlled by thermostat, hand switch, motor starting switch, or other means. Motor, safety type, closes on current failure. Write for literature.

Refrigeration Controls



No. 945-Z Regulator Showing detail of "freeze-proof" valve

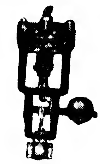
Adaptable wherever brine is used as the refrigerant. Latest development is a "freeze-proof" valve (illustrated at left on the popular Sylphon No. 945-Z Regulator). Bulletin HVG-20.

PACKLESS EXPANSION JOINTS

No. 110 Sylphon Expansion Joint

The Sylphon Packless Expansion Joint eliminates useless building height, expensive construction and non-revenue producing space. No costly leaks and repairs, no repacking, always tight, allows heating system to operate at full efficiency. Write for Bulletin HVG-140.

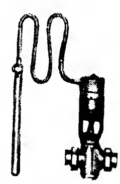
PRESSURE REGULATORS



No. 955 Interlocking Valve

No. 955—Sylphon Interlocking Valve is a safety appliance protecting oil-fired furnaces by shutting off oil flow should atomizing pressure fail or fall below the required minimum. Approved by Associated Factory Mutual Fire Insurance Companies. Bulletin HVG-100.

HOT WATER SUPPLY
No. 923 Temperature Regulator



No. 923 Temperature Regulator

For controlling temperature of water in heaters, open or closed tanks and various types of equipment. Operation is unaffected by temperature fluctuations at the valve, either above or below bulb temperature. Neat. Compact. All parts, except steel adjustment spring, made of non-ferrous metals. May be installed in any position. Ranges from 40 −80 F to 290 −330 F. Bulletin HVG-20.

Below No. 902 Sylphon Thermostatic Water Mixer—14 to 131 gpm depending on water pressure.

Above No. 905 Thermostatic shower mixer with temperature selector handle permitting setting for any desired, thermostatically maintained, temperature from cold to a safe maximum temperature of hot water.

Sylphon Thermostatic Water Mixers

Utilize hot water from any storage tank or instantaneous heater, and effectively regulate the amount of cold water required to temper it to the desired degree, actually mixing the hot and cold water together before delivery. Temperature remains constant in spite of fluctuations in supply water temperatures or pressures.

Four sizes with capacities ranging from 5 to 131 gpm. Bulletin HVG-40.

Julien P. Friez & Sons

(Division of Bendix Aviation Corporation)

Baltimore **Maryland**
Established **in 1876**

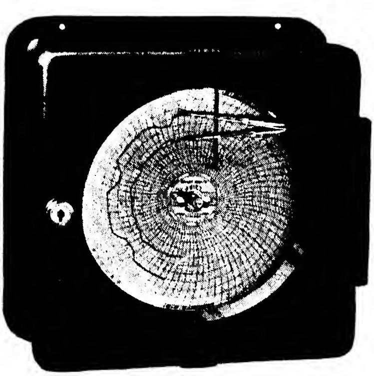
Manufacturers of a Complete Line of Automatic Electric Controls for Industrial and Comfort Applications. Also a Complete Range of Recording and Accurate Measuring Instruments for Indoor and Outdoor Applications

Humidistat—accurate over long periods and complete range; double length human hair element. **Bulletin A.**

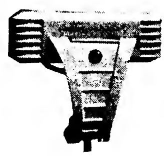
Thermostat—sensitive, accurate for highest grade work. **Bulletin T.**

Comfortrol—effective temperature Thermostat resetting itself as prevailing humidity varies, using human hair compensating element. *Exclusive.* **Bulletin E.**

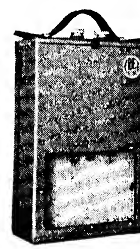
Hand Aspirated Psychrometer—replacing 'slings', no whirling; reliable, immediate reading; thermometers perfectly ventilated by typical air, induced by venturi action with hand operated bellows. *Exclusive.* **Bulletin S.**



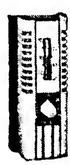
Remote Reading Temperature and Humidity Recorder—Electrically operated; humidity uniquely recorded from distant location directly in percent relative. *Exclusive.* **Bulletin R.**



Windowstat—placed at window indoors, positively preventing condensation from excess humidity. Controls humidity supply just below critical point as outdoor temperature varies. *Exclusive.* **Bulletin W.**

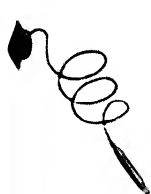


Portable-Recorder—for surveys or tests of humidity and temperature. Inked records on charts the size of filing cards (3″ x 5″). *Exclusive.* **Bulletin G.**



Microstat—Small sized thermostat, featuring powerful Alnico magnets. Though priced with the lowest, unsurpassed for accuracy and fine appearance. **Bulletin TM.**

Magnetic Gas Valve—New principle, free floating disc, no diaphragm; quiet, durable; range of sizes; low priced, low voltage, especially suited for control by Microstat pictured above. *Exclusive.* **Bulletin VG.**



NEW!
Hydraulic Action

Limit controls for fans, furnaces, hot water, ovens, refrigeration. A new line of controls for long reliable service. **Bulletin LC** and Data Sheet 225.

Write for Bulletins

MODERN ADVANCED CONTROLS FOR MODERN NEEDS

The Mercoid Corporation

SOLE MANUFACTURERS OF THE MERCOID SWITCH

Main Office and Factory, 4201 Belmont Ave., Chicago, Ill.

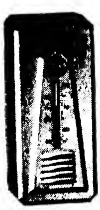
Branch Offices:	NEW YORK, N. Y. 330 W. 34th St.	PHILADELPHIA, PA. 3137 N. Broad St.	BOSTON, MASS. 25 Ivy St.

Distributors and Jobbers in all Principal Cities

COMPLETE LINE OF AUTOMATIC CONTROLS AND MAGNETIC VALVES FOR HEATING AND AIR CONDITIONING

Mercoid Controls are equipped throughout with sealed mercury contact switches. These switches cannot be affected by dust, dirt or corrosive gases. They are not subject to open arcing, pitting or sticking of contacts. Mercoid switches will operate indefinitely without deterioration. Write for catalog No. 200AS.

SENSATHERM



Extremely sensitive thermostat which requires no artificial stimulation to maintain an even room temperature. Operates on temperature variation of ½ deg above or below point set (total differential 1 F). Small in size, neat in appearance and unfailing in performance.

TRANSFORMER-RELAY

A reliable low voltage mercury contact relay which also acts as a transformer inducing low voltage (24 volts) on the pilot circuit. Does away with all hum and chatter. Available for 110 or 220 volts, 60, 50, or 25 cycle.

PRESSURE AND TEMPERATURE LIMIT CONTROLS

These instruments have proven their reliability over a long period of years. The outside double adjustment provided with a calibrated dial, is a special feature that saves considerable time when making the necessary operating adjustments. Available for steam, hot water and warm air furnaces. These controls are also used for various industrial applications.

COMBINED PRESSURE AND LOW WATER CONTROL

Type DA-121 low water and pressure controls have the new double adjustment feature which saves guess work and time in installation. Prevents firing into dry boiler and building up excessive steam pressure. Various ranges available.

SAFETY CONTROLS

"K" line controls have a number of desirable features which make them pre-eminent in the field. KMI illustrated herewith, is for burners employing intermittent ignition. Other types available. These controls offer positive protection against flame or ignition failure.

STOKER CONTROLS

A stoker fire-maintaining timer which eliminates overheating and waste of fuel. Has interlocking mechanism which prevents timer from operating immediately after thermostat shuts off. All adjustments are easily made without the use of tools. Write for bulletin No. 124 AS.

Johnson Service Company

AUTOMATIC TEMPERATURE AND AIR CONDITIONING CONTROL

General Offices and Factory

Milwaukee, Wis.

Branch Offices in all Large Cities

JOHNSON TEMPERATURE REGULATING CO. OF CANADA, LTD., 113 SIMCOE ST, TORONTO, ONT.
MONTREAL, QUE. WINNIPEG, MAN. CALGARY, ALTA VANCOUVER, B C.

Products and Services

Manufacturers, engineers, and contractors for Automatic Temperature and Humidity Control Systems applied to all types of *heating, cooling, ventilating, and air conditioning installations.* Also, for every range required in *manufacturing and industrial processes.* A single nation-wide organization devoted to Design, Manufacture and Installation for more than 50 years.

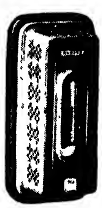
Room temperature control applied to radiators, unit ventilators, and heat delivery ducts. *Johnson "Duo-Stats"* to maintain the proper relationship between outdoor and radiator temperatures for groups of radiators or "heating zones."

A complete line of devices for automatic control of *air conditioning systems*, heating, cooling, humidifying, dehumidifying. Automatic seasonal shifting of control cycles.

Periodic Flush Systems for intermittent flushing in various sections of a building, reducing load on piping system and insuring economy in use of water.

Special bulletins and catalogues on request. Sales Engineers at direct branch offices in all principal cities.

Johnson All-Metal Thermostats



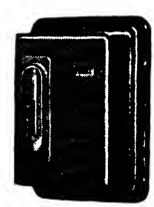
Room Thermostat

Johnson thermostats, room, insertion, and immersion types, are all-metal throughout, no soft or hard rubber parts to deteriorate and become inoperative Every thermostat precise in construction and thoroughly tested for accuracy, efficiency, and durability.

The Johnson *intermediate action thermostat* gives a true graduated action to mixing dampers and valves. It holds them in an intermediate position to maintain the temperature of the room accurately within one degree above or below the setting of the thermostat, if desired.

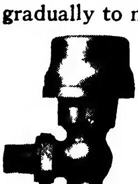
Johnson "Dual" or Two-Temperature Thermostats

The *Dual, two-temperature,* room thermostat especially adapted for use where various rooms or groups of rooms are occupied when the remainder of the building is not in use. Separate steam mains avoided. The shifting from "day" or occupancy temperature to an economy temperature for "non-occupancy" conditions, accomplished by a switch or Johnson program clock at a central point. Push buttons on the thermostat are provided when "occupant control" is desired. Dual thermostats are all-metal and operate valves and dampers gradually to maintain temperature accurately within one degree.



Dual Thermostat

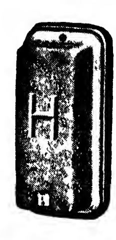
Johnson Valves



"Sylphon" Radiator Valve

Johnson *diaphragm valves* are simple and rugged. Seamless metal bellows and heavy spring operate the valve stem. No complicated moving parts. Made in all standard sizes and patterns. Direct acting (normally open) or reverse acting (normally closed). Three-way mixing and three-way bypass valves.

For steam, water, brine, and special service. Johnson valves are available, if desired, with diaphragms of special moulded rubber. super-aged and heat-resistant. Pilot operated valves for "smooth" gradual control, independent of fluctuating pressure and friction.



Room Humidostat

Four-Point Insertion Thermostat

Humidostats and Humidifiers

The *Johnson Humidostat* automatically controls the supply of moisture delivered to the air by a humidifier or air washer and maintains a constant percentage of relative humidity. Available in both room and insertion patterns, and with various types of elements as determined by requirements, controlling within 1 per cent at relative humidity of 95 per cent and 100 F if desired.

Johnson humidifiers are furnished in steam "grid" type or pan type with copper evaporating pan, brass heating coil, and float control.

Air Conditioning Control

Summer-Winter room thermostats for operation of valves and dampers in reverse sequence for cooling and heating. *Insertion* and *immersion* thermostats in one, two, three, and four-point patterns for operating valves and dampers successively at different temperatures.

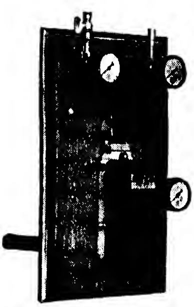
Remote readjustable thermostats, reset from a distant point by pilot or differential thermostat or by pressure switch. *Differential thermostats* to maintain desired temperature differences between two points, such as outdoors and treated space. *Solenoid air switches, manual switches, static pressure regulators, velocity regulators*, and all types of *dampers* to regulate flow of air in ducts.

The *Johnson sensitivity adjustment* is an important development in the field of automatic temperature and humidity control for air conditioning. A unique and convenient means of adjusting the sensitivity of *Johnson* thermostats and humidostats *on the job*, with respect to the capacity of the conditioning apparatus.

Summer-Winter Thermostat

Zone Control

Johnson "Duo-Stats" to regulate the flow of heat in a group of radiators constituting a "heating zone" by maintaining the proper relationship between outdoor and radiator temperatures.

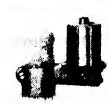


Remote Readjustable Duct Thermostat

Process Control

Calibrated insertion thermostats for controlling temperature of liquids, air and gases. Mercury *extended tube thermostat* for remote location of sensitive element.

Wet-bulb thermostats for close regulation of humidity. "Record-O-Stats," combination instruments to record and control temperatures.

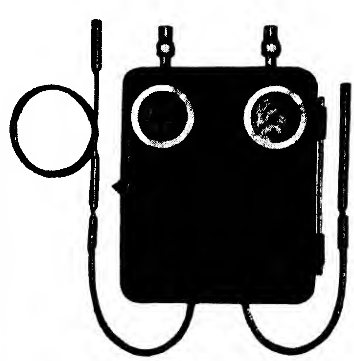


Modulating Attachment for Expansion Valves

Rubber Diaphragm Coil Valve

"Proportioning" Damper Motor

Louvered Damper



Johnson "Duo-Stat"

Minneapolis-Honeywell Regulator Company

Automatic Control Systems for Heating, Ventilating, Air Conditioning

BROWN INSTRUMENTS for Indicating, Recording, Controlling
NATIONAL PNEUMATIC CONTROLS for Heating and Air Conditioning

Factories: MINNEAPOLIS, MINN., PHILADELPHIA, PA., WABASH, IND, CHICAGO, ILL

Branch and Distributing Offices. *Stock carried

Albany, N Y	Columbus	Haverhill, Mass	New Orleans	San Francisco*
Allentown	Dallas	Houston	New York*	Scranton
Atlanta*	Dayton	Indianapolis*	Oklahoma City	Seattle
Baltimore	Denver*	Jackson, Mich	Omaha*	Sioux Falls, S D
Birmingham	Des Moines	Kalamazoo	Peoria	Springfield, Mass
Boston*	Detroit*	Kansas City*	Philadelphia*	Syracuse*
Bridgeport, Conn	East Orange	Los Angeles*	Pittsburgh*	Toledo
Buffalo*	Eau Claire, Wis	Louisville, Ky	Portland*	Tulsa*
Butte*	El Paso, Texas	Mason City, Iowa	Providence*	Wabash, Ind
Charlotte, N C.	Fargo, N D	Milwaukee*	St Louis*	Washington, D C *
Chicago*	Hartford*	Minneapolis— St Paul*	Salt Lake City*	Wichita*
Cincinnati	Harrisburg, Pa		San Antonio*	Worcester, Mass
Cleveland*				Youngstown

In Canada: Montreal, Toronto,* Calgary, Vancouver, London, Winnipeg

In Europe: Amsterdam, Holland,* London, England,* Stockholm, Sweden*

Minneapolis-Honeywell is ready to assume the complete responsibility for the supply and installation of automatic controls and instruments specified for any building that you design. M-H serivce is complete. Through our nationwide organization, we are prepared to make complete installation, supervise installation, provide periodic service or supply control equipment. Minneapolis-Honeywell can offer unbiased advice on your control requirements—manufacture and install complete electric control systems, complete pneumatic control systems, or a combination of the two

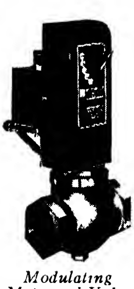
Duct Type Temperature Controller

Each Minneapolis-Honeywell office maintains a factory trained engineering personnel. Your Minneapolis-Honeywell engineer will be glad to furnish you with recommended control layouts and cost estimates.

He is trained to recommend control results before installation of equipment and to produce control results after the installation has been completed

THE MODUTROL SYSTEM OF ELECTRIC CONTROL

The Modutrol System designation is applied to any combination of Minneapolis-Honeywell Automatic Electric or Pneumatic Controls or Self-contained Automatic Valves used to govern the operation of air conditioning or heating systems other than the small domestic installations A wide variety of both modulating and two position motors, controllers and valves are available thus making the Modutrol System extremely flexible as to the selection of control equipment to produce the desired results.



Modulating Motorized Valve

Complete electric control systems are available for those installations where precise, flexible and dependable results are required. Electric controls of the **Modutrol System** provide a dependable means of effecting modulation through the use of the "Series 90" control circuit. All electric motor power units used in this system are completely oil immersed in order to insure quiet operation and years of trouble-free service.

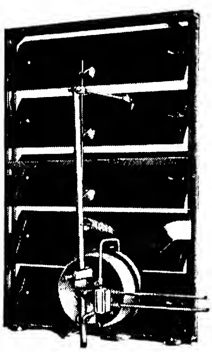
National Pneumatic Thermostat

THE GRADUTROL SYSTEM OF PNEUMATIC CONTROL

Minneapolis-Honeywell offers a complete line of pneumatic controls. To such features as "Helmet Seal" Thermostats and Metaphram construction of valve and damper motors has been added the accurate and infinite positioning of the Gradutrol Relay. For commercial air conditioning and space heating installations, the Gradutrol System offers a truly remarkable advance in pneumatic control.

COMBINATION ELECTRIC AND PNEUMATIC SYSTEMS

The outstanding advantages of both the electric Modutrol System and pneumatic Gradutrol System of control may be combined in a single installation. Thus maximum flexibility and low installation cost are obtained. Minneapolis-Honeywell can offer either an electric or pneumatic system, or a combination of the two. This is your guarantee of an unprejudiced recommendation.

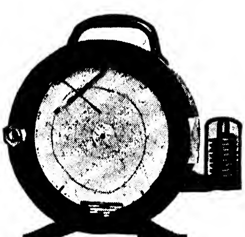


Gradutrol Motor and Damper

BROWN INSTRUMENTS

The extent to which air conditioning equipment is being used in office buildings, theatres, stores, industrial buildings, etc., has opened up a wide demand for indicating and recording resistance thermometers because the temperatures throughout these air conditioning systems should be checked periodically in order to obtain the best results at minimum operating cost. To obtain uniform conditions from modern equipment, it is necessary that the engineer in charge of operation have a visual picture of actual conditions.

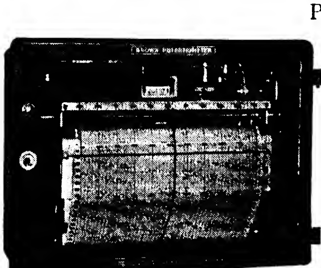
Brown Resistance thermometers are available for indicating, recording, and controlling service and are applicable to all types of air conditioning and space heating installations.



Brown Portable Recorder

In addition to Resistance Thermometers, The M-H Brown Instrument Division manufactures:

Thermometers	Flow Meters
Hygrometers	CO_2 Meters
Pressure Gauges	Tachometers
Vacuum Gauges	Liquid Level Gauges
Potentiometer Pyrometers	Protectoglo System



Brown Recording Resistance Thermometer

RESPONSIBILITY FOR ENTIRE CONTROL SYSTEM

Minneapolis-Honeywell Regulator Co. is equipped to assume the entire responsibility for any control installation, thereby eliminating the difficulties and misunderstandings which division of responsibility may create.

THE POWERS REGULATOR CO.

48 Years of Temperature and Humidity Control

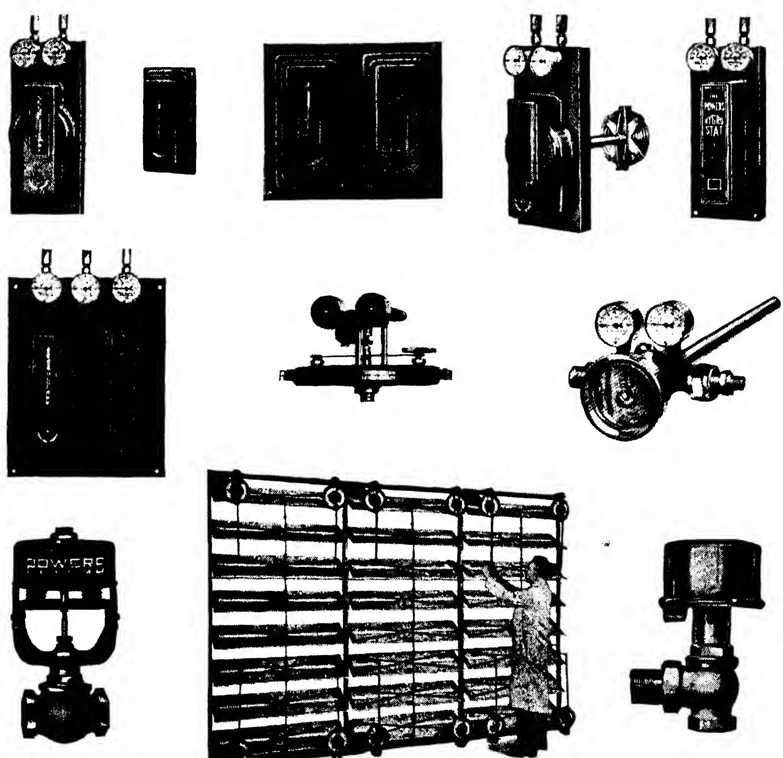
Offices in 47 Cities—See Your Phone Directory

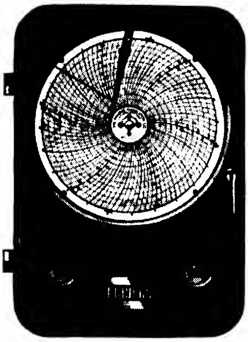
General Offices and Factory: 2719 Greenview Ave., CHICAGO;
General Eastern Office: 231 East 46th Street, NEW YORK;
1808 W. Eighth Street, LOS ANGELES;
The Canadian Powers Regulator Co., 195 Spadina Ave., TORONTO, ONT.

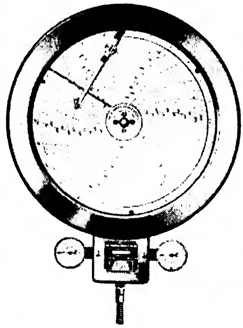
PRODUCTS—A very complete line of compressed air operated and self-operating temperature, humidity and air flow controls for automatically regulating heating, cooling, ventilating and air conditioning systems and industrial processes.

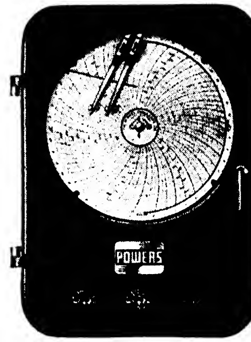
A complete line of self-operating and compressed air operated valves and regulators made for: Controlling steam heated hot water heaters, and submerged type heaters; and for automatically mixing hot and cold water or steam and cold water delivering a mixture at a predetermined temperature. Dial Indicating and Recording Thermometers. Thermometer-Regulators. High pressure steam traps and pressure reducing valves.

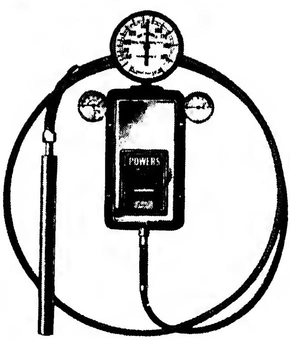
Powers Compressed Air Operated Apparatus

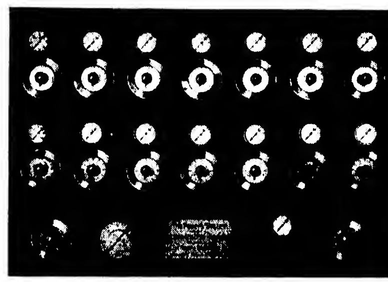




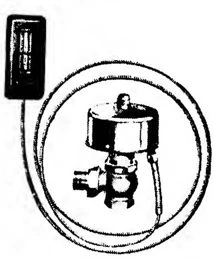








Three of the Many Types of Powers Self-Operating Regulators



POWERS ENGINEERING SERVICE

As the accurate performance of a heating, cooling, ventilating or an air conditioning system, or an industrial process is so dependent upon its automatic control equipment, and as the cost of such control is but a fraction of the entire system, the use of the proper type of regulation is always sound economy.

To secure the maximum return on the investment in automatic control equipment, it is exceedingly important that proper selection of control apparatus be made when each installation is being planned.

Forty-eight years of experience in furnishing and installing temperature and humidity control for every conceivable purpose in all types of buildings has given us a wealth of experience from which you can draw in selecting the proper type of control for any purpose.

CATALOGS AND BULLETINS describing any or all of our products furnished upon request. Phone or write our nearest office. See your phone directory.

Penn Electric Switch Co.

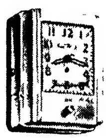
Goshen, Indiana

Offices—NEW YORK, BOSTON, PHILADELPHIA, DETROIT, DAYTON, CHICAGO, MOLINE, (ILL.), ST LOUIS
Export—100 Varick St., NEW YORK CITY

Representatives—GARLAND-AFFOLTER ENGRG CORP., San Francisco, Los Angeles, Seattle, Portland, MONARCH SALES, Denver, FORSLUND PUMP AND MACHINERY CO., Kansas City, VINCENT BRASS AND COPPER CO., INC., Minneapolis

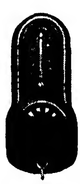
IN CANADA—POWERLITE DEVICES LTD., TORONTO, ONT

Distributors and Jobbers in All Principal Cities



Tem-Clock

Temtrols



Heavy Duty Thermostats

Immersion Temperature Controls

Automatic Controls for Heating, Air Conditioning, Refrigeration, Pumps, Air Compressors.

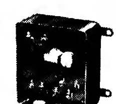
Heating, air conditioning and refrigeration control installations that once seemed complex are made simple to plan . . . easy to install—with Penn Controls. Penn has pioneered many outstanding improvements in temperature and pressure control during the last 20 years . . . many noteworthy contributions to control functions and dependability.

Yet, during this time, Penn also has simplified control constructions, eliminating unnecessary parts . . . selecting new materials and new alloys . . . constantly building to more than satisfy practical engineers and installation men, not just to suit a laboratory technician's whims.

Illustrated are only a few Penn units for oil, stoker and gas heating, industrial temperature regulation, refrigeration, air conditioning and pressure control. Where temperature, pressure, liquid level or humidity control problems are to be solved, consult Penn engineers

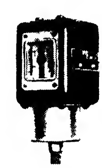
Write for catalog on Penn controls to cover your particular applications, or 'phone the nearest Penn office or representative. Penn engineers always are available for consultation on control problems, without obligation, of course.

Pump and Air Compressor Controls

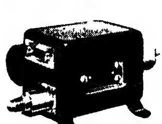


Stoker Timer Relays

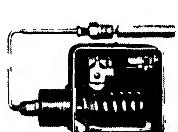
Water Valves and Regulators



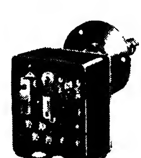
Steam Pressure Controls



Refrigeration Controls



Remote Immersion Temperature Control



Oil Burner Stack Switches

Warm Air Fan and Limit Controls

Penn control engineers have simplified design and production problems for others! Let them assist you.

Spence Engineering Company, Inc.

28 Grant Street, **Walden, N. Y.**

SPENCE METAL DIAPHRAGM "DEAD END" REGULATORS

Advantages of Spence Regulators

Dead-end Shutoff—Spence Regulators are guaranteed to hold a dead-end.

Single Seat—Spence design makes possible a balanced single seat even in large sizes.

Metal Diaphragms—Under normal conditions never require replacement.

Accurate Regulation—Regardless of fluctuations in either load or initial pressure.

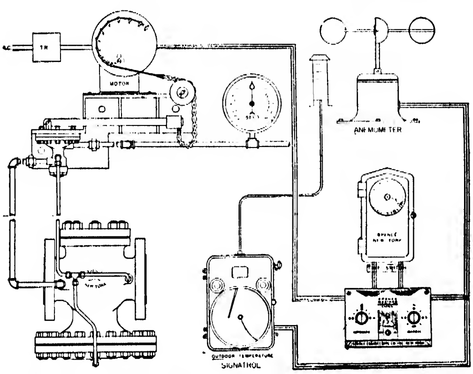
SECO Metal—Guaranteed to resist the wiredrawing action of steam.

Interchangeable Pilots—Any type of pilot will fit any size main valve.

Accessibility—Pilot is connected to main valve with unions.

No Stuffing Boxes—All main valves and most pilots are packless.

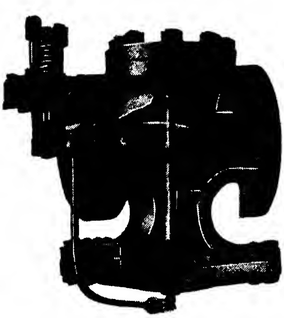
Spence Weather Compensator—Type EWM3T

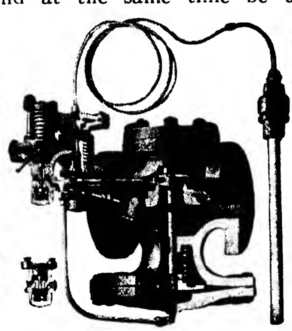


This simple, dependable Control, when installed on a properly designed orificed heating system, will show a substantial degree-day steam saving, at a low maintenance cost.

The delivery pressure of the Regulator is automatically adjusted in direct proportion to the building heat losses. In other words, as the losses become greater, steam pressure on the system is automatically increased.

Any number of zones can be controlled by one automatic Signatrol, automatic Wind Loss Compensator (Anemometer). Time Switch and Master Control Panel equipped with Manual and Automatic Dials for each zone. In this way each zone can be set individually and at the same time be under the Master Control.





Pressure Regulator—Type ED

Designed to regulate a steady or varying initial pressure so as to maintain a constant, adjustable, delivery pressure. Applicable to heating systems, power plant operations, or manufacturing processes.

Combined Temperature and Pressure Regulator —Type ETD

Self-contained, pilot operated, dead-end. Designed to control flow of fluid to a heating or cooling element, so as to maintain a constant, adjustable temperature, and protect the element against excessive pressure.

Electrically Operated Valve—Type EM

Can be opened or closed independently by an electrical switch.

Type ET—Same as ETD except pressure control is omitted.

Order a SPENCE Regulator for 40 days' free trial.

White-Rodgers Electric Company

1209 Cass Avenue, **St. Louis, Mo.**

CHICAGO	NEW YORK	CLEVELAND	CINCINNATI
844 N RUSH ST	245 W 55TH ST	14104 LAKOTA	457 E 6TH
SUPERIOR 1336	CIRCLE-1235	ORCHARD 7542-W	MAIN 4175

Distributors in Principal Cities

The basis of all White-Rodgers line voltage controls is the stainless steel seam-welded diaphragm, and the "solid liquid charge" obtained through a new method of evacuating all traces of air and gas from the liquid. The greater pressure available through the expansion of the liquid charge permits the use of a snap-action switch of unusually sturdy construction. This has resulted in Underwriters' approved ratings of 25 amp 120 v. A-C, 15 amp 240 v. A-C, and 1½ hp 120-240 v. A-C. Send for latest condensed control catalog.

LOW VOLTAGE THERMOSTAT

The appearance of this extremely sensitive instrument permits it to harmonize with the modern American home An accurate thermometer and temperature selector are concealed behind a hinged cover. Anticipating and non-anticipating types are available. Ivory finish with chrome trim

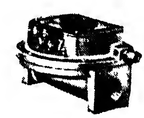
LINE VOLTAGE THERMOSTAT

This "Hydraulic Action" control is ideal for air-conditioning and heating installations in schools, hospitals, stores, theatres, etc. and for use with unit heaters. Due to high rating shown above, several unit heaters may be handled by only one control. Two finishes available—ivory and chrome, and black and chrome

STOKER TIMER

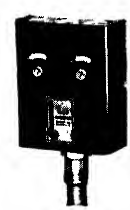
This modern designed stoker control employs a warp switch relay, thereby assuring at all times quiet operation and absence of annoyance due to thermostat chatter and relay hum. Also available with fused line switch and for automatic night set-back.

ELECTRIC GAS VALVE



This quick opening diaphragm valve is actuated by a bi-metal pilot valve to give noiseless operation at all times Available in ¾ in , 1 in , 1¼ in. and 1½ in sizes.

DUAL IMMERSION CONTROL



It is possible to obtain many economical combinations of two switches in a single housing, such as Combination Blower and Limit Controls and Dual Immersion Controls. The latter control (as illustrated) eliminates the need for an extra boiler tapping

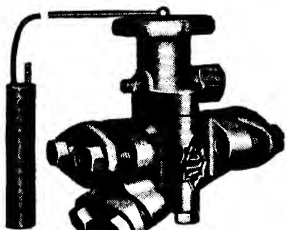
PRESSURE CONTROL

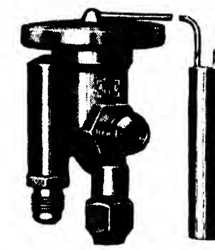
Pressure, warm air and hot water limit controls are all heavy-duty line or low voltage instruments offering positive safety service on steam boilers and warm air and hot water furnaces. The quick response of these controls to rapid changes in temperature or pressure prevents dangerous over-run.

Alco Valve Company, Inc.

ENGINEERED REFRIGERANT CONTROL VALVES

2626 Big Bend Blvd. **St. Louis, Mo.**





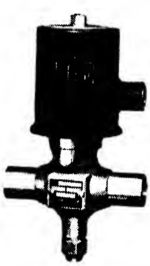
THERMO EXPANSION VALVES—To secure the greatest efficiency from an evaporator, as much of its surface as possible must be used to absorb latent heat by vaporizing the liquid refrigerant.

Alco Thermo Valves assure positive independent and automatic control of the liquid feed in an evaporator in accord with the refrigerating load.

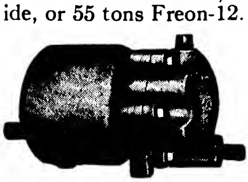
They are responsive to any change in the suction gas superheat—this means less evaporator surface is required for securing control and a higher average suction pressure is maintained, with a corresponding increase in compressor capacity and shortened running time.

There is a size and type of Alco control for all the usual refrigerants, with capacities from fractional tonnage to 60 tons Ammonia, 100 tons Methyl Chloride or 50 tons Freon-12.

MAGNETIC LIQUID STOP VALVES—are positive acting, and tight closing. They are indispensable wherever instantaneous closing of the liquid supply line is indicated. They are used extensively with expansion valves where the temperature difference between the refrigerant and the refrigerated substance or area is very small. They are also used extensively with Alco Float Switches to maintain a constant liquid level in flooded evaporators.



All types are available in all the ordinary pipe sizes up to 1¼ in., and for tonnage capacities ranging from fractional tonnage to 350 tons Ammonia, 115 Methyl Chloride, or 55 tons Freon-12.



LIQUID FLOAT VALVES—are provided with a vent tube which prevents gas binding, and permits the valve to be installed at the highest point on a full flooded system even though many feet above the liquid receiver. They may also be installed at a low point in the system and will perform equally as well. Available in a variety of capacities up to 25 tons Ammonia, 10 tons Methyl Chloride, or 5 tons Freon-12.

MAGNETIC SUCTION STOP VALVES—are designed for use as suction line shut-off or as low side by-pass valves. They are built without packing so as to operate successfully on heavily frosted lines. Makes possible individual control of two or more evaporating units in a multiple system even though there is a wide difference in temperature requirement or load conditions. They provide individual temperature control in any number of refrigerated units in a series system either flooded or fed by a constant pressure expansion valve.

Built in ½ in., ¾ in., 1 in., 1¼ in., 1½ in., or 2 in., sizes.

ELECTRIC FLOAT SWITCHES—will stand high pressures and may be used on either a-c or d-c current. They will maintain a liquid level within 1 in. when used to operate a Magnetic Stop Valve in the liquid line on individual flooded evaporators or coolers. They may be used as a high or low level alarm or to start small motors.

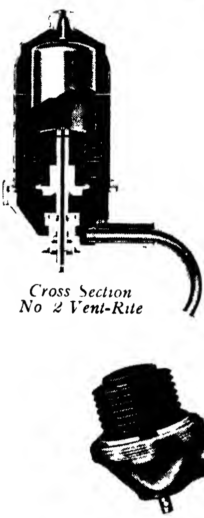
Write for the complete story of ENGINEERED REFRIGERANT CONTROL

Anderson Products, Incorporated

Cambridge, Massachusetts

Vent-Rite Radiator Air Valves. The Vent-Rite Balancer. Originators of "Balanced Radiation by Controlled Venting."

No 2 Vent-Rite Vacuum Valve



Cross Section No 2 Vent-Rite

The complete Line of Vent-Rite Air Valves comprises nine valves of various types, sizes, outlets and venting capacities. All are noiseless in operation, positive in action, close thermostatically under temperature.

Vent-Rite Vacuum Valves do not depend upon a ball or disc to maintain vacuum in a system. Vacuum is maintained by means of a sensitive and positive acting bellows This bellows is internally subjected at all times to atmospheric pressure which elongates the bellows, and closes the vent when the internal pressure in the valve is less than the atmospheric pressure.

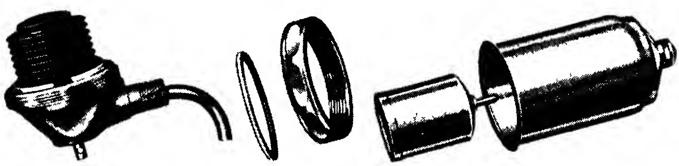
Venting takes place through an adequate, straight line venting orifice, which is accurately set by means of a modulating adjustment of the valve pin, toward or away from the valve seat.

The adjustment is underneath the valve—out of sight, cannot be disturbed by accident or meddlesome fingers

These Vent-Rite features insure Permanent, "Balanced Radiation" of any one-pipe steam system, which means—the quick, uniform transfer of steam to all radiators regardless of their number, sizes or distances from the boiler

All Vent-Rites are made of the finest non-rusting—non-corroding materials throughout to insure years of trouble-free service. Bases are Brass Forgings, Valve Pins are nickel silver. Valves are of attractive modern design, finished in chromium.

Vent-Rites can be taken apart for examination or cleaning and can be tightly reassembled Union couplings with special heavy copper-asbestos gaskets insure leak-proof joints



Only Vent-Rites can be taken apart for thorough cleaning

THE VENT-VAC SYSTEM

Increases the Performance of One-Pipe Steam Heating—Automatically Fired—To Its Highest Efficiency

The Vent-Vac System functions as an Open Atmospheric and Vacuum System combined, retaining only the Desirable, Practical Features of the two systems It enlarges the scope of one-pipe steam heating and makes it practical for schools, hospitals, apartment houses, etc., as well as residences.

The Vent-Vac System modernizes old, one-pipe jobs into new, highly efficient heating systems; eliminates lagging radiators and overheated spots.

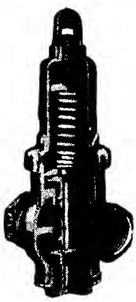
Write for new, illustrated booklet which gives complete information on Vent-Rite Valves and the Vent-Rite Balancer which makes possible the Vent-Vac System.

FOSTER ENGINEERING Company

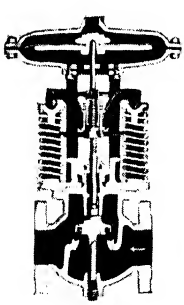
114 Monroe Street, **Newark, New Jersey**

Agencies in All Principal Cities in the United States and Canada

The Foster Engineering Company manufactures selected types of automatic valves and regulators for controlling high and low, saturated or superheated steam, liquid, or gas pressures for the widest range of services, and maintains a staff of thoroughly experienced engineers for the best solution of problems pertaining to automatic regulation or control.



Type 38-U



Type 37-A1

Pressure Reducing Regulators

Type 38-U. For intermittent and dead-end service on water or air; for initial pressures up to 300 lb with special constructions for higher pressures. Pilot or direct-acting diaphragm-actuated valves manufactured for low delivery pressures. Built in bronze and semi-steel; sizes ½ to 20 in.

Types 37-A1 and A2. These regulators offer every advantage of double spring loading. The Type 37-A1 is single seated for dead-end service where initial pressure is constant and is built in sizes of ½ through 12 in. The Type 37-A2 is double seated for continuous flow . . . an internally balanced regulator unaffected by variations in line pressure. Built in sizes of ½ to 12 in.

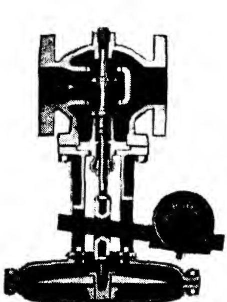
Types 37-B1 and B2. Sensitivity at low reduced pressures is assured by these weight-loaded regulators. The Type 37-B1 is single-seated for dead-end service where the initial pressure is constant and is built in sizes of ½ through 12 in. The Type 37-B2 is double-seated for continuous flow . . . internally balanced; unaffected by variations in pressure. Sizes ½ to 12 in.

Temperature Regulators

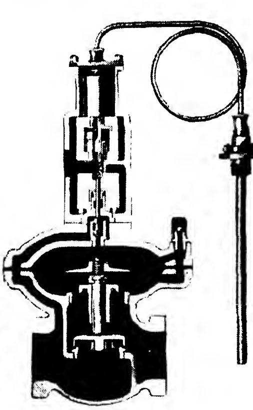
Types 34-T and 34-T2. Type 34-T is a pilot-operated self-contained, single-seated regulator for dead-end service. Built in sizes of ½ through 3 in. for pressures of 10 to 200 lb, this regulator maintains temperatures of gases or liquids within plus or minus one degree Fahrenheit. Double-seated in sizes of 3½ to 8 in. The Type 34-T2 is a direct-acting double-seated regulator built in sizes of ½ to 6 in. for pressures 0 to 125 lb.

Float Valves

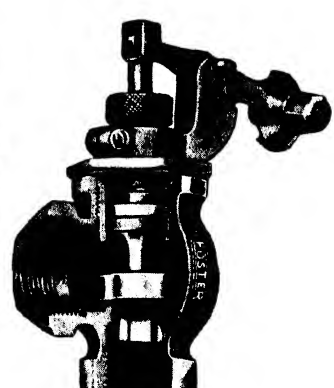
Auxiliary Operated Float Valves. Extremely sensitive valves for hot or cold water service, actuated by the pressure in the supply line. A tied-in auxiliary operated type has the piston loosely connected to the stem, combining the auxiliary and direct action. Both auxiliary and direct acting types made in sizes ¾ to 12 in., angle and globe.



Type 37-B2



Type 34-T



A. O. Float Valves

Henry Valve Company

1001-19 North Spaulding Ave., **Chicago, Ill.**

Manufacturers of complete line of Dryers, Strainers and Large Line Valves for Freon and Methyl Chloride. Also Ammonia Valves and Forged Steel Fittings.

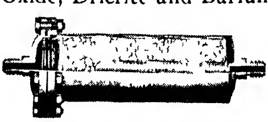
ABSO-DRY PRESSURE SEALED DRYERS

For Refrigeration and Air Conditioning

Exclusive Henry vacuum process first removes every trace of moisture, then the dryer is charged with dehydrated air. Loosening seal cap produces hissing sound, a guarantee of original factory dryness

OTHER FEATURES OF HENRY DRYERS—*Perforated Dispension tube* is connected to inlet port and exposes entire volume of dehydrant to penetration by refrigerant. *Minimum* pressure drop No channelling. *Compression Spring*, maintains uniform tension on dehydrant at all times and compensates for changes in volume. *Soldered or Flanged Shells*—models are available with either soldered cap or flanged end shells Flange is distortion-proof. Shells not exceeding 5½ in in length are drawn in dies, so that they have only one joint.

FIVE DEHYDRANTS—Choice of following dehydrants at same price Activated Alumina, Calcium Chloride, Calcium Oxide, Drierite and Barium Oxide.



Type 774 Cartridge Dehydrator

A flanged shell dehydrator with replaceable cartridge

Type 721 Dehydra-Tector

Combination dryer with capped liquid sight port for determining sufficiency of refrigerant in system Three sizes available with dehydrant capacity of 13.5, 31 5 and 47 2 cu in.

AUTOMATIC RELIEF VALVE

Angle type with push rod for emergency reseating Available pressure settings: 90 to 250 lb. Approved for use under many refrigerating and air conditioning safety codes.

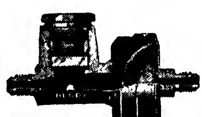
HENRY STRAINERS

There is a size and type of Henry Strainer for every installation requirement.

Type 895 Strainer

With solder fittings for use with copper pipe. Exceptional design. Welded steel construction. Negligible pressure drop Screen can be taken out for cleaning without removing strainer from line Very large screen area Light weight Baffle prevents heavy particles injuring screen

Strainer and Liquid Indicator

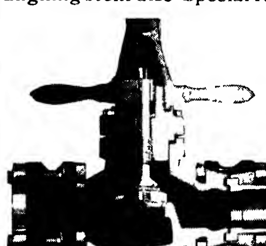


Type 888

Has sight port for determining sufficiency of refrigerant in system. Gas bubbles, passing under sight port glass, indicate shortage. Sight port is capped

WING CAP VALVES

Designed especially for Freon and Methyl Chloride. Have patented rotating self-aligning stem disc Special resilient packing



May be repacked under pressure. Wing cap can be inverted and socket used for operating valve Screw end, soldered and flanged connections

FREE CATALOG

It describes the complete line of Henry Dryers, Strainers and large line valves used in refrigeration and air conditioning.

ASK FOR IT.

Jenkins Bros.

BRONZE - IRON - STEEL VALVES

Mechanical Rubber Goods

80 WHITE ST., NEW YORK, N. Y.; 524 ATLANTIC ST., BOSTON, MASS.; 376 SPRING ST., ATLANTA, GA.; 133 N. SEVENTH ST., PHILADELPHIA, PA.; 1514 FULTON ST., CHICAGO, ILL.; 1112 WALNUT ST., HOUSTON, TEX.

BRIDGEPORT, CONN. (Office and Factory)

JENKINS BROS., LTD.: LONDON, W.C. 2; MONTREAL, QUE., (Works and Main Office).

IN VALVES Jenkins GIVES YOU EVERYTHING

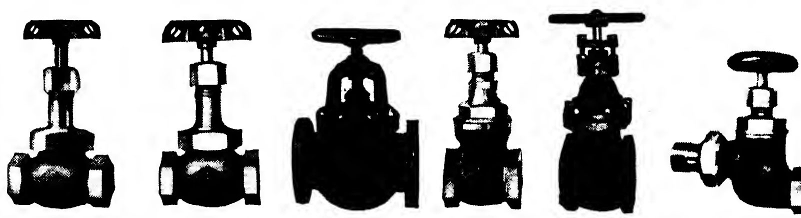


Fig. 106A Bronze Globe, Renewable Comp. Disc — *Fig. 950* Bronze Globe, Regrind-Renew — *Fig. 142* Iron Body Globe — *Fig. 370* Bronze Gate — *Fig. 325* Iron Body Gate — *Fig. 859* Radiator Offset Globe

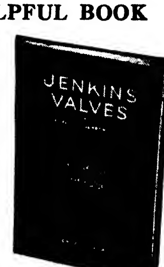
OVER 500 DIFFERENT JENKINS VALVES COVER EVERY HEATING AND AIR CONDITIONING NEED

To adequately describe the complete Jenkins line of valves requires a Catalog of more than 300 pages. There are over 500 different types and patterns of valves that bear the trusted "Diamond" trade mark. Practically speaking, Jenkins can furnish any valve that you may require for plumbing, heating, air conditioning, general industrial or engineering service.

General Classifications of Jenkins Valves Include—Bronze Valves fitted with Jenkins renewable composition disc. Bronze Regrind-Renew Valves with bevel and plug type seats. Bronze Gate Valves. Iron Body Valves fitted with Jenkins renewable composition disc. Iron Body Regrinding Valves. Iron Body Gate Valves with solid wedge and double disc parallel seats. All-Iron Valves. Cast Steel Gate Valves and Swing Check Valves. Electrically and Hydraulically Operated Valves. Radiator Valves. Fire Line Valves. Quick-opening and Self-closing Valves, Needle Valves, Y Valves, Solder-End Valves, Stainless Steel Valves.

Other Jenkins Products Are— Colored Valve Wheels with or without service markings molded in relief letters. Composition Valve Discs exactly suited to service conditions. Sheet Packing. Gaskets. Moncrieff Scotch Gage Glasses.

CONSULT THIS HELPFUL BOOK



This 307 page Jenkins Catalog not only gives complete details on over 500 Jenkins Valves, but also it has a large section of engineering data and practical information about valves and layouts. Make sure you have a copy, including the new Supplement "B."

JENKINS VALVES ARE SOLD BY GOOD SUPPLY HOUSES EVERYWHERE

New York Air Valve Corporation

Since 1898

Nyavco

CHICAGO BOSTON — DETROIT ST LOUIS — CLEVELAND PITTSBURGH

Nyavco

Orifice Control Air Valves — 611-621 Broadway, **New York** — **Vacuum Orifice Control Air Valves**

NYAVCO ORIFICE "CONTROL BY VENTING" VALVES

Air which fills each unheated radiator must be driven out by entering steam before heat is obtained. Using this air as an "air brake" and thus limiting or increasing its evacuation time on each radiator, accordingly slows or increases speed of entering steam and its consequent heating time.

The NYAVCO Orifice Control Air Valve incorporates in one valve six gradually ascending vent speeds, which make possible the simultaneous heating of the largest and smallest radiator—or the equalization in steam delivery of the farthest unit from or nearest unit to the boiler. Thus balancing the job.

The NYAVCO method of metered venting can so "time" each room radiator large or small—near or far—as to heat simultaneously, on coal fired constant heat jobs, or with room containing the automatic control (if an automatic gas, oil or stoker fired job).

NYAVCO venting is fast because of unusually large vent port. It can only be set by the heating man—is consequently tamper-proof—because it is "locked in" the "armored cap" by him. Has a definite metered disc—Does not depend on thread or needle valve, but actual room or distance equalizing on each setting.

Nyavco ORIFICE CONTROL

Air Valve—For one or two pipe gravity steam jobs. —Note open cut showing indestructible construction and control disc.
Made also in 1/8 in, 1/4 in., 1/2 in, Straight and 3/4 in Quick Vent

Fig No 1

Nyavco ORIFICE CONTROL

Vacu-Seal Vacuum—A ball check type vacuum valve—for use on intermittent heating jobs where price is a consideration.
Made also in 1/8 in, 1/4 in., 1/2 in, Straight Shank and 3/4 in Quick Vent

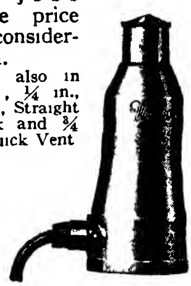


Fig No 2

Nyavco ORIFICE CONTROL

Gold-Seal-Lock-Vacuum—The Bellows operated atmospheric pressure locked — Vacuum Valve, will maintain vacuum on one pipe gravity steam jobs over very long periods
Made in angle and quick vent types only

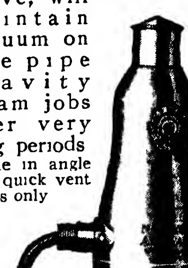


Fig No 20

OPERATION OF VALVE

INDESTRUCTIBILITY

NYAVCO because of its open float and bi-metallic mechanism, depending upon actual steam contact functions immediately and will operate efficiently under all service requirements of one pipe systems.

NYAVCO factory adjustment cannot be injured and valve is guaranteed fatigue-proof — rust-proof — shock-proof. Write for details.

GIANT
Three Speed Air ELIMINATOR
For Rapid Air Elimination from Large Mains — Coils — Air Conditioning Units —Unit Heaters, etc.

3 Speed control—1/8 in, 3/16 in, 1/4 in —Secured by removing screw from speed size desired

1/8 in. Size for load up to 1500 ft.

3/16 in Size for load of 1500 ft to 3000 ft

1/4 in Size for load of 3000 ft up.

Made in regular Venting and Vacuum Valve 6¾ Actual Height.

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TO

MODERN EQUIPMENT

In the Index to Modern Equipment are complete detailed listings of heating, ventilating and air conditioning equipment and materials.

Arranged alphabetically according to names of products are more than 300 items listing not only those products shown in the Catalog Data Section but also many other products made by the manufacturers represented in The Guide.

On pages 1137-1160, under each index heading—Air Cleaning Equipment, Fans, Humidifiers, Ventilators, etc.—will be found, fully cross-indexed, a complete list of manufacturers of any desired products and page numbers in the Catalog Data Section where the products are described. By reference to these index headings, the manufacturers names and the page numbers, any item of equipment or materials may be located quickly.

On pages 859-864 will be found an alphabetical list of manufacturers whose products are shown in the Catalog Data Section of The Guide.

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1939

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Radiation with Gravity Air Circulation—M. K. Fahnestock*, *Chairman,* B. C. Benson, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, D. W. Nelson*, T. A. Novotney, W. A. Rowe.

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Cooling Load in Summer Air Conditioning—J. H. Walker*, *Chairman;* C. M. Ashley, John Everetts, Jr., F. H. Faust, A. E. Knapp, L. S. Morse, A. E. Stacey, Jr.*, R. M. Strikeleather.

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1939

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BAKER, Lorne P. (*J* 1937) Air Cond. Engr. (for mail) Canadian General Electric Co., Ltd., Royce Ave Works, 830 Lansdowne Ave., and 115 Wembley Rd., Toronto, Ont, Canada.
BAKER, Roland H. (*M* 1928; *A* 1924) Pres, R. H. Baker Co, Elkins, N H
BAKER, Thomas (*M* 1938) Suburban Air Conditioning Corp, 7 Depot Plaza, White Plains N Y
BAKER, William C. (*M* 1938) Pres and Treas. (for mail) Electric Appliances, Inc, 155-7th Ave., N, and Westover Drive, Nashville, Tenn
BAKER, William H., Jr. (*A* 1935) Standard Air Conditioning Corp, 40 West 40th St, New York, N Y.
BALDI, G. (*A* 1936) Engr (for mail) Copagnia Italiana Westinghouse, Via Pier Carlo Boggio 20, and Corso Racconigi 39, Torino, Italy
BALDWIN, Karl F., Jr. (*J* 1938) Engr (for mail) McCrea Equipment Co, 324 Independence Ave., S W, Washington, D C, and 4810 Cedar St., Decatur Hts, Hyattsville, Md
BALDWIN, William H. (*M* 1921) Sales Engr. (for mail) C A Dunham Co, 5757 Cass Ave, and 2432 Atkinson Ave, Detroit, Mich
BALL, William (*A* 1936) Pres (for mail) Interstate Heating & Plumbing Co, 521 Southwest Blvd, Kansas City, Mo, and 1026 Shawnee Rd., Kansas City, Kan
BALLANTYNE, George L. (*A* 1936) Mgr, Htg. Sales Dept (for mail) Crane Ltd, P. O. Box 840, and 141 Bedbrook Ave, Montreal, West, P Q., Canada
BALLMAN, William H. (*M* 1937) Chief Engr. Air Cond Div (for mail) Nash-Kelvinator Corp, Long Island City, N Y
BALSAM, Charles P. (*M* 1932) 324 Fourth St., Brooklyn, N Y.
BAMOND, Manuel J. (*M* 1936) Engr, Reynolds Corp, 1400 Wabensia Ave., and (for mail) 4715 Magnolia Ave, Chicago, Ill.
BANKS, John B. (*A* 1937) Branch Mgr (for mail) Minneapolis-Honeywell Regulator Co, 2405 N. Maryland Ave, and 2928 N Maryland Ave, Milwaukee, Wis
BANNER, Francis L. D. (*M* 1937) Branch Mgr. (for mail) Minneapolis-Honeywell Regulator Co, 378 Saunders-Kennedy Bldg, and 5523 Corby St, Omaha, Nebr
BANNON, Lucas E. (*A* 1935) Archt., 16 Church St, Paterson, N J
BANOWSKY, Aubra B. (*M* 1938) Director of Commercial and Industrial Sales, United Gas Corp, Rusk Bldg, and (for mail) 3735 Ingold, Houston, Tex
BARBIERI, Patrick J. (*J* 1936, *S* 1933) Engr., Armo Cooling & Ventilating Co., 30 West 15 St., and (for mail) 2166 Belmont Ave, New York, N Y
BARNARD, M. Everett (*A* 1931; *J* 1929) Sales Engr (for mail) Carrier Corp, 12 S 12th St., and 341 Vernon Rd., Philadelphia, Pa.
BARNES, Arthur F. (*M* 1920) Owner (for mail) Texas Engineering Co, 726 Electric Bldg, and 3015 Jarrard St, Houston, Tex.
BARNES, Arthur R. (*M* 1924) Chief Engr (for mail) U S Supply Co, 1315 West 12th St, and 326 East 70th Terrace, Kansas City, Mo.
BARNES, Harry P. (*A* 1936) Mgr, Construction Dept (for mail) Johns-Manville Sales Corp., 2030 Walnut St, and 6101 Walnut St, Kansas City, Mo
BARNES, Herbert (*M* 1936) Mgr. (for mail) Herbert Barnes Plumbing & Heating, Delta Block, and 114 Grosvenor Ave, S, Hamilton, Ont, Canada
BARNES, Lewis L. (*J* 1937) Air Cond & Refrig. Engr, Carrier Atlanta Corp, 348 Peachtree St., and (for mail) 3995 N. Stratford Rd., Atlanta, Ga.

BARNES, Walter E. (*M* 1933) Pres, Barnes & Jones, Inc, 128 Brookside Ave, Jamaica Plain (Boston) and (for mail) 7 Woodlawn Ave, Wellesley Hills, Mass.

BARNEY, William E. (*M* 1936) Mgr (for mail) Hydraulic-Press Brick Co., South Park, and 4929 E 108th St, Cleveland, O

BARNSLEY, Frank Richard (*A* 1936) Mgr, Air Cond Div (for mail) Canadian General Electric Co, Ltd., 1000 Beaver Hall Hill, and 5245 Byron Ave, Montreal, P Q, Canada

BARNUM, Marvin C. (*M* 1930; *A* 1928) Eastern Repr. (for mail) Waterman-Waterbury Co, P O. Box 284, Suffern, and Cherry Lane, Tallman, N. Y.

BARNUM, Willis E., Jr. (*M* 1933, *J* 1930) Mgr., Air Cond Div, York Ice Machinery Corp, Roosevelt Ave, and (for mail) 35 N Rockburn St, York, Pa

BARR, George W. (*Life Member*; *M* 1905) (Board of Governors, 1910) Dist Mgr, Aerofin Corp, 2030 Land Title Bldg, Philadelphia, and (for mail) Woods End, Villa Nova, Pa

BARRY, James G., Jr. (*M* 1933) Vice-Pres (for mail) Elliott & Barry Engineering Co, 4060 W. Pine Blvd, and 5051 Queens Ave, St Louis, Mo

BARRY, Patrick I. (*M* 1920) (Peace Commissioner) M I H V E Managing Director (for mail) M. Barry, Ltd, 4 Marlboro St, and 8 Sidney Park, Cork, Ireland

BARTH, Herbert E. (*M* 1920) Vice-Pres (for mail) American Blower Corp, 6000 Russell St., and 829 Wardell, 15 E. Kirby, Detroit, Mich

BARTLETT, Amos C. (*M* 1919) Mgr. Htg and Vtg. Dept (for mail) B F Sturtevant Co, Damon St, Hyde Park, and 30 Hollingsworth Ave, Braintree, Mass

BARTLETT, C. Edwin (*M* 1922) Pres, Bartlett & Co, Inc, 3223 Arch St, and 3111 W Coulter St, Philadelphia, Pa

BARTLEY, Henry E. (*M* 1938) Dir and Works Mgr, Matthews & Yates, Ltd, Swinton, Manchester, and (for mail) "The Grange," Hospital Rd, Pendlebury, Lancs, England

BARTON, Jay (*M* 1937) Mgr, National Manufacturing & Engineering Co, 628 E Forest Ave, and (for mail) Box 221, Detroit, Mich

BASSETT, James W. (*A* 1938) Sales Engr (for mail) McQuay, Inc, 2832 E Grand Blvd, Detroit, and Birmingham, Mich

BASTEDO, Albert E. (*M* 1919) Vice-Pres. and Treas, Burnham Boiler Corp, Irvington, N Y

BASTEDO, George R. (*J* 1937) Lab Asst, Standard Air Conditioning, Inc, New Rochelle, and (for mail) 102-36 86th Rd, Richmond Hill, N Y

BAUER, Albert E. (*M* 1935) Engr, United States Air Conditioning Corp., 2101 N E Kennedy, Minneapolis, and (for mail) 59 S. Victoria St, St Paul, Minn

BAUGHMAN, L. R. (*M* 1935) Htg Engr, Helms & Baughman, 103 N Sheridan Rd, Waukegan, and (for mail) 2706 Eschol Ave, Zion, Ill.

BAUM, Albert L. (*M* 1916) Member of Firm (for mail) Jaros, Baum & Bolles, 415 Lexington Ave, and 600 West 111 St, New York, N Y

BAUMGARDNER, C. M. (*M* 1928) Mgr, Chicago Branch (for mail) United States Radiator Corp, 3254 N Kilbourn Ave, Chicago, and 602 Michigan Ave, Evanston, Ill

BAXTER, William E. (*A* 1939) Vtg and Air Cond (for mail) H E Baxter, Ltd, 87 Vitre, W., Montreal, and 89-51st Ave., Lachine, P. Q., Canada

BAY, Charles H. (*A* 1938) Salesman (for mail) The Detroit Edison Co, 2000 Second Ave, and 17323 Wildemere St, Detroit, Mich.

BAYSE, Harry V. (*M* 1923) Pres. (for mail) American Furnace Co, 2719-31 Delmar Blvd, and 6959 Hancock Ave, St Louis Mo

BEACH, Walter R. (*A* 1936) Sales Engr. (for mail) Cleveland Electric Illuminating Co., 75 Public Sq., Cleveland, and 1185 Yellowstone Rd., Cleveland Heights, O.

BEAN, George S. (*A* 1935) Mgr. Stoker Div and Stoker Engr (for mail) North Western Fuel Co, E. 1203 First Natl. Bank Bldg, St Paul, and 4949-16th Ave, S, Minneapolis, Minn.

BEARMAN, Alexander A. (*M* 1937) Engr. (for mail) 20th Century-Fox Film Corp, 444 W 56th St, New York, and 47 Edward St, Baldwin, N Y.

BEAULIEU, Adrian A. (*M* 1937) Utilization Engr., Boston Edison Co, 39 Boylston St, Boston, and (for mail) 535 N. Elm St., W, Bridgewater, Mass

BEAURRIENNE, Auguste* (*M* 1912) Consulting Engr (for mail) 25 Rue des Marguettes, and 18 Avenue du Petit Val, Lucyen Brie Setoise, Paris, France.

BEAVERS, George R. (*M* 1929) Chief Engr, Canadian Blower & Forge Co, Ltd, Woodside Ave, and (for mail) 60 Church St, Apt "D," Kitchener, Ont, Canada

BECHTOL, Jack J. (*J* 1937) Conditioner (for mail) Jos E Seagram & Sons, Inc, Lawrenceburg, Ind, and 4766 Rapid Run Pike, Cincinnati, O.

BECK, Don (*J* 1938) Supvr Air Cond Sales, York Ice Machinery Corp, 5051 Santa Fe Ave, and (for mail) 310 N. Orange Drive, Los Angeles, Calif

BECKER, Roger K. (*M* 1938) Dept Mgr, Ohio Valley Hardware & Roofing Co, and (for mail) 1017 E Powell Ave, Evansville, Ind.

BECKER, Walter A. (*M* 1935) Sales Engr (for mail) Grinnell Co, Inc, P O Box 292, and 215 Maple Terrace, Oconomowoc, Wis

BEEBE, Frederick E. W. (*A* 1915) Johnson Service Co, 28 East 29th St, New York, N Y

BEERY, Clinton E. (*M* 1913) Owner Pres (for mail) Heat & Fuel Engineering Co, 646 N Michigan Ave, and 3750 Wayne Ave, Chicago, Ill

BEGGS, William E. (*M* 1927) Owner, W E Beggs Co, 907 Lloyd Bldg, and (for mail) 3639 Palatine Ave, Seattle, Wash

BEIGHEL, H. A. (*A* 1927) Sales Repr (for mail) Herman Nelson Corp, 503 Columbia Bldg, Pittsburgh, and 207 Puritan Rd, Rosslyn Farms, Carnegie, Pa

BEITZELL, Albert E. (*A* 1933, *J* 1930) Mgr, Air Cond Div, Combustioneer Corp, 409 10th St, S W, and (for mail) 1213 Hamilton St, N. W, Washington, D C

BELDING, Harry H. (*A* 1937) Chief Engr, Gathright, Inc, 1840 W Broad St, and (for mail) 2616 Chamberlayne Ave, Richmond, Va

BELING, Earl H.* (*M* 1936; *A* 1930, *J* 1925) Owner, Beling Engineering Co, 405 State Trust Bldg, and (for mail) 2428-13th St, Moline, Ill

BELL, E. Floyd (*M* 1933) Mgr (for mail) Bell & Eiss, Inc., 2102 Foshay Tower, and 5337 Girard Ave, S, Minneapolis, Minn

BELSKY, George A. (*A* 1937) Air Cond Engr, Nash Kelvinator Corp, 27th St and Pearson Pl, Long Island City, and (for mail) 53 N Grove St, Valley Stream, L I, N Y

BELT, Newton O. (*M* 1929) Engrg Dept (for mail) E. I Du Pont de Nemours & Co, and 824 West 10th St, Wilmington, Del

BEMAN, Myron C. (*M* 1926) (Council 1934-1938) Consulting Engr. (for mail) Beman & Candee, 374 Delaware Ave, and 699 Richmond Ave, Buffalo, N Y.

BENHAM, Colin S. K. (*J* 1937) Dir (for mail) Benham & Sons, Ltd, 66 Wigmore St, London W 1, and 31 Ormonde Terrace, London N W 8, England

BENHAM, Ford C., Jr. (*J* 1938) Sales Engr (for mail) c/o C H Ruebeck Co, P O Box 141, and 1106 N 16th St, Waco, Tex

BENNETT, Charles A. (*M* 1936) 1751 Kilbourne Place, N. W, Washington, D. C.

BENNETT, Edwin A. (*M* 1936; *J* 1929) Sales Engr (for mail) American Blower Corp., 50 W 40th St., New York, and 45 Pondfield Rd, W, Bronxville, N. Y.

BENNITT, George E. (*M* 1918) 44 Cedar St., Wakefield, Mass.

BENOIST, LeRoy L. (*M* 1934) Mgr (for mail) Benoist Bros. Supply Co., 117 S. Tenth St., and 1500 Main St, Mt. Vernon, Ill

BENOIST, Raymond E. (*M* 1936) Partner, Benoist Bros, and Secy-Treas, Benoist Bros Supply Co., 117 South 10th St., and (for mail) 811 North 12th St., Mt. Vernon, Ill

BENSEN, Clarence L. (*J* 1935) Chief Engr. (for mail) McQuay, Inc, 1600 Broadway, N E, and 2722 Benjamin St, N E, Minneapolis, Minn

BENSINGER, Mark (*J* 1936) Asst Mech Engr (Htg & Air Cond) War Dept, U S Government, Room 2347, Munitions Bldg, and (for mail) 2737 Devonshire Pl, N. W, Washington, D. C

BENSON, Bernard C. (*M* 1937) Sales Promotion Mgr., Chicago Branch (for mail) American Radiator Co, 820 S Michigan Ave, and 8127 Clyde Ave, Chicago, Ill

BENSON, Merrill L. (*M* 1938) Mgr, Air Cond Coil Div. (for mail) McQuay, Inc, 1600 Broadway, N E, and 4521 Harriet Ave, S, Minneapolis, Minn

BENTLEY, Clyde E. (*M* 1937) Consulting Engr, 216 Pine St, San Francisco, and (for mail) 1875 San Antonio Ave, Berkeley, Calif

BENTZ, Harry (*M* 1915) Vice-Pres (for mail) Davis Engineering Corp, 1064 E Grand St, Elizabeth, and 18 Holland Terrace, Montclair, N J

BERGAN, John R. (*J* 1937) Dist Repr (for mail) Minneapolis-Honeywell Regulator Co, 1220 Madison Ave, Toledo, and 525 W. Broadway, Maumee, O

BERMAN, Louis K. (*M* 1908) Pres (for mail) Raisler Corp, 129 Amsterdam Ave, and 285 Central Park West, New York, N Y

BERMEL, Alfred H. (*A* 1933; *J* 1928) Estimator and Engr, August Arace & Sons, Inc, 642-3rd Ave, Elizabeth, and (for mail) 16 William St, No Arlington, N J

BERNERT, Lawrence A. (*A* 1937) Mgr, Htg & Air Cond Dept, The Maag Co, 831 N Milwaukee St, Milwaukee, Wis

BERNHARD, George (*M* 1935, *A* 1929) Managing Engr, Associated Heating & Power Corp, 1 Hanson Pl., and (for mail) 985 Park Place, Brooklyn, N Y

BERNSTROM, Bert* (*M* 1930) Consulting Engr, B Bernstrom Air Cond Consultant, 2853 Dickens Ave, Chicago, Ill

BERRIDGE, Winston W. (*M* 1938) Sales Engr (for mail) McColl-Frontenac Oil Co, Ltd, Dominion Square Bldg, and 5169 Westbury Ave, Montreal, P Q, Canada

BERRINGER, Sidney H. (*M* 1926) Chief Engr, Holly Heating & Mfg Co, 21 S Chester, Pasadena, and (for mail) 2083 Maiden Lane, Altadena, Calif.

BERZELIUS, Carl E. (*M* 1936) Captain, Commanding Officer, CCC Camp (for mail) Co 784 CCC, and 101-Wisconsin, Neodesha, Kan

BETLEM, Henriette T. (*J* 1934) Air Cond Engr (for mail) Betlem Heating Co, 1926 East Ave, and 1293 Park Ave, Rochester, N. Y

BETTS, Howard M. (*M* 1927) Sr. Mech Engr, Htg & Vtg (for mail) Dept of Buildings, City of Minneapolis, 213 City Hall, and 4923 S Russell Ave, Minneapolis, Minn

BETZ, Harry D. (*M* 1928) Pres, Betz Air Conditioning Corp, Six West Ninth St, and (for mail) 1610 Valentine Rd, Kansas City, Mo

BEVINGTON, Curtis H. (*M* 1936) Mgr (for mail) C H Bevington Co, 600 S Michigan Ave, Chicago, and Park Ridge, Ill

BEWS, John (*A* 1938) Dist Sales Mgr (for mail) Canadian Ice Machine Co, Ltd, 628 Craig St, West, and 5546 Trans Island Ave, Montreal, P. Q, Canada

BIANCULLI, Vincent A. (*J* 1937) Draftsman, Navy Dept., Brooklyn, and (for mail) 557 Broome St, New York, N Y

BIBER, Herbert A. (*A* 1937) Engr (Htg, Air Cond Refrigeration) Mellon National Bank, 542 Smithfield St, Pittsburgh, and (for mail) 323 Barnes St, Wilkinsburg, Pa

BICHOWSKY, F. Russell (*M* 1935) Consulting Engr. (for mail) Dow Chemical Co., 309 S. State St, and 1508 Granger, Ann Arbor, Mich.

BIGELOW, Edward S. (*M* 1938) Mgr., Air Cond. Div, Trilling & Montague, 2409 Walnut St., Philadelphia, and (for mail) 413 Jericho Rd., Montgomery Co, Abington, Pa.

BILLINGSLEY, Oliver F., 2nd (*J* 1937) Owner and Engr, Foo Hobbylab, 424 N St Mary's St., and (for mail) P O Box 1740, San Antonio, Tex.

BINDER, Charles G. (*M* 1920) Mgr, Htg. Dept, Warren Webster & Co, 17th & Federal St., Camden, and (for mail) 115 Oak Terrace, Merchantville, N J

BIRD, Charles (*A* 1934) Treas & Gen Mgr. (for mail) The Doermann Roehrer Co, 450 E. Pearl St, and Box 179, D Section Rd, R. R. No. 6, Cincinnati, O

BIRD, George L. H. (*J* 1937) Chief Engr (for mail) Refrigeration & Allied Products, Ltd., 92 Buckingham Palace Rd, London, S W 1, and No 9 Holmefield Court, London, N. W. 3, England

BISHOP, Charles R. (*Life Member*, *M* 1901) 22 Sagamore Rd, Bronxville, N Y

BISHOP, Frederick R. (*M* 1921) Mgr of Sales, The Brundage Co, Kalamazoo, and (for mail) 8011 Dexter Blvd, Detroit, Mich

BISHOP, Joseph W. (*M* 1939) Mgr. Air Cond Div, Toronto Dist (for mail) Canadian General Electric Co, Ltd, 214 King St, W, Toronto, and 62 Highland Crescent, York Mills, Ont, Canada

BISHOP, Marion W. (*J* 1935) Sales Engr. (for mail) American Blower Corp, 228 N LaSalle St, and 7024 Sheridan Rd, Chicago, Ill.

BJERKEN, Maurice H. (*M* 1937, *A* 1927) Sales Engr, Hoffman Specialty Co, and (for mail) 4952-17th Ave, S., Minneapolis, Minn.

BLACK, Edgar N., 3rd (*M* 1922) Philadelphia Mgr, Fitzgibbons Boiler Co, Inc, 927-28 Land Title Bldg, Philadelphia, and (for mail) 111 Woodside Rd, Haverford, Montgomery Co, Pa

BLACK, F. C. (*Life Member*; *M* 1919) Pres (for mail) F C Black Co, 622 W Randolph St, and 4535 N Ashland Ave, Chicago, Ill

BLACK, Frank M. (*A* 1937) Chief Engr, U S Government, Army Medical Center, Washington, D C, and (for mail) P. O. Box 164, Silver Spring, Md

BLACK, Harry G. (*M* 1917) Prop (for mail) P Gormly Co, 155 N 10th St, and 927 N 65th St, Philadelphia, Pa

BLACKBURN, E. C, Jr. (*M* 1929) Mech Engr, Crow, Lewis & Wick, Archts, 200 Fifth Ave, New York, and (for mail) 5 Kenwood Rd, Garden City, N Y

BLACKHALL, W. R. (*M* 1922) Partner (for mail) McKeller & Blackhall, 1104 Bay St, and 332 Waverly Rd, Toronto, Ont, Canada

BLACKMAN, Alfred O. (*M* 1911) Htg and Vtg. Engr, 145 W. 45th St, and (for mail) 450 W. 24th St, New York, N Y

BLACKMORE, F. H. (*M* 1923) Mgr, Mfg Dept. (for mail) U S Radiator Corp, 1056 Natl Bank Bldg, Detroit, and 515 Tooting Lane, Birmingham, Mich

BLACKMORE, George C. (*Charter Member*; *Life Member*) Pres (for mail) Automatic Gas Steam Radiator Co, 301 Brushton Ave, and Cathedral Mansions, Pittsburgh, Pa

BLACKMORE, J. J.* (*Charter Member*, *Life Member*) 32 West 40th St, New York, N Y

BLACKMORE, James S. (*J* 1931) Philadelphia Dist Mgr, H. A Thrush & Co., Peru, Ind, and (for mail) 728 Manoa Rd, Upper Darby, Pa.

BLACKMORE, Joseph J. (*J* 1937) Sales Engr. (for mail) "X" Laboratories and Bell & Gossett, 4006 Papin, St Louis, and 312 S. Fillmore, Edwardsville, Ill

BLACKSHAW, J. L.* (*M* 1937; J 1929) Air Cond Engr, Air and Refrigeration Corp, 11 West 42nd St, New York, and (for mail) 59 Joralemon St, Brooklyn, N Y

BLAKELEY, Hugh J. (*M* 1935) Consulting Engr. (for mail) Hubbard Rickerd & Blakeley, Consulting Engrs, 275 Orange St, and 5 Doty Place, New Haven, Conn

BLANDING, Robert L. (*M* 1938) Vice-Pres (for mail) Taco Heaters, Inc., 123 South St, and 1385 Smith St, Providence, R. I

BLANKIN, Merrill F. (*M* 1927; *A* 1926; *J* 1919) Pres (for mail) Haynes Selling Co, Inc, S. E Cor Ridge Ave, and Spring Garden St, and 528 E. Gates St, Roxboro, Philadelphia, Pa.

BLAS, Romualdo J. (*M* 1936) Sanchez & Co, Apartado Postal 1006, Caracas, Venezuela, South America

BLEDSOE, Raymond P. (*J* 1937) Sales Engr. Madison Gas & Electric Co, and (for mail) 531 N Pinckney St, Madison, Wis.

BLOOM, Louis (*M* 1935) Partner, Freeport Plumbing and Heating Engineers, 84-A, Broadway, Freeport, N. Y

BLUM, Herman, Jr. (*J* 1926) Engr (for mail) C Wallace Plumbing Co, 2224 Summer St, and 5912 Elliot St, Dallas, Tex.

BLUMENTHAL, M. I. (*M* 1936) Engr. in Charge Refr & Air Cond (for mail) National Schools, 4000 S Figueroa, and 648 W. Santa Barbara, Los Angeles, Calif

BOALES, William G. (*M* 1936; *A* 1923) Owner (for mail) Wm G Boales & Associates, 6439 Hamilton Ave, Detroit, and 195 McMillan Rd, Grosse Pointe Farms, Mich

BOCK, Bernard A. (*A* 1929; J 1927) Mech Draftsman, 57 Elizabeth Ave, Arlington, N J.

BOCK, I. I. (*A* 1934) Pres (for mail) Carrier-Bock Corp., 2022 Bryan St, and 2500 South Blvd, Dallas, Tex

BODEN, Walter F. (*A* 1937) Branch Mgr (for mail) Modine Mfg Co, 420 E Wells St, Milwaukee, and 606 Milwaukee Ave, South Milwaukee, Wis

BODINGER, Jacob H. (*M* 1931) Pres (for mail) Bodinger & Co, Inc, 530 Tenth Ave, New York, and 1429 East 19th St, Brooklyn, N Y.

BODMER, Emmanuel (*M* 1937) Engr, Head of Tech Dept, Ets. Dieny & Lucas, 223 Boulevard Pereire, and (for mail) 20, rue Leon, Paris (18e), France

BOESTER, Carl F., Jr. (*A* 1936) Air Cond. Engr., 101 E Essex, Kirkwood, St Louis, Mo

BOGATY, Hermann S. (*M* 1921) 735 E. Phil-Ellena St, Philadelphia, Pa.

BOLAND, Roy O. (*A* 1938) Mgr, Insulation Div (for mail) Alexander Murray & Co, Ltd, 4035 Richelieu St, Montreal, and 348 Kensington Ave, Westmount, P. Q, Canada.

BOLSINGER, R. C. (*M* 1916) (Council, 1936, 1937, 1938) Dist Repr, General Stokers, Inc' Philadelphia, Pa, and (for mail) 238 E Madison Ave, Collingswood, N. J.

BOLTON, Reginald P.* (*Honorary Member; Life Member*, *M* 1897) (*Presidential Member*) (Pres 1911; 1st Vice-Pres 1905-1910, 2nd Vice-Pres 1903, Board of Governors, 1901, 1905, 1910, 1911, 1912, 1913) The R. P Bolton Co., 116 East 19th St, New York, N Y.

BOND, Harry H. (*M* 1938) Partner (for mail) Edward E. Ashley, Cons Engr, 10 East 40th St., New York, and 141-49 181st St., Springfield, L. I, N. Y.

BOND, Horace A. (*M* 1930) Dist. Mgr, Warren Webster & Co., 152 Washington Ave, and (for mail) 12 Ramsey Pl., Albany, N. Y.

BONTHRON, Robert C. (*A* 1935) Headquarters Syndicate Repr, Air Cond Sales (for mail) Westinghouse Electric & Mfg. Co., 150 Broadway, New York, and 44 Ingraham Blvd, Hempstead, L I, N Y

BOOT, Arthur (*M* 1938) Mgr, Air Cond Div. (for mail) Boot & Co, 115 W. Fulton St., and 928 Orchard Ave, S. E, Grand Rapids, Mich.

BOOTH, Charles A. (*M* 1917) Vice-Pres & Sales Mgr. (for mail) Buffalo Forge Co., 490 Broadway, and 142 Summit Ave., Buffalo, N. Y.

BORAK, Eugene (*M* 1937) Chief Draftsman (for mail) Buensod-Stacey Air Cond Corp, 60 East 42nd St, New York, N. Y, and 261 Manhattan Ave., Jersey City, N. J

BORG, Elmer H. (*M* 1938) Partner (for mail) Proudfoot Rawson-Brooks & Borg, Archts, 815 Hubbell Bldg, and 3101 Easton Blvd, Des Moines, Ia

BORKAT, Philip (*J* 1936) 3116-18th St, N W, Washington, D C

BORLING, John R. (*A* 1934) Engr-Custodian (for mail) Chicago Board of Education 214 N. Lavergne Ave, and 953 East 84th Place, Chicago, Ill

BORNEMANN, Walter A. (*M* 1924; *J* 1923) Sales Engr. (for mail) Carrier Corp., 12 South 12th St., Philadelphia, and 123 W Wharton Ave., Glenside, Pa

BORNSTEIN, William (*A* 1937) (for mail) Wm. Bornstein & Son, 5424 Third St, N W, Washington, D C., and 222 Chestnut Ave, Takoma Park, Md.

BOTELHO, Nanto J. (*A* 1937) Engr and Mgr, Ceibrasil Representacoes Ltda, Rua General Camara, 64-70 andar, Rio de Janeiro, Brazil (S A)

BOTTUM, Edward W. (*J* 1938) Engr, Kelvinator Engrg Dept, Nash Kelvinator Corp, 14250 Plymouth Rd, and (for mail) 2051 W. Grand Blvd, Detroit, Mich

BOUEY, Angus J. (*A* 1937; *J* 1930) Sales Engr. (for mail) The B F. Sturtevant Co, 681 Market St, and 4810 Fulton St, San Francisco, Calif.

BOUILLON, Lincoln (*M* 1933) Consulting Engr. (for mail) 1411 Fourth Ave Bldg, and 2211-32nd South, Seattle, Wash

BOWEN, Harold C. (*A* 1938) Treas, Joseph A Bowen Co, 100 Pleasant St, and (for mail) Box 202, Warren, R I

BOWEN, John C. (*A* 1938) Sales Engr, Lennox Furnace Co., Marshalltown, Ia, and (for mail) 421 S 17th St, La Crosse, Wis

BOWERMAN, Everett L. (*A* 1937) Sales Engr., Francis Hankin and Co, Ltd, 165 Spadina Ave., and (for mail) 274 Belsize Dr, Toronto, Ont., Canada

BOWERS, Arthur F. (*A* 1919) Pres, Industrial Heating & Engineering Co, 828 N. Broadway, Milwaukee, Wis

BOWLES, Edmund N. (*A* 1937) Northwest Air Cond. Supvr (for mail) Westinghouse Electric & Mfg. Co., 20 N. Wacker Drive, and 6043 N. Paulina St, Chicago, Ill

BOWLES, Potter (*A* 1928) Pres. (for mail) Hoffman Specialty Co, Inc, 757 Pacific St, Stamford, and Box 61, New Canaan, Conn.

BOXALL, Frederick (*M* 1937) Mgr of Air Cond. Dept (for mail) J H. Vivian Co, Box 301, Johannesburg, South Africa, and Carbondale Div., Worthington Pump Co, Harrison, N. J.

BOYAR, Sidney L. (*J* 1937) Estimating Supvr, Sears Roebuck & Co, 925 S Homan Ave., Chicago, and (for mail) 1515 Schilling Ave., Chicago Heights, Ill

BOYD, Spencer W. (*M* 1937; *J* 1931) Consulting Engr. (for mail) Newcomb & Boyd, Trust Co of Georgia Bldg., and 1505 Fairview Rd, Atlanta, Ga.

BOYD, Thomas D. (*M* 1937) Sales Engr. (for mail) Johnson Service Co, 1113 Race St., and 4220 Erie Ave, Cincinnati, O.

BOYDEN, Davis S.* (*M* 1909) (Pres. 1937; 1st Vice-Pres, 1936; Treas, 1933-1934, Council, 1917, 1930, 1937, 1938) Engr., Boston Edison Co, 39 Boylston St, Boston, Goodrich St., Lunenburg, and (for mail) Box 386 Shirley, Mass.

BOYKER, Robert Owen (*J* 1935) Contractor, Mac Boyker & Son (for mail) 220 1st Ave., and 100 Kennebeck Ave., Kent, Wash.

BOYLE, John R. (*M* 1935) Traveling Sales Mgr., Westerlin & Campbell Co, 1113 Cornelia Ave., and (for mail) 6858 Osceola Ave, Chicago, Ill.

BOZEMAN, Richard (*M* 1936; *J* 1929) Chief Instructor, Air Cond. Div. (for mail) Jourden Diesel Schools, Inc, 2831 N. Broad St., and 709 Church Lane, Germantown, Philadelphia, Pa

BRAATZ, Chester J.* (*M* 1930) Sales Mgr, Temperature Control and Uni-Flo Dept., Barber-Colman Co, and (for mail) 1819 Clinton St, Rockford, Ill.

BRABBEE, Dr. Charles W.* (*M* 1925) 50 Lincoln Ave, Tuckahoe, N Y.

BRACKEN, John H. (*M* 1927) Mgr, Industrial Uses Dept. (for mail) The Celotex Corp, 919 N. Michigan Ave., and 455 Oakdale Ave, Chicago, Ill.

BRADFIELD, William W. (*M* 1926) Mech Engr. (for mail) 341 Michigan Trust Bldg, and 1352 Franklin St, S E, Grand Rapids, Mich

BRADFORD, Gilmore G. (*M* 1936) Mgr., Frigidaire Div, General Motors China Ltd, 406 Holland House, Hong Kong, China.

BRADLEY, Eugene P. (*M* 1906) Pres (for mail) Hester-Bradley Co, 2835 Washington Ave, and 6935 Pershing Ave, St Louis, Mo

BRADLEY, J. M. (*A* 1938) Chicago Repr (for mail) Airtherm Mfg Co, 312 N Loomis St, and 1352 Astor St, Chicago, Ill

BRANDI, O. H. (*M* 1930) Dipl Ing, Rud Otto Meyer, Hamburg 23, and (for mail) Reinbek b Hamburg, Hamburgerstr 14, Germany

BRANDT, E. H., Jr. (*M* 1928) Pres (for mail) Reliance Engineering Co, Inc, P. O Box 1292, and 1101 Providence Rd, Charlotte, N C

BRASHAW, Clarence J. (*A* 1938) Sales Engr, J F. Stampfer Co, 8th & Main St, and (for mail) 765 Chestnut St, Dubuque, Ia

BRATT, Hero D. (*M* 1937) Sales Engr, Warren Webster & Co, 228 Ottawa Ave, and (for mail) 2259 Stafford Ave, Grand Rapids, Mich

BRAUER, Roy (*M* 1926) Mgr, Pittsburgh Office (for mail) The Trane Co, Magee Bldg, and 576 Austin Ave, Mt Lebanon, Pittsburgh, Pa.

BRAUN, John J. (*M* 1932) Factory Mgr, The United States Playing Card Co, and (for mail) 4305 Floral Ave, Norwood, O

BRAUN, Louis T. (*M* 1921) Executive Secy. (for mail) Chicago Master Steamfitters Assn, 228 N LaSalle St, and 1548 Pratt Blvd, Chicago, Ill

BRAYMAN, Albert I. (*J* 1937) Draftsman and Estimator, Edward Brayman, Htg Contractor, 81 Chamber St, Boston, and (for mail) 340 Boulevard, Revere, Mass

BRECKENRIDGE, L. P.* (*Honorary Member*; *Life Member*; *M* 1920) Prof. of Mech. Engrg, Emeritus Yale University (for mail) "The Brackens," North Ferrisburg, Vt

BREDESEN, Bernhard P. (*A* 1931) Engr (for mail) Reese & Bredesen, 403 Essex Bldg, and 3319 Knox Ave, N, Minneapolis, Minn

BRENEMAN, Robert B. (*A* 1931, *J* 1927) Sales Engr (for mail) Armstrong Cork Co, 191 Orchard Lane, Columbus, O

BRIDE, William T. (*M* 1928, *J* 1925), Supt. Engrg, Bride-Grimes & Co, 9 Franklin St (for mail) P O Box 777, Lawrence, and 28 Albion St., Methuen, Mass

BRIERLY, Keppel (*J* 1938) Air Cond Engr., Public Service Co of Colorado, 900 15th St., and (for mail) 1378 Dexter St, Denver, Colo

BRIGHAM, Clare M. (*M* 1925) Vice-Pres in charge of Sales (for mail) C. A Dunham Co, 450 E. Ohio St, and 420 Maple Ave, Winnetka, Ill

BRINKER, Harry A. (*M* 1934) 2521 University Ave, Kalamazoo, Mich.

BRINTON, Joseph W. (*M* 1920) Mgr, Boston Dist (for mail) American Blower Corp, 1003 Statler Bldg, Boston, and 42 Gleason St, West Medford, Mass

BRISSETTE, Leo A. (*M* 1930) Treas (for mail) Trask Heating Co, 4 Merrimac St., Boston, and 168 Florence St., Melrose, Mass.

BRITTAIN, Alfred, Jr. (*M* 1938) Engr. Weathermakers (Canada) Ltd, 593 Adelaide St., and (for mail) 138 Wheeler Ave., Toronto, Ont, Canada

BROCHA, John F. (*M* 1936) Buyer of Plbg. and Htg, Montgomery Ward & Co, 619 W. Chicago Ave., and (for mail) 5475 Hirsch St., Chicago, Ill.

BROCKINTON, C. E. (*A* 1937) Sales Engr. (for mail) Advanced Refrigeration, Inc., 350 Peachtree St, and 756 Elkmont Dr., N. E., Atlanta, Ga.

BRODERICK, Edwin L.* (*M* 1933) Research Asst. in Mech. Engrg. (for mail) University of Illinois, 213 M. E. Lab., Urbana, and 909 S. First St., Champaign, Ill.

BRODNAX, George H., Jr. (*M* 1938) Htg. Engr. (for mail) Georgia Power Co, Electric Bldg., and 1564 Westwood Ave, S. W., Atlanta, Ga.

BROKAW, George K. (*S* 1938) Student, University of California, and (for mail) 2634 B College Ave., Berkeley, Calif

BRONSON, Carlos E.* (*M* 1919) Chief Mech. Engr, Kewanee Boiler Corp., Kewanee, Ill.

BROOKE, Irving E. (*M* 1937) Consulting Engr. (for mail) 189 W. Madison St, Chicago, and 830 Keystone Ave, River Forest, Ill

BROOM. Benjamin A. (*M* 1914) Sales Promotion Engr, Weil-McLain Co, 641 West Lake St, and (for mail) 1534 Fargo Ave, Chicago, Ill.

BROOME, Joseph H. (*A* 1936) Sales Engr, Minneapolis-Honeywell Regulator Co, 604 Central Ave, East Orange, and (for mail) 89 Edgemont Rd., Montclair, N. J.

BROWN, Alfred P. (*M* 1927) Vice-Pres (for mail) Reynolds Corp, 1400 Wabansia Ave., Chicago, and 439 Maple St, Winnetka, Ill.

BROWN, Aubrey I.* (*M* 1923) Prof of Htg and Vtg (for mail) Ohio State University, and 169 Richards Rd, Columbus, O.

BROWN, David (*M* 1936) Owner (for mail) 67 Cooper Square, and 54 West 174th St., New York, N Y.

BROWN, Foskett* (*M* 1926) Pres (for mail) Gray & Dudley Co, 222 Third Ave, N., and 2314 West End Ave, Nashville, Tenn.

BROWN, John S., Jr. (*J* 1937) Sales Engr., Smith Distributing Co, and (for mail) 2230 Talbott Ave, Louisville, Ky.

BROWN, Joy S. (*M* 1938) Mgr, Air Cond & Refrig Dept (for mail) J P Baldwin Co, 1304 W Washington Blvd, and 5706 E. Circle Ave., Chicago, Ill

BROWN, Mack D. (*M* 1938; *J* 1936) Mech Engr., Htg and Vtg. (for mail) Northup & O'Brien, Archts, 602-03 Reynolds Bldg, and 915 East 21st St, Winston-Salem, N C.

BROWN, Maurice W. (*J* 1938) Sales Engr (for mail) American Blower Corp., 619 Mercantile Bldg, and 1201 E 7th, Dallas, Tex.

BROWN, Tom (*M* 1930) Vice-Pres & Gen Mgr (for mail) Autovent Fan & Blower Co, 1805-27 N Kostner Ave, and 5325 N. Laramie Ave., Chicago, Ill

BROWN, William H. (*A* 1923) Mgr., Brown Bros, Inc, 3015 North 22nd St, Milwaukee, Wis.

BROWN, W. Maynard (*A* 1930) Warren Webster & Co, 17th and Federal Sts, Camden, N. J.

BROWNE, Alfred L. (*M* 1923) 253 Highland Ave, South Orange, N. J.

BRUNETT, Adrian L. (*M* 1923) Mech. Engr., U. S Supervising Architects Office, Treasury Dept, Washington, D C, and (for mail) P. O. Box 36, Rockville, Md

BRUST, Otto (*M* 1930) Consulting Engr., Wilmersdorfer Strasse 95, Berlin-Charlottenburg 4, Germany.

BRYANT, Alice Gertrude, A.B., M.D., F.A.C.S., AS-E.E. (*Life Member*; *M* 1921) Otologist and Laryngologist, Physician and Surgeon, 405 Marlborough St, Boston, Mass

BRYANT, Percy J. (*M* 1915) Chief Engr. (for mail) Prudential Insurance Co. of America, 763 Broad St, Newark, and 754 Belvidere Ave., Westfield, N. J.

BUCK, David T. (*A* 1936) Chairman (for mail) Buck Engineering Co., Ltd., 37-41 Marcy St., and 116 W. Main St, Freehold, N. J.

BUCK, Lucien (*M* 1928) Engr., Proctor & Schwartz, Inc, 7th St. & Tabor Rd., Philadelphia and (for mail) 101 Waverly Rd., Wyncote, Pa.

BUCKERIDGE, Victor L. (*A* 1938) Partner (for mail) H. Buckeridge & Son, 15108 Kerchival Ave., and 601 Fisher Rd., Grosse Pointe, Mich.
BUCKLEY, Malcomb L. (*A* 1939) Estimator & Supt., Phillips-Getschow Co., and (for mail) 4542 Beacon St., Chicago, Ill.
BUENGER, Albert* (*M* 1920; *J* 1917) (Council, 1934-37) Mgr., National Business Dept. (for mail) Delco-Frigidaire Cond. Div., General Motors Sales Corp, 300 Taylor St., and 224 Schantz Ave., Dayton, O.
BUENSOD, Alfred C. (*M* 1918) Pres, Buensod-Stacey Air Conditioning, Inc, 60 East 42nd St., and (for mail) 33 Fifth Ave., New York, N. Y.
BULKELEY, Claude A. (*M* 1923) Consultant & Sales Engr. (for mail) 2045 N Broad St., Philadelphia, Pa., and Newton, N. J
BULLEIT, Charles R. (*M* 1932; *J* 1930), 1811 Bayard Park Drive, Evansville, Ind.
BULLER, Charles R. (*A* 1938) Asst Chief Engr, Oil Burner Div., The Heil Co., 3000 W Montana Ave., and (for mail) 2650 S. Shore Drive, Milwaukee, Wis
BULLOCK, Howard H. (*A* 1933) Commerical Engr. (for mail) General Electric Co, 212 No Vignes St, Los Angeles, and 2442 Cudahy St, Huntington Park, Calif
BULLOCK, Thomas A. (*M* 1930) 35 Everett St., Arlington, Mass
BUR, Julien R. C. (*A* 1936; *J* 1931) Chief Engr (for mail) Bur & Co, 10 rue du Chapeau Rouge, and 1 Place Francois Rude, Dijon, France
BURCH, Laurence A. (*M* 1934) Sales Mgr, R L. Deppmann Co., 957 Holden Ave, Detroit, and (for mail) 78 Amherst Rd., Pleasant Ridge, Royal Oak, Mich.
BURKHART, Elder M. (*J* 1935) Sales Engr, Overly Mfg Co, 574 W. Otterman St, and (for mail) 529 E Pittsburgh St, Greensburg, Pa
BURNAM, C. M., Jr. (*M* 1938, *A* 1937) Engrg Editor (for mail) Heating, Piping and Air Conditioning, 6 N Michigan Ave, and 10563 S Hale Ave, Chicago, Ill.
BURNETT, Earle S. (*M* 1920) Senior Mech Engr, Petroleum and Natural Gas Div, U S Bureau of Mines, Amarillo Helium Plant, P O. Box 2025, and (for mail) 4223 W. Eleventh Ave, Amarillo, Tex
BURNS, Edward J. (*M* 1923) Harris Bros Plumbing Co, 217 W Lake St, and (for mail) 4716 Aldrich Ave, Minneapolis, Minn
BURNS, John R. (*J* 1936; *S* 1933) Htg Dept, Crane Co, 279 Madison Ave, New York, N Y and (for mail) 504 N Main St, Wallingford, Conn.
BURR, Griffith C. (*M* 1937) Engr, Modern Automatic Heat Co, 14 Wood Lane, Hyde Park, N Y.
BURR, Kimball (*A* 1936) 320 Crocker St, Los Angeles, Calif
BURRITT, Charles G. (*A* 1916) Mgr, Minneapolis Office (for mail) Johnson Service Co, 922 2nd Ave, S, and Minneapolis Athletic Club, Minneapolis, Minn.
BUSHNELL, Carl D. (*A* 1921) Pres (for mail) The Bushnell Machinery Co, 311 Ross St., Pittsburgh, and 94 Pilgrim Rd, Rosslyn Farms Carnegie, Pa.
BUSSE, Herbert (*M* 1938) Chief Engr, Fisher & Co, Fisher Bldg Div., W. Grand Blvd., and (for mail) 16771 Burgess, Detroit, Mich
BUTLER, Peter D. (*M* 1922) Salesman, U. S. Radiator Corp, Detroit, Mich, and (for mail) 127 Edgewater Rd, Cliffside Park, N. J
BUTT, Roderick E. W. (*A* 1936, *J* 1930) Air Cond. Mgr., Refrigeration & Allied Products, Ltd., 92 Buckingham Palace Rd, and (for mail) 605 Beatty House, Dolphin Square, London, S W. 1., England
BYRD, Tom (*A* 1936) Market Development Div. (for mail) The American Rolling Mill Co, and 2403 Fleming Rd, Middletown, O.
BYSOM, Leslie L. (*M* 1915) Mech. Engr., Puget Sound, Navy Yard, P.S N Y., Public Wks. Dept, and (for mail) 1214 8th St, Bremerton, Wash

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CABOT, Mathew A. (*J* 1937) Mech. Engr. (for mail) College of Engineering, University of Kentucky, and 316 Mentelle Park, Lexington, Ky.
CADY, Edward F. (*J* 1937) Engr., 1003 Euclid Ave., Syracuse, N. Y.
CAIRNS, John H. (*A* 1936) Asst. Sales Engr., Frigidaire Corp, and (for mail) 127 Scarboro Rd., Toronto, Ont, Canada
CALDWELL, Arthur C. (*M* 1930) Engr and Estimator, P Gormly Co, 155 N Tenth St, and (for mail) 550 South 48th St, Philadelphia, Pa
CALEB, David (*M* 1923) Engr. (for mail) Kansas City Power & Light Co, 1330 Baltimore Ave., and 141 Spruce St, Kansas City, Mo
CALL, Joseph (*M* 1938; *J* 1936) Air Cond. Engr, Elliott-Lewis Co, 2518 N Broad St, and (for mail) 669 Jamestown St, Roxborough, Philadelphia, Pa
CALLAHAN, Peter J. (*M* 1934) Inspecting Engr, Central Hanover Bank & Trust Co, 60 Broadway, New York, and (for mail) 4057 Amboy Rd, Great Kills, Staten Island, N Y
CALVER, Robert W. (*A* 1937) Prop (for mail) P. O. Box 832, and 77 Queen St, Kirkland Lake, Ont, Canada
CAMERON, Robert T. (*S* 1938) Asst, Alex Cameron, Inc, 5035 Forbes St, Pittsburgh, Pa, and (for mail) 98 Herrick Rd, Southampton, N Y
CAMERON, William R. (*A* 1936) Dist Mgr, L J Mueller Furnace Co, Milwaukee, Wis, and (for mail) 3337 Highland Ave, Kansas City, Mo.
CAMPBELL, Alfred Q., Jr. (*J* 1933) Engr, E. K Campbell Heating Co., Kansas City, Mo, and (for mail) 1083 Meriwether Ave, Memphis, Tenn
CAMPBELL, Andy O. (*J* 1939) Engr (for mail) Oklahoma Gas & Electric Co, Room 408, and 2749 N W. 21st St., Oklahoma City, Okla
CAMPBELL, Bowen (*M* 1938) Engr (for mail) Campbell Heating Co, 3127 Dean Ave, and 2404 E 29th, Des Moines, Ia
CAMPBELL, Everett K* (*M* 1920) (Council, 1931-1933) Pres (for mail) E K Campbell Heating Co, 2445 Charlotte St, and 3717 Harrison, Kansas City, Mo
CAMPBELL, E. Kirker, Jr. (*M* 1938; *J* 1930) Secy, E K Campbell Heating Co, 3420 Haynie, Dallas, Tex
CAMPBELL, Frank B. (*A* 1927) Sales Engr, American Radiator Co, Old Hermitage Rd, Richmond, Va
CAMPBELL, George S. (*J* 1937) Sales Engr, John Bouchard & Sons, Nashville, and (for mail) 1906 Ivy St, Chattanooga, Tenn
CAMPBELL, Ralph L. (*A* 1937) Sales Engr, Chrysler Airtemp, 1316 Nicollet Ave, S, and (for mail) 5241 Knox Ave, S, Minneapolis, Minn
CAMPBELL, Robert E. (*J* 1935, *S* 1934) Sales Engr, Consolidated Air Conditioning, 114 East 32nd St, New York, and (for mail) 3520 Newkirk Ave, Brooklyn, N Y.
CAMPBELL, Thomas F. (*M* 1928) (for mail) T F Campbell Co, 1013 Penn Ave, and R D No 1, Wilkinsburg, Pa
CANDEE, Bertram C. (*M* 1933) Partner, Beman & Candee, 374 Delaware Ave, Buffalo, and (for mail) 19 Tremont Ave, Kenmore, N Y
CANON, Herbert A. (*A* 1938) Mgr, Carrier Dept. (for mail) Dravo Corp, 300 Penn Ave, and 604 Olympia Rd, Pittsburgh, Pa
CAPLE, Ira (*S* 1938) Sr in Mech Engrg, University of Minnesota, and (for mail) 318 Harvard S E, Minneapolis, Minn
CAPPS, Edgar Lee (*A* 1937) (for mail) Capps & Sutherland, 619 W. 35th St, and 619 Pennsylvania Ave, Norfolk, Va
CARBONE, James H. (*M* 1937) Htg-Vtg. Inspector, City of New York, New York, and (for mail) 121-13 198th St, St Albans, L I, N Y
CAREY, James A. (*M* 1928) Carrier Corp, Syracuse, N Y., and (for mail) Villa Nova, Pa.

CAREY, Paul C. (*M* 1930) Consulting Engr (for mail) Runyon & Carey, 33 Fulton St, Newark, and 31 Claremont Drive, Maplewood, N J

CARLE, William E. (*M* 1926) Pres (for mail) Carle-Boehling Co, Inc, 1641 W Broad St, and 2220 Floyd Ave, Richmond, Va

CARLOCK, Marion F. (*M* 1936) Dist Repr, American Foundry & Furnace Co, and (for mail) 505 Henry, Alton, Ill.

CARLSON, C. O. (*A* 1937) Owner (for mail) C O Carlson Htg. Co, 1627 Washington Ave, N., and 1806 Thomas Ave, N, Minneapolis, Minn

CARLSON, Conrad V. (*J* 1937) Air Cond Engr (for mail) Realty Syndicate, 1321 Sharp Bldg, Lincoln, and Axtell, Nebr.

CARLSON, Everett E. (*M* 1932; *A* 1929) Branch Mgr. (for mail) The Powers Regulator Co, 1010 Louderman Bldg, and 6652 Washington Ave, St Louis, Mo

CARNAHAN, John H. (*J* 1937) Design Engrg Dept, Oklahoma Gas and Electric Co, 321 N Harvey St, and (for mail) 3116 N W. 26th St, Oklahoma City, Okla

CARON, Hector (*A* 1938) Mgr-Owner (for mail) Heating & Air Conditioning, 421 S. 3rd St, Rochelle, Ill

CARPENTER, Raymond D. (*S* 1938) California Polytechnic, San Luis Obispo, and (for mail) 534 Bay St, Santa Cruz, Calif

CARPENTER, R. H. (*M* 1921) (Council, 1930-1935) Mgr, New York Office (for mail) Nash Engineering Co, Graybar Bldg, 420 Lexington Ave, New York, and 20 Jefferson Ave, White Plains, N Y

CARR, Maurice L.* (*M* 1931) Director, Pittsburgh Testing Laboratory, Stevenson & Locust Sts, Pittsburgh, Pa

CARRIER, Earl G. (*M* 1936 *J* 1929) Supvr Engr, Carrier Corp, Philadelphia Savings Fund Bldg, Philadelphia, Pa, and (for mail) 228 Robineau Rd, Syracuse, N Y

CARRIER, Willis H.* (*M* 1913) (*Presidential Member*) (Pres, 1931, 1st Vice-Pres, 1930, 2nd Vice-Pres, 1929, Council, 1923-1932) Chairman of the Board (for mail) Carrier Corp, and 2570 Valley Drive, Syracuse, N Y

CARROLL, Arthur F. (*M* 1938) Development Engr, Automatic Products Co, 2450 N 32nd St, and (for mail) 5525 Brooklyn Pl, Milwaukee Wis

CARROLL, William M. (*J* 1938) Sales Engr (for mail) Tom Dolan Heating Co, 614 W Grand, and 908 East Drive, Oklahoma City, Okla

CARTER, Alexander W. (*J* 1936) Htg Engr (for mail) Monarch Brass Mfg Co, Ltd, 71 Browns Ave, and 2178 Queen St, E, Toronto, Ont, Canada

CARTER, Doctor (*M* 1934) Consulting Engr, 34 Cunningham Ave, St Albans, Herts England

CARTER, John H. (*M* 1936) Special Repr (for mail) Frick Co, 100 N Broadway, St Louis, and 529 Atlanta Ave, Wesbter Groves, Mo

CARY, Edward B. (*M* 1935) Partner (for mail) John Paul Jones Cary & Millar, Consulting Engrs, 448 Terminal Tower, Cleveland, and Chillicothe Rd, Aurora, O

CASE. Delbert V. (*M* 1937) Owner, Case Engineering Co, 428 Dwight Bldg, and (for mail) 2005 East 33rd St., Kansas City, Mo

CASE, Roy H. (*A* 1936) Resident Mgr (for mail) 417 Central Bldg, and 3322 Hunter Blvd, Seattle, Wash.

CASE, Walter G. (*A* 1930) Asst. Mgr, Ideal Boilers & Radiators, Ltd., Ideal House, Great Marlborough St, London, W 1, and (for mail) 66 The Ridgeway, Kenton, Harrow, Middlesex, England

CASEY, Byron L. (*M* 1921) Sales Engr (for mail) Ilg Electric Ventilating Co., 182 N. LaSalle St, Chicago, and 307 Vine Ave, Park Ridge, Ill

CASKEY, Luther H., Jr. (*S* 1938) Student (for mail) Carnegie Institute of Technology, Box 310, Pittsburgh, Pa, and 513 N Queen St, Martinsburg, W Va

CASPERD, Henry W. H. (*A* 1938; *J* 1930) Engr., Carrier Co, Ltd., 24 Buckingham Gate, London, and (for mail) 21 Robin Hood Lane, Sutton, Surrey, England.

CASSELL, John D.* (*Life Member*; *M* 1913) Retired, 2008 Walnut St., Philadelphia, Pa.

CASSELL, William L. (*M* 1936) Owner (for mail) William L Cassell Mechanical Engr., 2501 Telephone Bldg, Kansas City, and R. F. D. No. 6, Independence, Mo

CAWBY, Elmer L. (*A* 1938; *J* 1935) Sales Engr. (for mail) Carrier Corp, 748 E. Washington Blvd, and 2315 S Flower St, Los Angeles, Calif.

CHALMERS, Charles H. (*M* 1925) Gen. Mgr. (for mail) Chalmers Oil Burner Co, 1234 Central Ave, and 523-7th St, S. E, Minneapolis, Minn.

CHAMBERS, Fred W. (*M* 1936) Pres. (for mail) F W. Chambers & Co, Ltd, 96 Bloor St, West, and 122 Garfield Ave, Toronto, Ont, Canada

CHAMPLIN, Robert C. (*A* 1938) Assistant to Air Cond Engr, Timken Silent Automatic Div., 100-400 Clark Ave, Detroit, and (for mail) 152 Cortland Ave, Highland Park, Mich.

CHAPIN, C. Graham (*M* 1933) Treas (for mail) Hopson & Chapin Mfg. Co., 231 State St., and 66 Faire Harbour Place, New London, Conn.

CHAPIN, Harvey G. (*M* 1935) Sales Engr (for mail) Westerlin & Campbell Co., 1113-15 Cornelia Ave, and 8151 Ingleside Ave, Chicago, Ill.

CHAPMAN, William A., Jr. (*M* 1936) Sales Planning Div (for mail) Delco-Frigidaire Cond Div, General Motors Sales Corp, 300 Taylor St, and 2515 Shafor Blvd, Dayton, O

CHARLES, Paul L. (*M* 1938) Mgr (for mail) Walsh & Charles, 406 Tribune Bldg., and 145 Ash St, Winnipeg, Man, Canada

CHARLES, Thomas J. (*M* 1934) Engr, 209 West 38th St, New York, and (for mail) 175 Marine Ave, Brooklyn, N Y

CHARLET, Louis W. (*M* 1934) Branch Mgr (for mail) Kewanee Boiler Corp, 37 W 39th St, New York, and 427 Rich Ave, Mt Vernon, N. Y.

CHARTERS William A. (*A* 1937) Salesman and Field Engr, Canada Foundries & Forgings, Ltd, Brockville, and (for mail) 57 Newton Ave., Hamilton, Ont, Canada

CHASE, Arthur M., Jr. (*M* 1938) Sales Engr. (for mail) York Ice Machinery Corp, P. O Box 359, and 5124 Ennis St, Houston, Tex.

CHASE, Chauncey L. (*M* 1931) Partner (for mail) Edward E Ashley, Consulting Engr, 10 East 40th St, New York, and 8829 Fort Hamilton Pkwy, Brooklyn, N Y.

CHASE, Louis R (*M* 1938, *J* 1931) Mgr Air Cond Dept (for mail) The Carter-Waters Corp., 2440 Pennway, and 4915 Bell, Kansas City, Mo

CHEATWOOD, William H. (*J* 1937) Commercial Engr (for mail) Air-Rite Corp, 3123 Holmes St, and 1815 Grand Ave, Dallas, Tex

CHEESEMAN, Evans W. (*J* 1937; *S* 1934) Engr, Perfection Stove Co, 7609 Platt Ave, and (for mail) 2200 Prospect Ave, Cleveland, O.

CHEN, Sarcey T. (*M* 1936) Vice-Pres and Gen Mgr (for mail) American Engineering Corp, 989 Bubbling Well Rd, and 45/24 Great Western Rd, Shanghai, China

CHENEVERT, J. Georges (*M* 1938) Consulting Engr (for mail) Arthur Surveyer & Co, Consulting Engrs, Room 1003-1010 St Catherine St, W, Montreal, and 536 Outremont Ave, Outremont, P. Q, Canada

CHENOWETH, Dale M. (*J* 1938; *S* 1936) 1000 Washington St, South Braintree, Mass

CHERNE, Realto E. (*M* 1938; *J* 1929) Engrg Supervisor, Carrier Corp, and (for mail) 606 Charmouth Drive, Syracuse, N. Y.

CHERRY, Lester A.* (*M* 1921) Consulting Engr. (for mail) Industrial Planning Corp, 271 Delaware Ave, and 155 Euclid Ave, Buffalo, N. Y.

CHERRY, Virgil H.* (*M* 1937) Instructor, University of California, Dept of Mech Engrg, and (for mail) 1269 Hearst Bldg, Berkeley, Calif.

CHESTER, Thomas* (*M* 1917) Consulting Engr, c/o Davidson & Co., Ltd, Central House, Kingsway, London, England

CHEYNEY, Charles C. (*A* 1913) Asst. Sales Mgr. (for mail) Buffalo Forge Co., 490 Broadway, and 255 Lincoln Parkway, Buffalo, N. Y.
CHILDS, Lewis A. (*M* 1938) Dist. Sales Mgr. (for mail) Clarage Fan Co., 520 Commercial Trust Bldg., and 1320 Foulkrod St., Philadelphia, Pa.
CHRISTENSON, Harry (*A* 1931) Partner (for mail) Hunter-Prell Co, 38 S. Madison St., and 121 Sunset Blvd, Battle Creek, Mich.
CHRISTMAN, William F. (*A* 1932; *J* 1931) Engr. (for mail) Kroeschell Engineering Co., 215 W. Ontario St., and 5649 Artesian Ave., Chicago, Ill.
CHRISTOPHERSON, Andrew E. (*M* 1935) Engr.-Custodian (for mail) Board of Education, Spalding School, 1628 Washington Blvd., and 2923 N. Kilpatrick Ave., Chicago, Ill
CHRONE, Robert E. (*M* 1938) Chief Engr, R. F. Taylor, Consulting Engr., 909 Bankers Mortgage Bldg., Houston, Tex.
CHURCH, H. J. (*M* 1922) Mgr. (for mail) Darling Brothers, Ltd., 137 Wellington St., W., Toronto, and 358 Main St., N., Weston, Ont, Canada.
CITRON, Daniel J. (*S* 1938) 2055 Ryer Ave., New York, N. Y.
CLAR, Robert, Jr. (*A* 1938) Sales Engr. (for mail) United States Radiator Corp, 127 Campbell Ave., and Lee Crest Apartments, 2nd Blvd. at Blaine, Detroit, Mich.
CLARE, Fulton W. (*M* 1927) 935 Plymouth Rd, N. E, Atlanta, Ga
CLARK, E. Harold (*M* 1922) Manufacturer's Agent, 600 Michigan Theatre Bldg., and (for mail) 2539 Lakewood, Detroit, Mich
CLARK, Lynn W. (*A* 1938) Engr. and Salesman (for mail) Hall-Neal Furnace Co., and 737 West 32nd St., Indianapolis, Ind.
CLARKSON, John R. (*S* 1938) David Ranken Jr. School of Mech. Trade, Air Cond, Dept., 4431 Tinney Ave., and (for mail) 4620-A Newberry Terrace, St. Louis, Mo.
CLAY, Wharton (*A* 1938) Secy (for mail) National Mineral Wool Assn, 1270 Sixth Ave, New York, and 127 S Broadway, Nyack, N Y
CLEGG, Carl (*M* 1922) Dist. Mgr. (for mail) American Blower Corp, 311 Mutual Bldg., and 3513 Gillham Rd, Kansas City, Mo
CLEVELAND, Clyde C. (*A* 1936) Htg Engr, Johnson & Cleveland, 192 Main St., and (for mail) 64 E. Main St., Bradford, Pa.
CLIFTON, John A. (*A* 1938) Salesman, Warden King, Ltd., 299 Adelaide St., W., and (for mail) 369 Belsize Drive, Toronto, Ont, Canada.
CLINE, Edward A. (*M* 1937) Consultant on Air and Water Conditioning, Room 512, Architects Bldg, 816 West 5th, Los Angeles, Calif
CLOSE, Paul D.* (*M* 1928) (for mail) The Celotex Corp., 919 N. Michigan Ave, Chicago, and 564 Meadow Rd., Winnetka, Ill
CLOSE, Robert (*M* 1938) Chief Air Cond. Engr., National Broadcasting Co., 30 Rockefeller Plaza, New York, N. Y. and (for mail) 199 Ames Ave, Leonia, N J.
CLOSNER, J. J. (*J* 1938) Sales Engr (for mail) Robischung-Kiesling, Inc., 4848 Main St., and 4435 Jefferson, Houston, Tex.
COCHRAN, Charles C. (*A* 1935) Asst Sales Mgr., Chicago Office, Minneapolis-Honeywell Regulator Co., 433 East Erie, Chicago, and (for mail) 840 S Clifton Ave, Park Ridge, Ill
COCHRAN, Lex H. (*M* 1934) Sales Mgr, Western Div. (for mail) American Blower Corp., 625 Market St., and 130 Camino Del Mar, San Francisco, Calif.
COCHRAN, William B. (*J* 1936; *S* 1935) Air Cond Engr. (for mail) Cochran Air Conditioning Co. (Westinghouse) 1303 Lamar Ave, and 3316 Telephone Rd, Box 16, Houston, Tex.
COCKINS, William W. (*J* 1937) Sales Engr (for mail) The Trane Co, 1129 Folsom St, San Francisco, and 576 The Alameda, Berkeley, Calif.
CODY, Henry C. (*M* 1936) Sales Engr, Pierce Butler Radiator Corp., 19th & Glenwood Ave, and (for mail) 7336 North 21st St., Philadelphia, Pa.
COGHLAN, Sherman F. (*A* 1937) Metropolitan Water Dist of Southern Calif, 306 W. Third St., Los Angeles, and (for mail) 414 Ninth St., Santa Monica, Calif
COHAGEN, Chandler C. (*M* 1919) Archt., 211 Hedden Bldg. (for mail) Box 2100, and 235 Avenue G., Billings, Mont.
COHEN, Harry (*J* 1937; *S* 1936) 3630 East 140th St, Cleveland, O.
COHEN, Philip (*M* 1932) Dist. Mgr. (for mail) B. F. Sturtevant Co., 401 E Ohio Gas Bldg., and 7100 Euclid Ave, Suite No 6, Cleveland O.
COLBY, Clyde W. (*M* 1915) Vice-Pres. and Gen. Mgr., Air Devices Corp, 70 Britannia St., and (for mail) 167 Buckingham St, Meridan, Conn.
COLCLOUGH, Otho T. (*A* 1933) Custodian, American Legation, and (for mail) 726 Parkdale Ave, Ottawa, Ont., Canada.
COLE, C. Boynton (*J* 1937) Chief Engr, Air Cond. Dept., The Murray Co., Howell Mill Rd., P. O Box 1517, and (for mail) 1843 Flagler Ave., N E., Atlanta, Ga
COLE, Grant E. (*A* 1925) Vice-Pres. & Gen. Mgr., Trane Co. of Canada, Ltd, 4 Mowat Ave., Toronto, Ont., Canada.
COLEMAN, John B. (*M* 1920) Chief Engr (for mail) Grinnell Co, Inc., 260 West Exchange St., and 237 Cole Ave, Providence, R I.
COLFORD, John (*A* 1937) Pres, John Colford, Ltd, 2007 Guy St., Montreal, and (for mail) 51 Upper Bellevue Ave, Westmount, P Q, Canada
COLLE, Samuel S. (*A* 1938) Engr & Owner (for mail) Air Conditioning Engineering Co, 361 Youville Square, and 1489 Atwater Ave., Montreal, P. Q, Canada
COLLIER, William I. (*M* 1921) Pres. (for mail) W I Collier & Co., 522 Park Ave, Baltimore, and Ellicott St, Ellicott City, Md.
COLLINS, John F. S., Jr. (*M* 1933) Supervisor of Steam Utilization (for mail) Allegheny County Steam Heating Co, Philadelphia Co Bldg, 435 Sixth Ave, and 827 N. Euclid Ave., Pittsburgh, Pa
COLMENARES, Gaspar Vizoso (*A* 1938) Vice-Pres & Gen Mgr (for mail) Castel-Vizo, Refrigeracion y Aire Acondicionado S A, Obrapia 407, P O Box 210, and 19th St, No. 1001, Vedado, Havana, Cuba
COMB, Fred R., Jr. (*J* 1938; *S* 1937) Sales Engr., Delco-Fridigaire Conditioning Div, 2446 University Ave, St Paul, and (for mail) 2425 Bryant Ave, S, Minneapolis, Minn
COMPTON, Warren E. (*J* 1936; *S* 1935) Engr., 904 W Green St, Urbana, Ill.
COMSTOCK, Glen M. (*A* 1926) Sales Repr Engrg (for mail) L J Wing Mfg Co, 604 Chamber of Commerce Bldg, Pittsburgh, and 154 College Ave, Beaver, Pa
CONATY, Bernard M. (*M* 1935) Sales Mgr (for mail) American District Steam Co, North Tonawanda, and 1306 Delaware Ave, Buffalo, N Y
CONE, William E. (*J* 1937) Air Cond Engr (for mail) Shook & Fletcher Supply Co, 1814 1st Ave., N., and 1037 10th Ave, S., Birmingham, Ala
CONNELL, Harold (*M* 1935) Engr, Armo Cooling & Ventilating Co, 30 West 15th St., New York, and (for mail) 163 Arlington Ave, Mariners Harbor, S. I, N Y
CONNELL, Richard F. (*M* 1916) Mgr., Capitol Testing Laboratory (for mail) U S. Radiator Corp, 1056 National Bank Bldg, and 2970 Burlingame, Detroit, Mich
CONNER, Raymond M. (*M* 1931) Dir Testing Laboratories (for mail) American Gas Association, 1032 East 62nd St, Cleveland, and 271 East 216th St, Euclid, O
CONRAD, Roy (*M* 1935) Salesman, Carrier Corp, 748 E. Washington, Los Angeles, Calif, and (for mail) 3416 Colfax "B", Denver, Colo
CONROY, Martin J. (*A* 1938) Dealer, Link-Belt Co, Stoker Div, Lycoming at Broad St, Philadelphia, and (for mail) 122 Kent Rd, Upper Darby, Pa.

CONSTANCE, John D. (*J* 1937) Draftsman, York Ice Machinery Corp., 42nd St & 2nd Ave, Brooklyn, N. Y, and (for mail) 407 27th St., North Bergen, N. J.
CONSTANT, Earl S. (*J* 1935) Air Cond. Sales Engr., P H. Hensarling Co., 713 Bankers Mortgage Bldg., and (for mail) 2802 Bagby St, Houston, Tex
COOK, Arthur L. (*A* 1938) Engr, Power Plant, A. &. M. College of Texas, and (for mail) P. O. Box 248, College Station, Tex.
COOK, Benjamin F. (*M* 1920) Consulting Engr, 114 W. Tenth St. Bldg, Kansas City, and (for mail) 1720 Overton Ave, Independence, Mo
COOK, George E. (*A* 1937) Pres (for mail) AirConditioning, Inc, 2324 Hampden Ave, St. Paul, and 2115 Blaisdell Ave., Minneapolis, Minn.
COOK, H. Dale (*A* 1938) Temp Control Engr. (for mail) R L. Deppmann Co, 957 Holden St, Detroit, and 73 East 10th St., Holland, Mich.
COOK, Ralph P. (*M* 1930) Asst Supt, Engrg & Maintenance Dept. in charge of Engrg Div. (for mail) Eastman Kodak Co, Kodak Park, and 663 Seneca Parkway, Rochester, N. Y
COOKE, Thomas C. (*J* 1937) Htg & Air Cond Engr. (for mail) Tomlinson Co, Inc, 400-402 E Peabody St, P. O. Box 217, and 1118½ Eighth St, Durham, N C
COOLEY, Edgerton C. (*M* 1937) Mfrs Agent, Chicago Pump Co and Ross Heater & Mfg Co (for mail) 625 Market St, San Francisco, and P. O Box 789 B Route 1, Los Altos, Calif
COOMBE, James (*A* 1932) Vice-Pres (for mail) William Powell Co, 2525 Spring Grove Ave, and 2363 Grandin Rd, Cincinnati, O.
COON, Thurlow E. (*M* 1916) Pres (for mail) The Coon-DeVisser Co, 2051 W. Lafayette, and 826 Edison Ave, Detroit, Mich
COOPER, Albert W. (*M* 1935) Branch Mgr (for mail) Johnson Service Co, 1230 California St, and 639 Cook St, Denver, Colo
COOPER, Dale S. (*M* 1938, *A* 1937) Chief Engr. & Vice-Pres (for mail) Air Conditioning Co, 4304 Main St, and 6528 Rutgers, Houston, Tex
COOPER, John W. (*M* 1932, *A* 1925, *J* 1921) Repr (for mail) Buffalo Forge Co, 1598 Arcade Bldg, St Louis, and 612 Hawbrook Drive, Kirkwood, Mo
COOPER, Tom E. (*J* 1937, *S* 1936) 309 West 50th St, Minneapolis, Minn
COOPER, William B. (*J* 1937) Head of Prefabricated Filtering Dept, Mueller Furnace Co, 2005 Oklahoma Ave, and (for mail) 3825 N Newhall, Milwaukee, Wis
COPPERUD, Edmund R. (*J* 1933) Asst Mgr. (for mail) Minneapolis Plumbing Co, 1420 Nicollet Ave, and 17 West 25th St, Minneapolis, Minn
COREY, George R. (*M* 1936) Engr, 40 Sias Lane, Milton, Mass
CORNWALL, Charles C. (*J* 1935) Research Engr, The Bahnson Co, 1001 S Marshall St, and (for mail) 473 Carolina Circle, Winston-Salem, N C
CORNWALL, George I. (*M* 1919) Sales Engr., Burnham Boiler Corp, Elizabeth, N J.
CORRAO, Joseph (*A* 1936; *J* 1933) Engr., City of San Francisco, Dept of Works, Engrg Dept, City Hall, and (for mail) 854 31st Ave, San Francisco, Calif
CORRIGAN, James A. (*J* 1935, *S* 1930) Engr (for mail) Corrigan Co, 2501 St Louis Ave, and 6130 McPherson Ave, St. Louis, Mo
COST, George W. (*S* 1938) Johnson Service Co, 1238 Brighton Rd, and (for mail) 5210 Forbes St, Pittsburgh, Pa
COVER, E. B. (*M* 1937) Sales Engr, York Ice Machinery Corp, 115 S 11th St, St Louis, and (for mail) 3252 Waverly, East St Louis, Ill
COVER, Richard R. (*A* 1936) Engr, Carrier Corp, 4th & Channing, N E, and (for mail) 1302 Gallatin St, N W, Washington, D C
COWARD, Charles W. (*M* 1935) Pres (for mail) Coward Engineering Co, 411 Cooper St, Camden, and 812 Lincoln Ave, Palmyra, N J.
COWELL, Robert J. (*M* 1922) Consulting Engr., 769 Seventh Ave, New York, N. Y.
COX, Harrison F. (*A* 1930) Htg. & Air Cond., 243 Carroll St, Paterson, N. J.
COX, Philip E. (*M* 1938) Engr., Societe des Cine-Theatres d'Indochine, and (for mail) 50 Boulevard Dongkhank, Hanoi, Tonkin, French Indo China.
COX, Thomas M., Jr. (*J* 1937) Refrigeration and Air Cond, Dept. Head (for mail) Neal & Massy Engrg Co., Ltd., 61 Edward St, and Maravol, Port of Spain, Trinidad, B. W. I.
COX, William W. (*Life Member*; *M* 1923) Pres. and Mgr. (for mail) Heating Service Co, Inc, 326 Columbia St, and 6232 31st Ave, N. E., Seattle, Wash
CRAMER, Wesley G. (*A* 1938) Dist Sales Mgr. (for mail) The Marley Co., 1801 Carew Tower, and 3174 Portsmouth Ave., Cincinnati, O.
CRANE, Robert S. (*M* 1938) Dist Engr. (for mail) Air Cond Frigidaire Div., General Motors Sales Corp, 4 Cummin Station, and 301 32nd Ave., S, Nashville, Tenn
CRANSTON, William E., Jr. (*M* 1931) Vice-Pres. & Gen Mgr (for mail) Thermador Electric Mfg. Co, 2821 E Pico St, Los Angeles, and 240 Hacienda Drive, Arcadia, Calif
CRAWFORD, Arthur C. (*A* 1938) Sales Engr, Potomac Electric Power Co, 10th & E St., N W, and (for mail) 429 Butternut St., N. W, Washington, D C
CRAWFORD, John H., Jr. (*A* 1936; *J* 1930) Air Cond Engr, Hitchen Co, 441 Lexington Ave, New York, N Y, and (for mail) 289 Reynolds Terr, Orange, N. J.
CRIBARI, Hugo E. (*A* 1937) Salesman, American Radiator Co, 40 West 40th St, New York, and (for mail) 468 N. Fulton Ave., Mt. Vernon, N Y.
CRICHTON, Howard C. (*S* 1936) 769 Beaver Ave, Midland, Pa
CRIQUI, A. A.* (*M* 1919) Chief Engr, Htg. & Vtg Dept Buffalo Forge Co, 490 Broadway, Buffalo, and (for mail) 39 St. Johns Ave., Kenmore, N Y
CRONE, Charles E., Jr. (*M* 1922) Secy-Treas. (for mail) Wendt & Crone Co, 2124 N Southport Ave, and 1320 N State St, Chicago, Ill
CRONE, Thomas E. (*Life Member*; *M* 1920) Apt. E-24 Highland Hall, Rye, N Y.
CRONEY, P. Alfred (*M* 1938) Chief of Mech. Section, U S Housing Authority, Interia Bldg, N, Washington, D C, and (for mail) 21 Philadelphia Ave, Takoma Park, Md
CROPPER, Robert O. (*M* 1938) Operating Engr., War Dept, c/o Quartermaster, Fort Knox, and (for mail) Vine Grove, Ky
CROSBY, Edward L. (*M* 1936) Pres. (for mail) Henry Adams Inc, Consulting Engrs., 1263-1269 Calvert Bldg, and 5323 Belleville Ave, Baltimore, Md
CROSS, Freeman G. (*M* 1936) Sales Mgr, Controls Div (for mail) Fulton Sylphon Co, and 31 Nokomis Circle, Knoxville, Tenn.
CROSS, Robert C.* (*M* 1937) Fuel Engr (for mail) Battelle Memorial Institute, 505 King Ave., and 1178 Virginia Ave, Columbus, O
CROSS, Robert E. (*M* 1938, *A* 1931) Dist. Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 271 Columbus Ave, and 68 Kimberly Ave, Springfield, Mass.
CROSS, Roy A. (*A* 1938) Jr. Engr, Buffalo Forge Co, 490 Broadway, Buffalo, and (for mail) 66 Enola Ave, Kenmore, N Y
CROUT, Marvin M. (*M* 1939, *A* 1938) Branch Mgr (for mail) York Ice Machinery Corp., 412 Houston St, and 2392 Hurst Drive, Atlanta, Ga.
CRUMP, Alvin L. (*M* 1937) Sales Engr (for mail) Powers Regulator Co, 2720 Greenview Ave, Chicago, and 2701 Payne St, Evanston, Ill.
CUCCI, Victor J. (*M* 1930) Consulting Engr. (for mail) 347 Madison Ave, New York, and 451-55th St, Brooklyn, N Y.
CULBERT, William P. (*A* 1929) Partner (for mail) Culbert-Whitby Co, 2019 Rittenhouse St., Philadelphia, and 929 Alexander Ave., Drexel Hill, Pa

CULLEN, Augustine G. (*A* 1936) Pres (for mail) Cullen, Inc, 20 L St, S W, Washington, D. C
CULLIN, William W. (*M* 1938) Chief Engr., Home Insulation Div., Johns-Manville Sales Corp., 22 East 40th St, and (for mail) 2995 Botanical Sq, New York, N Y.
CUMMING, Ford J. (*M* 1936) Pres (for mail) Beecher-Cumming, Inc, 820 2nd Ave, S, Minneapolis, and 120 Interlachen Rd, Hopkins, Minn
CUMMING, Robert W. (*M* 1928) Engr & Sales Executive, Sarco Co, Inc, 183 Madison Ave, New York, and (for mail) 81 Alkamont Ave, Scarsdale, N. Y.
CUMMINGS, Carl H. (*A* 1927; *J* 1926) Mgr. (for mail) Industrial Appliance Co. of New England, 110 Arlington St, Boston, and 41 Edgehill Rd, Chestnut Hill, Mass
CUMMINGS, G. J. (*M* 1923) Vice-Pres & Secy (for mail) The Scott Co, 113-10th St, and 2001 Hoover Ave, Oakland, Calif
CUMMINS, George H. (*M* 1919) Dist Mgr, Aerofin Corp, 918 United Artists Bldg, and (for mail) 16210 Ashton Rd, Detroit, Mich
CUMNOCK, H. (*A* 1938) Secy. & Treas (for mail) Little Rock Refrigeration Co, 417 W Capitol Ave, and 609 Rock, Little Rock, Ark
CUNNINGHAM, John S. (*J* 1937, *S* 1935) Htg Engr, Rudy Furnace Co, and (for mail) 311 N Front St, Dowagiac, Mich
CUNNINGHAM, Thomas M. (*M* 1931, *J* 1930) Production Mgr, Carrier Corp, 7-122 Merchandise Mart, Chicago, Ill
CURRIER, Charles H. (*M* 1919) (for mail) Ross Heater & Mfg. Co, Inc, 1407 West Ave, and Park Lane Apts, 33 Gates Circle, Buffalo, N Y.
CURRY, Roger F. (*S* 1938) Template Dept, Curtiss Wright Corp, Robertson, and (for mail) 3142 Sutton Blvd, Maplewood, Mo
CURTICE, Jean M. (*A* 1936) Htg Engr and Dist Mgr, Citizens Utilities Co, 15 W Fourth St, La Junta, Colo
CURTIS, Herbert F. (*A* 1934) Chief Engr (for mail) Henry Furnace & Foundry Co, 3471 East 49th St, Cleveland, and 59 Fourth Ave, Berea, O
CURTIS, Walter A. (*J* 1938, *S* 1936) Sales Engr, B. F Sturtevant Co, 220 Delaware Ave, and (for mail) 37 E Morris Ave, Buffalo, N Y
CUSHING, Charles F. (*M* 1938) Mgr Air Cond Sales, Bryant Heater Co (for mail) 17825 St Clair Ave, Cleveland, and 2204 Edgerton Rd, Cleveland Heights, O
CUTHBERTSON, Merle W. (*A* 1937) Supt Mech Equip and Bldgs, Hardware Mutual Insurance Co, 2344 Nicollet Ave, Minneapolis, and (for mail) 1466 Hague Ave, St Paul, Minn
CUTLER, Joseph A. (*M* 1916) (Council, 1920-1926) Pres and Gen Mgr (for mail) Johnson Service Co, 507 East Michigan St, and 4811 N Lake Drive, Milwaukee, Wis

D

DAFTER, Edwin H. (*M* 1938) Dist. Chief Engr (for mail) Carrier Corp, 12 S 12th St, and 236 Henley Rd, Penn Wynne, Philadelphia, Pa
DAHLSTROM, Godfrey A. (*A* 1927) Htg Sales Engr, Central Supply Co, 312 S Third St, and (for mail) 3721-47th Ave, S, Minneapolis, Minn
DAILEY, James A. (*A* 1920) 31-64-30th St, Astoria, L I, N Y
DAITSH, Abe (*J* 1937) Student Engr, York Ice Machinery Corp, 42nd St & Second Ave., and (for mail) Y M C A, 357 Ninth St., Brooklyn, N Y
DAKIN, Harold W. (*J* 1934) 169 Park Ave, Dalton, Mass.
DALY, Robert E. (*M* 1931) Dir of Engrg (for mail) American Radiator Co, 40 W 40th St, New York, and 270 Bronxville Rd, Bronxville, N Y
DAMBLY, A. Ernest (*M* 1924; *J* 1921) (for mail) 901 Architects Bldg, Philadelphia, Pa., and Harvey Cedars, N J
DANIELSON, E. B. (*A* 1936) Owner-Mgr. (for mail) Home Service Co, 132 West St, and 819 Main St, Russell, Kan.
DANIELSON, Ellsworth H. (*J* 1938) Dist Mgr., Minneapolis-Honeywell Regulator Co., 2831 Center St, Des Moines, Ia
DANIELSON, Lloyd C. (*J* 1938, *S* 1936) Sales Mgr., Home Service Co, 821 Main St, Russell, Kan
DANIELSON, Wilmot A. (*M* 1935) Lt Col, Quartermaster and Constructing Quartermaster, U S Army, Quartermaster Corps, Fort Knox, Ky.
DARBY, Marion H. (*A* 1938, *J* 1930) Chief Engr, Air Cond. Dept (for mail) General Electric, S A, Caixa Postal 109, and Alberto de Campos 299, Apt 6, Rio de Janeiro, Brazil
DARLING, Arthur B. (*A* 1929) Asst. Sales Mgr. (for mail) Darling Bros Ltd, 140 Prince St, Montreal, and 4326 Sherbrooke St, W, Westmount, P. Q, Canada
DARLINGTON, Allan P. (*M* 1930) Sales Engr, Power Apparatus (for mail) American Blower Corp., 6000 Russell St, and 5200 Haverhill, Detroit, Mich
DARTS, John A. (*M* 1919) Kewanee Boiler Co, Inc, 101 Park Ave, New York, N Y
DASING, Emil (*M* 1937) Designing Engr, Sears Roebuck & Co, 925 S Homan Ave, and (for mail) 4729 N Talman Ave, Chicago, Ill
DAUBER, Oscar W. (*M* 1937) Consulting Engr (for mail) 224 S Michigan Ave, Chicago, and 366 Winnetka Ave, Winnetka, Ill
DAUBERT, LeRoy L. (*J* 1937) Branch Mgr, Sidles Co, Airtemp Div, 805 Walnut St, and (for mail) 2315 Grand Ave, Des Moines, Ia
DAUCH, Emil O. (*M* 1921) Secy-Treas (for mail) McCormick Plumbing Supply Co, 1675 Bagley Ave., Detroit, and 729 Bedford Rd., Grosse Pointe, Mich
DAUMAN, Arnold (*S* 1938) 1700 Grand Concourse, New York, N Y
DAVEY, Geoffrey I. (*M* 1937) Consulting Engr, Haskins & Davey, 60-66 Hunter St, Sydney, N S W, Australia
DAVIDSON, John C. (*J* 1936) Air Cond Inspector, City of Minneapolis, 213 City Hall, and (for mail) 4412-46th Ave, S, Minneapolis, Minn
DAVIDSON, L. Clifford (*M* 1927) Associate Dist Mgr (for mail) Buffalo Forge Co, 220 S 16th St, Philadelphia, and 322 Winding Way, Merion, Pa.
DAVIDSON, Philip L (*M* 1924, *J* 1921) Consulting Engr (for mail) 1600 Walnut St, Philadelphia, and Radnor, Pa
DAVIES, George W. (*M* 1918) Mgr, G W Davies & Co, 19 Maclaggan St, Dunedin, C 1 (for mail) P O Box 390, Dunedin, N 2, and Colinswood, Macandrew Bay, New Zealand
DAVIS, Arthur C.* (*M* 1920) Supt of Maintenance, The Port of N Y Authority, 111 Eighth Ave, New York, N Y, and (for mail) 73 Preston St, Ridgefield Park, N J
DAVIS, Arthur F. (*M* 1934) Pres (for mail) Johnson & Davis Plumbing & Heating Co, 2235 Arapahoe St, and 1901 Ivanhoe St, Denver, Colo
DAVIS, Bert C. (*Life Member*, *M* 1904) (Council, 1917) Pres and Treas (for mail) American Warming & Ventilating Co, 317 Pennsylvania Ave, and 603 W. Church St, Elmira, N Y
DAVIS, Calvin R. (*M* 1927) Branch Mgr (for mail) Johnson Service Co, 2328 Locust St, and 7534 Westmoreland Drive, St Louis, Mo
DAVIS, Charles (*M* 1938) Engr (for mail) Rathe Heating Corp, 598 Grand Concourse, and 281 Wadsworth Ave, New York, N Y
DAVIS, Edward J. (*J* 1938) Sales Engr (for mail) Gurney Foundry Co, Ltd, 4 Junction Rd, Toronto, and Lakeview, Ont, Canada
DAVIS, George C. (*J* 1936) Vice-Pres (for mail) Northern Public Service Corp, Ltd, 307 Power Bldg, and 923 Somerset Ave, Winnipeg, Man, Canada
DAVIS, George L., Jr. (*A* 1938) Estimator (for mail) L L McConachie Co, 1003 Maryland Ave., and 1300 Wayburn St., Detroit, Mich
DAVIS, Joseph (*M* 1927; *A* 1926) Owner, Heating & Refrigeration Contractor (for mail) 70 West Chippewa St, and 166 Huntington Ave, Buffalo, N Y

DAVIS, Keith T. (*M* 1937) Engr. Dept (for mail) L. J. Mueller Furnace Co., and 1445 N. 40th, Milwaukee, Wis

DAVIS, Otis E. (*M* 1929; *A* 1925) Sales Engr., Hoffman Specialty Co., Waterbury, Conn, and (for mail) Box 98, Scottsbluff, Nebr.

DAVIS, Rowland G. (*A* 1921) Sales Repr, 887 Nela View Rd, Cleveland Heights, O

DAVISON, Robert L. (*M* 1934) Director of Housing Research (for mail) John B Pierce Foundation, 37 West 39th St, New York, and Meadow Glen Rd, Fort Salonga, L I., N Y

DAWSON, Eugene F. (*M* 1934) Assoc Prof of Mech Engrg (for mail) University of Oklahoma, and 225 E. Frank St, Norman, Okla

DAWSON, Thomas L. (*M* 1930) Pres (for mail) Thomas L. Dawson Co, 2035 Washington St, Kansas City, Mo, and Shawnee Mission Rd, Rosedale Station, Kansas City, Kan

DAY, Harold C. (*A* 1934) Mgr (for mail) American Radiator Co, 1807 Elmwood Ave, Buffalo, and 223 Woodcrest Blvd, Kenmore, N Y

DAY, Irving M. (*A* 1936) Sales Engr (for mail) 709 Mills Bldg, Washington, D C, and 405 Cumberland Ave, Chevy Chase, Md

DAY, V. S.* (*M* 1924) Director Dealer Engrg (for mail) Carrier Corp, and 306 Highland Ave, Syracuse, N Y

DAYNES, Joseph H. (*M* 1938) Application Engr (for mail) Canadian General Electric Co, Ltd, 212-214 King St, W, and 469 Soudan Ave, Toronto, Ont, Canada

DEAN, Carl H. (*M* 1936) Air Cond Engr (for mail) Oklahoma Natural Gas Co, P O Box 871, and 1007 North Main, Tulsa, Okla

DEAN, Charles L. (*M* 1932) Asst Prof Mech Engrg, University of Wisconsin, 305 University Extension Bldg, and (for mail) 102 Grand Ave, Madison, Wis

DEAN, David (*M* 1937) Sales Engr, New York Plumbers Specialties Co, Inc, 334 E 98th St, New York, and (for mail) 171 Radford St, Yonkers, N Y

DEAN, Frank J, Jr. (*J* 1935, *S* 1934) Sales Engr, Gustin-Bacon Mfg Co, 1412 West 12th St, and (for mail) 6028 Walnut St, Kansas City, Mo

DEAN, Marshall H. (*J* 1938, *S* 1936) Secy-Treas, Dean-Hagny Corp, 14th & McGee St, and (for mail) 1030 W 55th St, Kansas City, Mo

DEE, Leo H. (*J* 1937) Junior Engr, Carrier Corp, International Div, and (for mail) 1026 W Onodaga St, Syracuse, N Y

DEGLER, Howard E (*M* 1938) Prof of Mech Engrg (for mail) University of Texas, and 1405 Hardovin Ave, Austin, Tex

DeLAND, Charles W. (*M* 1924, *J* 1923) Secy-Treas (for mail) C W Johnson, Inc, 211 N Desplaines St, and 2021 Estes Ave, Chicago, Ill

DELANY, John V. (*S* 1938) Worcester Polytechnic Institute, and (for mail) 34 Fruit St, Worcester, Mass

DELAVAN, Nelson B. (*M* 1938) Senior Partner (for mail) Delavan Engineering Co, 414 12th St, and 338 42nd St, Des Moines, Ia

DELL'ORTO, Luciano (*J* 1938) General Engr, Ing Giuseppe Dell'Orto, 18 Via Merano, Milano (139) Italy

DEMAREST, Richard T. (*J* 1938) Air Cond Sales Engr, Easternoil Inc, 133 Marginal Way, and (for mail) 64 Deering St, Portland, Me.

DEMPSEY, Stephen J. (*A* 1938) Pres-Treas, Stephen J Dempsey Co (for mail) 79 Harvard, Battle Creek, Mich

DENISE, John R. (*A* 1937, *J* 1935) Development Engr (for mail) Surface Combustion Corp, 400 Dublin Ave, and 136 E Broad St, Columbus, O.

DENNY, Harold R. (*A* 1934) Eastern Merchandise Mgr (for mail) American Blower Corp, 50 W 40th St, New York, N Y, and 429 Edgewood Ave, Westfield, N J

DEPPMANN, Ray L. (*A* 1937) Pres (for mail) R L. Deppmann Co, 957 Holden Ave, and 13201 Cloverlawn Ave, Detroit, Mich.

DE SOMMA, Anthony E. (*J* 1937) Engr., Heating, Ventilating & Air Cond., Worlds Fair, Flushing, and (for mail) 2052 Homecrest Ave., Brooklyn, N. Y.

DES REIS, John F. (*M* 1936) Regional Mgr, Carrier Corp., and 102 Century Drive, Syracuse, N Y.

DETERLING, William C. (*A* 1937) Salesman (for mail) General Electric Co, 570 Lexington Ave, New York, and 32 W Milton St., Freeport, N Y.

DEVER, Henry F. (*M* 1936; *A* 1935) Branch Mgr., Minneapolis-Honeywell Regulator Co, Wayne & Roberts Ave, Philadelphia, and (for mail) 502 Merwyn Rd, Narberth, Pa

DeVILBISS, Parker T. (*A* 1937) Htg and Air Cond Engr, 11½ N. Lee Ave., Oklahoma City, Okla

DEVORE, Angus B. (*A* 1937) Sales Engr. (for mail) James A. Messer Co, Inc, 1206 K St, N W, Washington, D C, and 2016 Queens Chapel Rd (Avondale) Hyattsville, Md

DEWEY, Ritchie P. (*M* 1934) Mgr, Temperature Control & Uni-Flo Depts (for mail) Barber-Colman Co, River & Loomis Sts, and 2301 Oxford St, Rockford, Ill

DeWILDE, Marinus Pieter (*J* 1938) Ing. M. P. DeWilde, M T S H, N V Ind My Gebr. Van Swaay, Banka Straat 134, Den Haag (The Hague) Holland

DeWITT, Earl S. (*A* 1936) Branch Mgr, Washington Office (for mail) American Blower Corp, 438 Woodward Bldg, Washington, D C, and 3224 Oliver St, Chevy Chase, Md

DIAMOND, David D. (*J* 1937) Design Engr, Twin City Furnace Co, 13 S 3rd St, Minneapolis, and (for mail) 118 E. Congress St, St Paul, Minn

DIBBLE, S. E.* (*M* 1917) (*Presidential Member*) (Pres, 1925; 1st Vice-Pres, 1924; 2nd Vice-Pres, 1923, Council, 1921-1926) Supt, Thomas Ranken Patton School, Elizabethtown, Pa

DICK, Andrew V. (*J* 1935) Pres, A V Dick Heating Co, 141 Jay St, Albany, N. Y

DICKASON, Gray D. (*M* 1938) Pres (for mail) Genesee Heating Service, Inc, 950 Mercantile Bldg, and 140 Windemere Rd, Rochester, N Y.

DICKENSON, Frederick R. (*M* 1936, *A* 1934) Asst Sales Mgr (for mail) American Blower Corp, 6000 Russell St, Detroit, and 715 Pilgrim Rd, Birmingham, Mich

DICKENSON, Malcolm E. (*M* 1936) Vice-Pres & Mgr (for mail) Livingston Stoker Co, Ltd, 78 Catharine St, N and 964 Cumberland Ave, Hamilton, Ont, Canada

DICKEY, Arthur J. (*M* 1921) Vice-Pres, Gen Mgr., C A Dunham Co, Ltd, 1523 Davenport Rd, and (for mail) 9 Mossom Place, Toronto, Ont, Canada

DICKINSON, Robert P., Jr. (*J* 1938) Warehouse Mgr (for mail) Burnham Boiler Corp. of Ohio, 301 Brushton Ave, Pittsburgh, and 219 Meade St, Wilkinsburg, Pa

DICKSON, George P. (*M* 1919) Pres (for mail) B F. Sturtevant Co, 89 Broad St, Boston, Mass, and P O Box 22, Canterbury, N H

DICKSON, Robert B. (*M* 1919) Pres (for mail) Kewanee Boiler Corp, Franklin St., and Q Tracks, and 145 E Division St, Kewanee, Ill

DICKSON, Robert W., Jr. (*J* 1938) Sales Engr, American Blower Corp, 1433 Oliver Bldg, Pittsburgh, Pa

DIETZ, C. Fred (*M* 1937) Sales Engr (for mail) Haynes Selling Co, Inc, 1124 Spring Garden St, and 1215 Allengrove St, Philadelphia, Pa

D'IMOR, Elton J. (*M* 1933) Factory Repr, U S Air Conditioning Corp, 750 Union Ave, and (for mail) 2102 Cowden Ave, Memphis, Tenn.

DION, Alfred M. (*M* 1937) Sales Engr. (for mail) Trane Co of Canada, King & Mowat Sts., and 356 Bloor St, E, Toronto, Ont, Canada.

DISNEY, Melvin A. (*A* 1934) Pres (for mail) Disney-Leffel Co, Inc, 3323 Main St, and 6648 Kenwood, Kansas City, Mo

DISTEL, Robert E. (*J* 1938) Distel Heating Equipment Co., 404-6 Kalamazoo Plaza (for mail) Post Office Box 133, Lansing, and 547 Bailey St, East Lansing, Mich
DIVER, M. L. (*M* 1925) Consulting Engr., P. O. Box 1016, San Antonio, Tex.
DIXON, Arthur G. (*M* 1928) Sales Mgr. (for mail) Modine Mfg Co, and 442 Wolff St, Racine, Wis.
DIXON, Meridith F. (*A* 1937) Combustion Engr, Fuel Oil and Oil Burner Div, Imperial Oil, Ltd, P O. Box 1440, and (for mail) 2281 Wilson Ave, Montreal, P Q., Canada.
DODDS, Forrest F. (*M* 1920) Mgr., K. C. Branch (for mail) American Radiator Co, 1023 Grand Ave., and Park Lane Hotel, 4600 Mill Creek Pkwy., Kansas City, Mo
DODGE, Harry A. (*M* 1936) Elec Engr., S. H Kress & Co., 114 Fifth Ave, and (for mail) 425 East 86th St, New York, N. Y.
DOERING, F. L. (*M* 1919) Salesman, American Radiator Co, Richmond, and (for mail) 238 Boston Ave, Lynchburg, Va
DOERR, Carl F. (*S* 1938) Draftsman, American Can Co., 13th Ave & St Charles, and (for mail) 1118 N 8th Ave, Maywood, Ill
DOLAN, Raymond G. (*M* 1926, *J* 1922) Secy-Treas (for mail) Tom Dolan Heating Co, Inc, 614 W Grand, and 2112 West 20th, Oklahoma City, Okla
DOLSON, Charles N. (*A* 1937) Sales Engr, Illinois Iron & Belt, 908 S. Michigan Ave, and (for mail) 46 Hawkins Ave, Chicago, Ill
DOME, Alan G. (*A* 1938, *J* 1936) Engr, Elliott-Lewis Co, 2514-18 N. Broad St, and (for mail) 7129 Chew St, Philadelphia, Pa
DONELSON, William N. (*J* 1937) Htg Engr. (for mail) T J Conner, Inc, 3290 Spring Grove, and 795 Ludlow, Cincinnati, O
DONNELLY, James A.* (*Life Member*, *M* 1904) (Treas, 1912-1914) Largent, W Va
DONNELLY, Martin A. (*J* 1937) Sub-Inspector, Frankford Arsenal, Philadelphia, and (for mail) 500 Manoa Rd, Brookline, Del Co, Pa
DONNELLY, Russell (*M* 1923) Sales Engr., Nash Engineering Co, Graybar Bldg, 420 Lexington Ave, New York, N Y
DONOHOE, John B. (*A* 1937, *J* 1935) Engr. & Estimator (for mail) B F Donohoe Co, 51 Albany St., Boston, and 23 Primrose St, Roslindale, Mass
DONOVAN, William J. (*A* 1930) 2239 North 27th St, Philadelphia, Pa
DORFAN, M. I. (*M* 1929) Dust Control Specialist, Pangborn Corp, 604 Chamber of Commerce Bldg, and (for mail) 1217 Malvern Ave, Pittsburgh, Pa
DORNHEIM, G. A. (*M* 1912; *J* 1906) 15 Hamilton Ave, Bronxville, N Y
DORSEY, Francis C. (*M* 1920) Engr-Contractor (for mail) Francis C Dorsey, Inc, 4520 Schenley Rd, Roland Park, and 212 Gittings Ave., Baltimore, Md
DOSTER, Alexis (*A* 1934) Vice-Pres. and Secy, The Torrington Mfg Co, 70 Franklin St, Torrington, Conn.
DOUGHTY, Charles J. (*M* 1925) Pres-Managing Director (for mail) C J Doughty & Co, Fed Inc, U S. A, 30 Brenan Rd, and 1920 Ave, Joffre, Shanghai, China
DOUGLAS, Howard H. (*A* 1936) Air Cond and Htg. Engr (for mail) Southern California Edison Co., 601 W 5th St, and 2317 Kelton Ave, Los Angeles, Calif
DOVOLIS, Nick J. (*J* 1936, *S* 1935) 3403 Chicago Ave, Minneapolis, Minn
DOWLER, Edward A. (*M* 1937) Sales Engr, B. F Sturtevant Co of Canada, Ltd, 137 Wellington St West, and (for mail) 9 Prince Arthur Ave, Toronto, Ont, Canada
DOWNE, Edward R. (*M* 1927) Vice-Pres, American Gas Products Corp., 40 West 40th St, New York, and (for mail) 35 Howell Ave., Larchmont, N. Y.
DOWNES, Alfred H. (*A* 1937) Draftsman, English & Lauer, 1978 S. Los Angeles St., and (for mail) 1342½ Bond St, Los Angeles, Calif.
DOWNES, H. H. (*M* 1923) Dist. Mgr., Mgr. Navy Equip. Div. (for mail) American Blower Corp, 438 Woodward Bldg, Washington, D. C., and 4621 Chevy Chase Blvd., Chevy Chase, Md.
DOWNES, Nate W. (*M* 1917) Chief Engr. & Supt of Bldgs., School Dist of Kansas City, Mo, 317 Finance Bldg, Kansas City, Mo
DOWNING, Clarence B. (*A* 1938) Secy-Treas, N. B. Downing Co, Jefferson Ave, and (for mail) Clark Ave, Milford, Del
DOWNS, Charles R. (*M* 1936) Pres, Calorider Corp., Old Greenwich, Conn, and (for mail) 50 E 41st St., New York, N Y.
DOWNS, Sewell H. (*M* 1931) (Council, 1937-1938) Chief Engr., Clarage Fan Co., and (for mail) 211 Creston Ave., Kalamazoo, Mich
DOXEY, Harold E. (*A* 1937) Safety Engr (for mail) Ocean Accident & Guarantee Corp, Ltd., 308 Phoenix Bldg., and 4251 Quincy St, N E., Minneapolis, Minn.
DOYLE, Bernard J. (*A* 1938) Sales Engr, Weil-McLain Co, and (for mail) 1430 Collingwood, Detroit, Mich.
DOYLE, William J. (*M* 1920) Designing Engr, Williamson Heater Co, 337 W. Fifth St, Cincinnati, and (for mail) P O. Box 52, Goshen, O
DRAKE, George M. (*J* 1936) Vice-Pres (for mail) George H Drake, Inc, 218 Lexington Ave, Buffalo, and 163 Renwood Ave, Kenmore, N. Y.
DREHER, Louis F. (*S* 1938) Student of Air Cond, David Rankin Jr School of Mech Trades, and (for mail) 4816 Margaretta Ave, St Louis, Mo
DRESCHER, Francis E. (*A* 1938) Sales Engr, Straus-Frank Co, and (for mail) 117 Estelle, Houston, Tex
DRESSELL, Russell E. (*A* 1938) Mech Engr, Riggs Distler Co, Inc, 216 N Calvert St, and (for mail) 918 E Preston St, Baltimore, Md
DRIEMEYER, Ray C. (*J* 1937) Sales Engr, Airtherm Mfg Co, 1474 S Vandeventer, and (for mail) 5410 Vernon Ave, St Louis, Mo
DRINKER, Philip* (*M* 1922) Prof of Industrial Hygiene (for mail) Harvard School of Public Health, 55 Shattuck St, Boston, and Newton Center, Mass
DRISCOLL, Marvin G. (*M* 1937) Vice-Pres (for mail) Bryant Equipment Co, Inc, 1725 Rhodes Haverty Bldg, and 20 Collier Rd, Atlanta, Ga
DRISCOLL, William H.* (*M* 1904) (*Presidential Member*) (Pres, 1926, 1st Vice-Pres, 1925; 2nd Vice-Pres, 1924, Treas, 1923, Council, 1918-1927) Vice-Pres (for mail) Carrier Corp, Syracuse, N Y, and 50 Glenwood Ave, Jersey City, N J
DROPPERS, C. J. (*A* 1937) Partner, Wichita Insulation Co, and (for mail) 566 W Douglas Ave, Wichita, Kan
DU BOIS, Louis J. (*M* 1931) Air Cond Engr, York Ice Machinery Corp, 117 S 11th St, St Louis, and (for mail) 7451 Bland Drive, Clayton, Mo
DUBRY, Ernest E (*M* 1924) Asst Supt, Central Heating, Detroit Edison Co, 2000 Second Ave, and (for mail) 9116 Dexter Blvd, Detroit, Mich
DU CHATEAU, Manuel F. (*J* 1938) Sales Engr (for mail) Crane Co, 824 Broadway, and 7 Briarwood Lane, Greenhills, Cincinnati, O.
DUFAULT, Felix H (*A* 1936) Mgr Furnace Div, Quebec & Maritime Provinces (for mail) General Steel Wares, Ltd, 2355 Delisle St, and 2275 Visitation St, Montreal, P Q, Canada
DUGAN, Thomas M. (*M* 1920) Sanitary-Htg Engr, National Tube Co, Fourth Ave and Locust St, and (for mail) 1308 Freemont St, McKeesport, Pa
DULL, Edgar J. (*A* 1937) Engr-Contractor (for mail) 218 Water St, and 3614-3rd St, Brooklyn, Baltimore, Md
DULLE, Willferd L. (*J* 1936) Asst Secy, E E. Souther Iron Co, 1952 Hienlen Ave, St Louis, and 2910 Lincoln Ave, Normandy, Mo.
DUNCAN, James R. (*M* 1923) Sales Engr, Air Cond, Carrier Corp, Room 408, Chrysler Bldg., New York, N. Y.

DUNCAN, William A. (*A* 1930) Mgr., Process Service (for mail) Dominion Oxygen Co, Ltd, 159 Bay St, W, and 71 Jackson Ave., Toronto, Ont, Canada.
DUNHAM, Clayton A.* (*M* 1911) Pres (for mail) C A Dunham Co, 450 E Ohio St, Chicago, and 150 Maple Hill Rd, Glencoe, Ill.
DUNNE, Russell V. D. (*M* 1937) Chief Engr, International Div, Air Cond & Ref (for mail) Carrier Corp, S Geddes St, and 216 Robineau Rd, Syracuse, N Y
DUPUIS, Joseph R. (*A* 1936) Dist Mgr (for mail) Trane Co of Canada, Ltd, 660 St Catherine, W, Montreal, and 331 Clarke Ave, Westmount, P Q, Canada
DURKEE, Merritt E. (*A* 1936; *J* 1931) Commercial Sales, Dallas Gas Co, Harwood and Jackson Sts, and (for mail) 1830 Moser St, Dallas, Tex.
DUTCHER, Harvey S. (*A* 1938) Design Div, A. L Hart, Inc, 315 Vanderbilt Ave, and (for mail) 430 Clinton Ave, Brooklyn, N Y
DWYER, Thomas F. (*M* 1923) Chief of Htg & Vtg. Div (for mail) Board of Education, 49 Flatbush Ave Ext, Brooklyn, and 1163 Clay Ave, New York, N Y
DYKES, James B. (*J* 1936) Estimator (for mail) T A Morrison & Co, Ltd, 1070 Bleury St, and Apt No. 2, 3141 Maplewood Ave, Montreal, P Q, Canada
DYKMAN, John G. (*A* 1938) Owner, John G Dykman, 205 Lane Ave, S W., Grand Rapids, Mich.

E

EADE, Hugh R. (*M* 1935) Archt (for mail) Eade & Co, 2 Imperial Bank Bldg, and 163 Cheriton Ave, North Kildonan, Winnipeg, Man, Canada
EADIE, J. G. (*M* 1909) Consulting Engr, Eadie, Freund & Campbell Co, 110 West 40th St, New York, N Y
EAGLETON, Sterling P. (*M* 1936) Group Supt, U. S Government, Room 2070, Interior Bldg, and (for mail) 3522 "S" St, N W, Washington, D C
EARL, Warren (*A* 1936) Vice-Pres & Gen Mgr, E S. Ko Mfg. Corp and (for mail) 515 Fargo Ave, Houston, Tex.
EARLE, Frederic E. (*M* 1937) Sales Engr (for mail) 520 Howard Ave, Bridgeport, and 1536 Main St, Stratford, Conn
EASTMAN, Carl B. (*M* 1932, *J* 1929) Sales Engr. (for mail) C A Dunham Co, 1500 Walnut St., Philadelphia, and 530 Brookview Lane, Brookline, Pa
EASTWOOD, E. O. (*M* 1921) Head of Dept of Mech Engrg (for mail) University of Washington, and 4702-12th Ave, N E, Seattle, Wash
EASTWOOD, Harry F. (*M* 1925) Mgr Anthracite Industries, Permanent Exhibit, Architects Sample Corp, 101 Park Ave, New York, and (for mail) 157 Frankel Blvd, Merrick, L I, N. Y
EATON, Byron K. (*M* 1920) Zone Mgr, Delco-Frigidaire Conditioning Div, General Motors Sales Corp, 1420 Wisconsin Ave, Dayton, O, and (for mail) 240 S. Brainard Ave., LaGrange, Ill
EATON, William G. M. (*A* 1934) Sales Engr, Pease Foundry Co, Ltd., 227 Victoria St., and (for mail) 300 Wellesley St, Toronto, Ont, Canada
EBERT, William A. (*M* 1920) Mech Contractor (for mail) 1026 W Ashby, and 2151 W. Kings Highway, San Antonio, Tex
ECKART, John H. (*A* 1938) Salesman, Htg. Dept, Sears Roebuck & Co., 6339 Market St, and (for mail) 6012 Lawndale Ave., Philadelphia, Pa
ECKSTEIN, Jacob E. (*A* 1938) Pres. (for mail) J. E Eckstein Co, 22 Penn Ave., and 1323 Denniston Ave, Pittsburgh, Pa
EDELMAN, Bernard P. (*A* 1935) Asst. Sales Mgr. (for mail) U. S Air Conditioning Corp, 2101 Kennedy St, N E, and 4338 Nicollet Ave, Minneapolis, Minn
EDGE, Alfred J. (*M* 1938) Engr. in charge Htg. & Air Cond (for mail) c/o John F. Reynolds, Consulting Engr., Room 316 Duval Bldg., and 2051 Herschel St, Jacksonville, Fla
EDWARDS, Arthur W. (*M* 1936) Branch Mgr., The Trane Co, 626 Broadway, and (for mail) 3423 Paxton Ave, Cincinnati, O.
EDWARDS, Daniel F. (*M* 1920) Pres. (for mail) D F. Edwards Heating Co, 2340 Pine St., St. Louis, Mo, and R R No 1, Millstadt, Ill.
EDWARDS, Don J. (*A* 1933) Vice-Pres. (for mail) General Heat & Appliance Co, 596 Commonwealth Ave, and 8 Devon Terrace, Boston, Mass.
EDWARDS, Henry B. (*J* 1935) (for mail) c/o W. A. Ramsats, Ltd., P. O. Box 1721, Honolulu, T H
EDWARDS, Junius D. (*M* 1936) Asst Dir. of Research (for mail) Aluminum Research Laboratories, Aluminum Company of America, P O. Box 772, New Kensington, and 536 Sixth St., Oakmont, Pa
EDWARDS, Lawrence V. (*M* 1938) Engr, Buensod-Stacey Air Conditioning, Inc, 60 E. 42nd St, New York, N Y, and (for mail) 1631 Edmund Terrace, Union, N J
EDWARDS, Paul A. (*M* 1919) Pres (for mail) The G F Higgins Co, 608 Wabash Bldg., Pittsburgh, and 3074 Pinehurst Ave, Pittsburgh (16), Pa
EGGLESTON, Herbert L. (*M* 1938) Mgr, Gas & Ref Depts, Gilmore Oil Co, 2423 E 28th St, Los Angeles, and (for mail) 1017 Cumberland Rd, Glendale, Calif
EHLERS, Jacobus (*J* 1937) Engr (for mail) Carrier Engineering South Africa, Ltd, Box 7821, and Jacwal Court, Quartz St, Johannesburg, South Africa
EHRLICH, M. William* (*M* 1916) Chief Engr., Commodore Heaters Corp, 11 West 42nd St., New York, N Y, and (for mail) 56 Ridge Rd., Lyndhurst, N J
EICHER, HuBert C. (*M* 1922) Chief, Div of School Plant, Pennsylvania State Dept of Public Instruction, State Capitol, and (for mail) 207 North 30th St, Harrisburg, Pa
EILS, Lee C. (*J* 1936) Zone Mgr, Barton D Wood, Inc, 206 Balter Bldg, New Orleans, La, and (for mail) 3223 Kennett Sq, Pittsburgh, Pa.
EISELE, Dudley E. (*A* 1938) Owner (for mail) Eisele Engineering Co, 121 N. Appleton St., and 1735 N Morrison St., Appleton, Wis
EISELE, Lewis G. (*A* 1937) Secy (for mail) Eisele Automatic Heating Co, Box 309, and 602 W Hughitt St, Iron Mountain, Mich
EISELE, William S. (*A* 1937) Supervising Engr. (for mail) Ideal Heating & Air Conditioning Co., 551 Seneca St, and 836 Tacoma Ave, Buffalo, N Y
EISS, Robert M. (*M* 1933; *J* 1930) Engr, Kimberly-Clark Corp, P. O Box 31, and (for mail) 714 Hewitt St, Neenah, Wis
EKINGS, Robert M., Jr. (*M* 1938) Engr, Air Cond Dept, General Electric Co, 5 Lawrence St, Bloomfield, N. J, and (for mail) 132 S. Oakhurst Drive, Beverly Hills, Calif.
EKLUND, Karl G. (*M* 1938) Consulting Engr. (for mail) Karl G Eklunds Ingeniorsbyra, Brunkebergstorg 15, Stockholm, and Storangen, Sweden
ELBERT, Ben F. (*J* 1937) Sales Engr, Sidles Co., Airtemp Div, 425 Stuart Bldg, Lincoln, Nebr, and (for mail) 1008 Eighth St., Des Moines, Ia.
ELLINGWOOD, Elliott L. (*M* 1909) Cons Mech. & Elect Engr (for mail) 124 W. 4th St, Los Angeles, and 210 S Los Robles Ave., Pasadena, Calif.
ELLIOT, Edwin (*M* 1929) (for mail) Edwin Elliot & Co, 560 North 16th St, Philadelphia, and 403 W Price St, Germantown, Philadelphia, Pa.
ELLIOT, Gerald B. (*M* 1938) Sales Engr., Francis Hankin Co., Ltd, and (for mail) 4971 Victoria Ave, Montreal, P. Q, Canada.
ELLIOTT, Irwin (*A* 1937) Chief Engr., Universal Oven Co, 271 Broadway, New York, and (for mail) 103 Penfield Ave., Croton, N. Y.

ELLIOTT, Louis B. (*M* 1932) Consulting Mech. Engr. (for mail) Ebasco Services Inc, 2 Rector St., and 33 Washington Square West, New York, N Y.
ELLIOTT, Norton B. (*A* 1934) Branch Mgr., American Blower Corp., 1011 Majestic Bldg, Milwaukee, Wis
ELLIS, Frederic R. (*M* 1913) Buerkel & Co, Inc, 18-24 Union Park St., Boston, and (for mail) 131 Beacon St, Hyde Park, Mass
ELLIS, Frederick E. (*M* 1923) Sales Mgr (for mail) Imperial Iron Corp, Ltd, 30 Jefferson Ave, Toronto, and 9 Princeton Rd, Kingsway P O, Toronto 3, Ont, Canada
ELLIS, Gershom P. (*M* 1935) Chief Engr (for mail) Board of Public Education, 341 Bellfield Ave, and 6601 Dalzell Place, Pittsburgh, Pa.
ELLIS, Harry W. (*Life Member*; *M* 1923, *A* 1909) Pres-Gen Mgr, Johnson Service Co, 507 E Michigan St, Milwaukee, Wis
ELLIS, Lester W. (*A* 1938) Sales Engr., Disney-Leffel Co, Inc (for mail) 3323 Main St, and 3835 Main St, Kansas City, Mo
ELWOOD, Willis H. (*M* 1936) Branch Mgr, Holland Furnace Co, 209 King St, Ithaca, N. Y
EMERSON, John J. (*J* 1938) Sales Engr, 424 Sherbrooke St, W, Apt 35, Montreal, P. Q, Canada
EMERSON, Ralph R. (*M* 1922) Pres, Emerson Swan Goodyer Co, 107 Arlington St, Boston, and (for mail) 44 Whitney Rd, Newtonville, Mass
EMMERT, Luther D. (*M* 1919) Repr (for mail) Buffalo Forge Co, Room 1909, 20 N Wacker Drive, Chicago, and 1704 Hinman Ave, Evanston, Ill
EMSWILER, John E.* (*M* 1917) Prof of Mech Engrg (for mail) University of Michigan, 221 W Engrg Bldg, and 1303 Granger, Ann Arbor, Mich
ENDERS, Clarence E. (*A* 1938) Pacific Coast Factory Repr & Engr (for mail) Electrol, Inc, 424 E. Burnside, and 1813 S E 60th Ave, Portland, Ore
ENGDAHL, Richard B. (*J* 1938) Special Research Asst (for mail) University of Illinois, 102 Mech Engrg Lab, and 1108 W Stoughton, Urbana, Ill
ENGLE, Alfred (*A* 1923) Secy (for mail) Jenkins Bros, 80 White St, New York, and 1 Edgewood Rd, Scarsdale, N Y
ENGLISH, Harrold (*M* 1935; *A* 1930) Pres (for mail) English & Lauer, Inc, 1978 S Los Angeles St, and 515 S Norton, Los Angeles, Calif
ENSIGN, Willis A. (*M* 1935) Vice-Pres, Frontier Engineering Corp, 986 Ellicott Square Bldg, Buffalo, and (for mail) Revere Drive, Derby, N Y.
EPPLE, Arnet B. (*J* 1934) 201 Benjamin Ave, S E, Grand Rapids, Mich
ERICKSON, E. Vincent (*M* 1936) Mgr, New York Export Office, Carrier Corp, 405 Lexington Ave, New York, N Y
ERICKSON, Harry H. (*A* 1929) Sales Engr (for mail) Haynes Selling Co, 1124 Spring Garden St, and 217 W Tulpehocken St, Philadelphia, Pa.
ERICKSON, Martin E. (*A* 1926) Supt Maintenance, Board of Education, and (for mail) 1533 South 74th St, West Allis, Wis
ERICSSON, Eric B. (*M* 1933) Engr, Board of Education, and (for mail) 605 West 116th St, Chicago, Ill
ERISMAN, Percival H., Jr (*M* 1936) Chief Engr, Washington Refrigeration Co, 1731-14th St, N. W, and (for mail) 2240-40th St, N W, 2, Washington, D C
ERRATH, Edward O. (*J* 1936) Htg and Air Cond Engr (for mail) The Heil Co, 3000 W Montana, and 2646 N. 45th St., Milwaukee, Wis
ESCHENBACH, Samuel P. (*J* 1935) Sales Engr (for mail) American Blower Corp, 135 Spring St, and 268 Dartmouth St., Rochester, N. Y
ESKELL, Joseph E. (*S* 1939) Student, Columbia University, New York, and (for mail) 18-27 21st Rd, Astoria, L. I, N Y
ESTEP, Leslie G. (*M* 1936) Asst Mgr., Technical & Training Dept, Kelvinator Div, Nash-Kelvinator Corp, 14250 Plymouth Rd., and (for mail) 14909 Marlowe Ave, Detroit, Mich.
ESTES, Edwin C. (*A* 1936) Mech. Draftsman (for mail) Railway Transportation, Rm. 820 Northern Pacific Ry., Gen Office, St. Paul, and Victoria Ave, Mendota, Minn
ETLINGER, Martin J. (*J* 1936) Sales Mgr., The Bradley Press, Inc, 103 Lafayette St., and 2979 Marion Ave, New York, N. Y.
EUTSLER, Eugene E., Jr. (*J* 1938) Engr. (for mail) Buffalo Forge Co, 490 Broadway, and 179 Lexington Ave, Buffalo, N. Y.
EVANS, Bruce L. (*M* 1938; *A* 1937) Designing Engr. (for mail) Oil Heat, Inc, 3217 Locust St. and 6322 Pershing Ave, St Louis, Mo
EVANS, Edwin C. (*M* 1919) Mgr, Syracuse Office, (for mail) B F Sturtevant Co, 504 Eckel Bldg., and 307 Montgomery St, Syracuse, N Y.
EVELETH, Charles F.* (*M* 1911) Air Cond Engr., Smith & Oby Co., 6107 Carnegie Ave, and (for mail) 2030 East 115th St, Cleveland, O
EVEREST, R. Harry (*M* 1935) Sales Engr, Sheldons, Ltd, Galt, and (for mail) 235 Waterloo St, Preston, Ont, Canada
EVERETTS, John, Jr.* (*M* 1938; *A* 1935; *J* 1929) Engr (for mail) Air & Refrigeration Corp, 11 West 42nd St, New York, N Y, and 55 Sound Beach Ave, Old Greenwich, Conn
EWENS, Frank G* (*M* 1937) Instructor in Mech Engrg (for mail) University of Toronto, Mechanical Bldg, University of Toronto, and 83 Madison Ave, Toronto, Ont, Canada.
EZZ EL DIN, Kamal (*J* 1938) Engr, Ministry of Public Works, and (for mail) 78 Helwan St, Mounirah, Cairo, Egypt

F

FABER, Dr. Oscar (*M* 1934) Consulting Engr (for mail) Romney House, Marsham St, Westminster, London and Hayes Court, Kenley, Surrey, England
FABLING, Walter D. (*A* 1937) Sales Mgr (for mail) Sterling Electric Motors, Inc, 5401 Telegraph Rd, and 2121 Glendon Ave, Los Angeles, Calif
FAGIN, Daniel J. (*M* 1932) Htg Engr, Laclede Gas Light Co, 1017 Olive St, St Louis, Mo
FAHNESTOCK, Maurice K.* (*M* 1927) Research Asst Prof (for mail) University of Illinois, 214, M E Laboratory, and 701 W California St. Urbana, Ill
FAILE, Edward H. (*M* 1934) Designing and Construction Engr (for mail) 608 Fifth Ave, New York, N Y, and R F D 1, Westport, Conn
FAIRBANKS, Frank L. (*M* 1937) Prof Agr Engrg & Agr Eng in Exp Station, Cornell University, and (for mail) 424 E. State St, Ithaca, N Y
FALK, David S. (*J* 1937) Sales Engr (for mail) The Trane Co., 8316 Woodward, Detroit, and 809 E Kingsley St, Ann Arbor, Mich
FALTENBACHER, Harry J. (*M* 1930) Pres, Harry J Faltenbacher, Inc, 235 E. Wister St, Philadelphia, Pa
FALVEY, John D. (*M* 1922) Consulting Engr., 316 N Eighth St, St Louis, and (for mail) 6636 Pershing Ave, University City, Mo
FAMILETTI, A. Robert (*M* 1938, *J* 1930) Chief Engrg Draftsman, Industrial Dept, Navy Yard, and (for mail) 6735 Guyer Ave, Philadelphia, Pa.
FARBER, Louis M. (*J* 1936) Engr (for mail) Natkin & Co., 1800 Baltimore, and 3714 Flora Ave, Kansas City, Mo
FARLEY, W. F. (*M* 1930) Salesman, American Radiator Co, 40 West 40th St, New York, and (for mail) 28 Elm St., New Rochelle, N Y.
FARNES, Bert W. (*A* 1938) Vice-Pres & Sales Mgr (for mail) Control Equipment Co, 304 Selling Blvd, and 3565 N. E. Hollyrood Court, Portland, Ore

FARNHAM, Roswell (*M* 1920) (Council, 1927-1933) Dist Mgr, Engrg. Sales (for mail) Buffalo Forge Co, P O. Box 985, and 5 Clarendon Place, Buffalo, N. Y

FARRAR, Cecil W. (*M* 1920; *A* 1918) (Treas, 1930; Council, 1930) Vice-Pres (for mail) W A Case & Son Mfg Co, 31 Main St, and 29 Oakland Place, Buffalo, N Y

FARROW, Ernest E. (*A* 1938) Pres, E E Farrow, Inc, 2808 Maplelawn Rd, and (for mail) 1518 Kings Highway, Dallas, Tex

FARROW, Hollis L. (*J* 1937) Air Cond Engr, Kelly Sales Corp., Arlington, and (for mail) 910 Lynnfield St., Lynn, Mass.

FATZ, Joseph L. (*M* 1935) Htg and Vtg Engr, Board of Education, Room 536, 228 N LaSalle St, and (for mail) 1634 N Mason Ave, Chicago, Ill

FAULKNER, Gordon (*J* 1937, *S* 1935) Engr, Standard Oil Development Co, Bayway, and (for mail) 43 Bellewood Place, Elizabeth, N J

FAUST, Frank H.* (*M* 1936; *J* 1930) Engr (for mail) General Electric Co, 5 Lawrence St, Bloomfield, and 239 Vreeland Ave, Nutley, N J

FAXON, Harold C. (*M* 1937) Engr, Appliance Section (for mail) Borneo Co, Ltd, Mercantile Bank Bldg, and 73 Grange Rd, Singapore, S S

FAY, Donald P. (*S* 1936) 91-38-115th St, Richmond Hill, L I., N Y

FAY, Frank C. (*M* 1925) Engr (for mail) Raisler Heating Co, 129-31 Amsterdam Ave, New York, and 92-17-54th Ave, Elmhurst, L I, N Y

FEAR, S. Lorne (*M* 1938) Asst Mech Engr (for mail) H. E P C, 620 University Ave, Toronto, 2, and 18 Vesta Drive, Toronto, 10, Ont, Canada

FEBREY, Ernest J. (*Life Member*, *M* 1903) Pres (for mail) E J Febrey & Co, Inc, 616 New York Ave, N W, and 2331 Cathedral Ave, N W, Washington, D C

FEDDERS, Melvin P. (*M* 1938) Development Engr, Minneapolis-Honeywell Regulator Co, 2747-4th Ave, S, and (for mail) 5212 W Nokomis Pkwy, Minneapolis, Minn

FEDER, Nathan (*J* 1938) Engr, East Side Sheet Metal Co, 1397 Second Ave, and (for mail) c/o Bialek, 909 Beck St, New York, N Y

FEEHAN, John B. (*Life Member*; *M* 1923) Pres-Treas (for mail) John B Feehan, Inc, 58 Spring St, Lynn, and 4 Long View Drive, Marblehead, Mass

FEELY, Frank J. (*M* 1935, *A* 1929) Mgr of Sales, Taylor Supply Co, 700 Monroe Ave, Detroit, and (for mail) 950 Trombley Rd, Grosse Pointe Park, Mich

FEHLIG, John B. (*Life Member*, *M* 1918) Pres-Treas (for mail) Excelsior Heating Supply Co, 528 Delaware St, and 2927 Brooklyn Ave, Kansas City, Mo

FEINBERG, Emanuel (*J* 1937) General Mgr, Thermalair Engineering Co, 439 Penobscot Bldg, and (for mail) 3246 Cortland Ave, Detroit, Mich

FEIRN, William H. (*M* 1937) Engr, C A Hooper Co, 453 W Gilman, and (for mail) Shorewood Hills, Madison, Wis

FELDERMANN, William (*A* 1937) Pres, Walton Laboratories, Inc, 1186 Grove St, Irvington, N J

FELDMAN, A. M.* (*Life Member*; *M* 1903) Consulting Engr, 40 West 77th St, New York, N Y

FELDSTEIN, Harold (*J* 1938) Consulting & Design Engr, Federal Supply Co, 120 E Main St, and (for mail) 1015 E Park, Oklahoma City, Okla

FELLOWS, Julian R. (*M* 1938) Associate in Mech Engrg (for mail) University of Illinois, 105 M E Laboratory, and 703 W. California, Urbana, Ill

FELS, Arthur B. (*M* 1919) Pres (for mail) The Fels Co, Portland, and Yarmouth, Me

FELTWELL, Robert H. (*Life Member*; *M* 1905) Heating Engr, United States Radiator Corp, 2321-4th St, N. E, and (for mail) 1370 Oak St, N W, Washington, D C

FENKER, Clement M. (*M* 1937) Designing Mech Engr (for mail) Edward J. Shulte, Archt., 920 E McMillan, Cincinnati, and 2268 Feldman Ave, Norwood, O

FENNER, Everett M. (*M* 1936; *A* 1928) Htg.-Vtg. Engr, 387 High St., Fall River, Mass.

FENNER, N. Paul (*A* 1928) (for mail) John G. Kelly, Inc, 210 East 45th St, New York, and 15 De Mott Place, Rockville Center, L. I., N. Y.

FENSTERMAKER, Sidney E. (*M* 1909) Pres (for mail) S E Fenstermaker & Co, 937 Architects & Builders Bldg, Indianapolis, and Carmel, Ind

FERDERBER, Dr. Murray B. (*M* 1938) Fellow of Dept of Industrial Hygiene, University of Pittsburgh Medical School, and (for mail) 5722 Fifth Ave, Pittsburgh, Pa

FERGESTAD, Marvin L. (*M* 1938; *J* 1935) Sales Engr, Bark Prod Div (for mail) The Pacific Lumber Co of Illinois, 59 East Van Buren St, and 2704 Arthur Ave, Chicago, Ill

FERGUSON, John H. (*A* 1938) Engr in Charge, John Ferguson Plumbing & Heating Co, 2700 Euclid Ave, and (for mail) 4169 West 50th St, Cleveland, O

FERGUSON, Ralph R. (*M* 1934, *A* 1927, *J* 1925) Mgr Air Cond Dept, American Blower Corp., 50 West 40th St., New York, N Y, and (for mail) 160 Prospect St, East Orange, N J

FERRARINI, Joseph (*J* 1937) Staff Asst, Washington Gas Light Co, 411-10th St, N W, Washington, D C, and (for mail) 1728 Queens Lane, Colonial Village, Apt No 180, Arlington, Va

FEYGE, Harold (*M* 1937) Mfrs Repr, 742 Market St, Room 230, and (for mail) 625 Scott St, San Francisco, Cal

FIDELIUS, Walter R. (*M* 1936) Sales Engr, Fitzgibbons Boiler Co, Inc, 101 Park Ave, New York, and (for mail) 135 Amersfort Place, Brooklyn, N Y

FIEDLER, Harry W. (*M* 1923) Owner (for mail) Air Conditioning Utilities Co, 8 West 40th St., New York, and 77 Hillside Ave, Mt Vernon, N Y

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FIGGIS, Thomas G. (*A* 1937; *J* 1936) Tech Sales Engr, J & E Hall, Ltd, Dartford Ironworks, Kent, England

FILLO, Frank B. (*A* 1934) Dist Mgr, Minneapolis-Honeywell Regulator Co, 1134 N Pennsylvania Ave, Indianapolis, Ind

FINAN, J. J. (*Life Member*, *M* 1923) Retired, 7149 Euclid Ave, Chicago, Ill

FINERAN, Edward V. (*J* 1935) Special Repr (for mail) Washington Gas Light Co, 411-10th St, N W, Washington, D C, and 305 Edgewood Ave, Silver Spring, Md

FINNERTY, John A. (*J* 1937) Sales Mgr, Auto Heat & Air Conditioning Div, The Herman Nelson Corp, Moline, Ill, and (for mail) 28 Ainsworth St, Roslindale, Mass

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FISCHER, Lawrence (*J* 1937) Engr in charge of Manufacture (for mail) Anemostat Corp of America, 10 East 39th St, New York, and Crosby St, Sayville, L I, N Y

FISHER, John T. (*J* 1936) Chief Engr (for mail) United Equipment & Supply Co, 1812 M St, N W, and 3228 Rittenhouse St, N W., Washington, D C

FITCH, Howard M. (*J* 1936) Asst Sales Mgr, Engine & Compressor Dept, American Air Filter Co, Inc, 215 Central Ave, and 269 Clare Ave, Louisville, Ky

FITTS, Charles D. (*M* 1920) Mgr (for mail) American Radiator Co, 692 Prior Ave, St Paul, and 2807 Dean Blvd, Minneapolis, Minn.

FITTS, Joseph C. (*M* 1930) Secy., Heating, Piping & Air Conditioning Contractors National Association, 1250 Sixth Ave , New York, N Y , and (for mail) 215 Kenilworth Rd , Ridgewood, N. J.
FITZ, Jean Chandler (*M* 1924) Mgr , Arco Thermo System Div. (for mail) American Radiator Co , 40 West 40th St , New York, N. Y.
FITZGERALD, Matthew J. (*M* 1934) Secy - Treas , Standard Asbestos Mfg. Co , 820 West Lake St , Chicago, and (for mail) 1117 N. Linden Ave , Oak Park, Ill.
FITZGERALD, William E. (*J* 1936; *S* 1935) Secy -Treas Engr (for mail) Fitzgerald Plbg & Htg Co., Inc., 939-41 Louisiana Ave , and 435½ Herndon, Shreveport, La
FITZSIMONS, J. P. (*J* 1934; *S* 1932) Mgr. Air Cond. Dept. (for mail) Trane Co of Canada, Ltd , Mowat & King St , W , and 151 Dowling Ave., Toronto, Ont , Canada.
FLARSHEIM, C. A. (*J* 1933) Pres , Clarence A. Flarsheim, Inc , 201-207 Pershing Rd , and (for mail) 3720 Holmes St , Kansas City, Mo
FLEAK, William D. (*A* 1938) Laboratory Engr , Refrigeration & Air Conditioning Institute, Inc , 2141 Lawrence Ave , and (for mail) 4535 N. Mozart St , Chicago, Ill
FLEISHER, Walter L.* (*M* 1914) (Council, 1937-1938) Consulting Engr (for mail) 11 West 42nd St , New York, and New City, N Y
FLINK, Carl H. (*M* 1923) Mech Engr (for mail) American Radiator Co , 8007 Joseph Campau Ave , and 5959 Yorkshire Rd , Detroit, Mich
FLINN, George S. (*J* 1936) Engr , W F Slayter Engineering Corp , 664 Union Ave , and (for mail) 190 N Avalon, Memphis, Tenn
FLINT, Coll T. (*M* 1919) Sales Mgr (for mail) H B Smith Co , 640 Main St , Cambridge, and 56 Brantwood Rd , Arlington, Mass.
FLUM, Joseph C. (*S* 1938) Henry Disston & Sons, Talcony, and (for mail) 4706 Rhawn St , Philadelphia, Pa
FOERSTNER, George C. (*A* 1938) Mgr , Amana Society, Amana, Ia
FOLEY, Daniel F. (*A* 1937) Asst Mgr , W B Young Supply Co. (for mail) 208 Delaware, Kansas City, Mo , and 25 Wint Ave., Fort Leavenworth, Kan
FOLEY, John J. (*A* 1938) Pres , Weathermakers (Canada) Ltd , 593 Adelaide St., W., Toronto, Ont , Canada.
FOLEY, J. Lester (*M* 1938) Mgr , Wholesale Dept , Avery Engineering Co , 2341 Carnegie Ave , Cleveland, and (for mail) 3567 Riedham Rd , Shaker Heights, O.
FOLLETT, Thomas L. (*S* 1936) 10900 Euclid Ave , Cleveland, and (for mail) 306 N. Main St , Hudson, O
FOLSOM, Rolfe A. (*M* 1938) Vice-Pres (for mail) W. R Ames Co , 150 Hooper St., San Francisco, and 2411 Easton Drive, Burlingame, Calif
FOOTE, A. G. (*M* 1937) Dist Engr (for mail) Frigidaire Div , General Motors Sales Corp., Oakland Dist , 1250-53rd St , Oakland, and 2466 Virginia St , Berkeley, Calif
FOOTE, Earl E. (*M* 1936) Gen Supt Consumers Central Heating Co , 108 East 11th St , and (for mail) 3412 North 28th St., Tacoma, Wash
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FORD, Edward F. (*A* 1937) Sales Repr (for mail) American Radiator Co , 8019 Joseph Campau Ave., Detroit, and 288 Ann St , Plymouth, Mich
FORDERBRUGGEN, Kevin J. (*J* 1938) Engr. (for mail) Mankato Natural Gas Co , 222 S Front St., and Ben Pay Hotel, Mankato, Minn.
FORFAR, Donald M. (*M* 1917) Mech. Engr , Grinnell Co , Inc , 240-7th Ave S., and (for mail) 4817 Emerson Ave., S., Minneapolis, Minn.
FORRESTER, Charles M. (*A* 1937) Air Cond. Sales Mgr. (for mail) Gurney Foundry Co., Ltd., 4 Junction Rd., and 77 Collegeview Ave., Toronto, Ont., Canada.
FORRESTER, Norman J. (*A* 1936) Mgr. Contract Div , Garth Co , 316 Westminster Ave , N., Montreal, W., P Q , Canada.
FORSBERG, William (*M* 1919) Secy. & Supt. (for mail) The Hopson & Chapin Mfg Co , 231 State St., New London, and Quaker Hill, Conn
FORSLUND, Oliver A. (*M* 1936) Gen Mgr. and Partner, Forslund Pump and Machinery Co , 1717-19 Main St , and (for mail) 108th St. & State Line, Kansas City, Mo
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FOSTER, John G. (*J* 1938) Sales Engr. (for mail) Air Conditioning Utilities Co , 8 West 40th St., and 2635 Sedgwick Ave , New York, N Y
FOSTER, Philip H. (*A* 1937) Mgr (for mail) Hudson Bay Plumbing Co , and Flin Flon Hotel, Flin Flon, Man , Canada.
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FOWLES, Harry H. (*J* 1934) Htg Engr (for mail) Carman-Thompson Co , 12-14 Lincoln St , Lewiston, and 176 Summer St., Auburn, Me
FOX, Edward L. (*J* 1936) Engr , American Foundry & Furnace Co , Bloomington, and (for mail) 715 Sanford St , Peoria, Ill
FOX, Ernest (*M* 1935) Asst to Engr (for mail) C. A. Dunham Co , Ltd , 1523 Davenport Rd., and 409 Glenholme Ave , Toronto, Ont , Canada.
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FRANCIS, Paul E. (*M* 1937) Asst Mgr of Sales, Northwestern Fuel Co , E-1203 First National Bank Bldg , St Paul, and (for mail) 5115 S. Colfax Ave , Minneapolis, Minn.
FRANCK, Peter (*J* 1938) Secy , Tiltz Air Conditioning Corp , 230 Park Ave , New York, and (for mail) 478 State St , Brooklyn, N Y
FRANK, John M. (*M* 1918, *A* 1912) Pres (for mail) Ilg Electric Ventilating Co , 2850 N Crawford Ave , Chicago, and 1152 Chatfield Rd , Hubbard Woods, Ill
FRANK, Olive E.* (*M* 1919) Vice-Pres , Frank Heaters, Inc , 150 Railroad Ave , and (for mail) 288 Graham Ave , Paterson, N J
FRANKEL, Gilbert S. (*M* 1926) Mgr , Federal and Marine Dept (for mail) Buffalo Forge Co , 820-24 Woodward Bldg , and 3000 Connecticut Ave , Washington, D C
FRANKLIN, Ralph S. (*M* 1919) Pres -Treas. (for mail) Albert B Franklin, Inc , 38 Chauncy St., Boston, and 320 Grove St , Melrose, Mass.
FRANKLIN, Sam H., Jr. (*A* 1938) Prop (for mail) S H Franklin, Jr., Heating Contractor, 921 Main St., and 204 Colonial Court, Lynchburg, Va.
FRANKLIN, Stephen D. (*A* 1938) Engr , Standard Oil Co of Pennsylvania, and (for mail) 5016 Greene St , Philadelphia, Pa
FRASER, James J. (*A* 1936) Director (for mail) Honeywell-Brown, Ltd , 70 St Thomas St , London, S E 1, and 60, The Grove, St Margaret's, Twickenham, Middlesex, England.
FRAZIER, J. Earl (*A* 1936) Vice-Pres and Treas (for mail) Frazier-Simplex, Inc , 436 East Beau St , and 417 East Beau St , Washington, Pa.
FREAS, Royal B. (*M* 1928) Vice-Pres., Freas Thermo Electric Co , 1750 N Springfield Ave , Chicago, Ill., and (for mail) Schodack Landing, N. Y.

FREDERICK, Holmes W. (*M* 1937) Asst Htg. Engr., Cornell University, Merrill Hall, and (for mail) 103 Harvard Place, Ithaca, N. Y.
FREDERICK, Kendall C. (*A* 1938) Sales Engr. (for mail) Sidles Co., Airtemp Div, 502 South 19th St, and 1515 Park Ave, Omaha, Nebr
FREDERICK, Walter L. (*A* 1937) Pres (for mail) Bryant Air Conditioning Corp, 1340 Connecticut Ave, and 3016 Tilden St, Washington, D. C.
FREEMAN, Edwin M. (*A* 1937) Vice-Pres & Sales (for mail) Canadian Asbestos Co, 316-322 Youville Sq., and 37 Sunset Ave, Montreal, P Q., Canada
FREEMAN, J. Albert (*J* 1938) Partner, Engr, Western Engrg. Co, 1611 S E 9th Ave, and (for mail) 3143 N. E Wasco St, Portland, Ore
FREEMAN, John C. (*J* 1936) Associate Mech Engr. (for mail) Div. of Architecture, and 2214-23rd St, Sacramento, Calif.
FREITAG, Frederic G. (*M* 1932) Engr., Sylvester Oil Co, Inc, 703 S. Columbia Ave, and (for mail) 9 Harrison St, Mt. Vernon, N. Y
FREITAS, Leo J. (*A* 1938) Branch Mgr (for mail) Fedders Manufacturing Co, Inc., 1036 Beaubien St., and 15429 Appoline, Detroit, Mich
FRENCH, Donald (*M* 1926) Vice-Pres (for mail) Carrier Corp, 302 S Geddes St., and 210 Brattle Rd., Syracuse, N. Y.
FRENTZEL, Herman C. (*M* 1936) Chief Engr, The Heil Co, 3000 W Montana St, and (for mail) 4363 N. Wildwood Ave., Milwaukee, Wis
FRIED, Harold V. (*A* 1935) Dealer Repr (for mail) Birmingham Electric Co, 2100 First Ave, N, and 2640 Canterbury Rd, Birmingham, Ala
FRIEDLINE, James M. (*J* 1937) Engr, General Air Conditioning Corp, Paramount Bldg, and (for mail) 1811-5th Ave, S E, Cedar Rapids, Ia
FRIEDMAN, Arthur (*A* 1936) (for mail) Air Controls, Inc, 1933 West 114th St, Cleveland, and 15700 S Moreland Blvd, Shaker Heights, O
FRIEDMAN, D. Harry, Jr. (*M* 1936) Air Cond & Industrial Engr (for mail) Peoples Water and Gas Co, 15th & Washington Ave, Miami Beach, and 1309 Brickell Ave, Miami, Fla
FRIEDMAN, Ferdinand J.* (*M* 1921) Mech Engr (for mail) McDougall & Friedman, 1221 Osborne St, Montreal, P Q., Canada, and 31 Union Square, New York, N. Y.
FRIEDMAN, Milton (*J* 1935; *S* 1933) c/o H. F Klawuhn, Genl Contractor, 34-24 82nd St., Jackson Heights, L I, and (for mail) 470 West End Ave, New York, N Y.
FRIMET, Maurice (*J* 1936) Engr., S I Heating & Air Cond Co, 28 Oxford Pl, Tompkinsville, and (for mail) 15 Mundy Ave, West Brighton, S I, N Y.
FRITZ, Charles V. (*J* 1936; *S* 1933) Designer and Estimator, Charles F. Fritz (for mail) 67 W. Merrick Rd, and 26 Cottage Court, Freeport, N Y
FUKUI, Kunitaro (*M* 1926) Auditor, Oriental Carrier Engineering Co, Ltd, Toyo Menka Bldg., Koraibashi-Sanchome, Osaka, Japan
FULLER, Elbridge W. (*M* 1938) Mgr, Commercial Air Conditioning-Delco Frigidaire, Taylar St., and (for mail) 1916 Emerson Ave, Apt. 5, Dayton, O.

G

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GALE, Hamilton A. (*J* 1936) Engr., Wallace Stebbins & Sons, Inc., 100 S. Charles St, Baltimore, and (for mail) Murray Hill, Annapolis, Md
GALLAGHER, Frank H. (*A* 1938) Asst. Chief Engr., Board of Public Education, Forbes St at Bellefield Ave., and (for mail) 2727 Strachan Ave, Pittsburgh, Pa.
GALLIGAN, Andrew B. (*M* 1921) 716 South 51st St., Philadelphia, Pa.
GALLAWAY, James F. (*A* 1938; *S* 1934) General Electric Co, 570 Lexington Ave, New York, and (for mail) 117-01 Park Lane, S, Kew Gardens, L I, N Y
GAMBLE, Cary B. (*A* 1935) Consulting Engr. (for mail) Leo S Weil & Walter B Moses, 427 S Peters St., and 732 St. Peter St., New Orleans, La
GAMMILL, Oscar E., Jr. (*A* 1937; *J* 1930) Sales Engr (for mail) Carrier Corp, 1413 Hibernia Bank Bldg, and 5515 Magnolia St., New Orleans, La
GANGE, Frank B. (*M* 1937) Managing Director, Gordon & Co, Ltd, 185 Yuen Ming Yuen Rd., Shanghai, China
GANT, H. P.* (*M* 1915) (*Presidential Member*) (Pres, 1923; 1st Vice-Pres, 1922; 2nd Vice-Pres., 1921; Council, 1918-1924), R. D No. 1, Glenmoore, Pa
GARBER, William E., Jr. (*J* 1938) Field Engr., Farquhar Furnace Co., Wilmington O, and (for mail) 3643 Graceland Ave, Indianapolis, Ind.
GARDNER, Clifton R. (*A* 1937) Vice-Pres (for mail) Martyn Brothers, Inc, 911 Camp St., Dallas, and 3708 Watonga, Fort Worth, Tex.
GARDNER, S. Franklin (*M* 1911) Pres (for mail) Standard Engineering Co, 2129 Eye St, N. W., and 4901 Hillbrook Lane, Washington, D. C.
GARDNER, William, Jr. (*A* 1921) Vice-Pres. (for mail) Garden City Fan Co, 1842 McCormick Bldg, and 7836 Loomis Blvd, Chicago, Ill
GARNEAU, Leo (*M* 1938; *J* 1930) Sales Engr (for mail) C A Dunham Co., Ltd., 931 Dom. Square Bldg, and 2541 Maplewood Ave. (Apt. 2), Montreal, P Q, Canada
GARNETT, Ralph E. (*A* 1936) Sales Engr., 3114 Benton Blvd, Kansas City, Mo.
GATES, Robert A. (*M* 1936) Sole Owner, Gates Engineering Co, 510-77 Sheet St, and (for mail) 248 Bay 38th St, Brooklyn, N Y.
GAULEY, Ernest R. (*A* 1935) Pres (for mail) Age Publications, Ltd., 31 Willcocks St., and 156 Dewhurst Blvd, Toronto, Ont, Canada.
GAULT, George W. (*J* 1937; *S* 1934) Army Officer, 2nd Lt, C. C C Camp (for mail) C C C., Co 2340, S-118, and 216 W. Market St., Clearfield, Pa.
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GAWTHROP, Fred H. (*M* 1919) Pres, Gawthrop & Bro Co, 705 Orange St., and (for mail) 2211 Shallcross Ave, Wilmington, Del.
GAYLOR, William S. (*M* 1919) 161 Pelham Rd., New Rochelle, N. Y.
GAYLORD, Frank H. (*M* 1921) Western Sales Mgr. (for mail) Hoffman Specialty Co, Inc., 130 N Wells St, Chicago, and 362 N. York St, Elmhurst, Ill
GAYMAN, Paul D. (*M* 1938) Branch Mgr (for mail) Johnson Service Co, 2142 East 19th St., Cleveland and 20875 Endsley Ave., Rocky River, O.
GAYNER, James (*M* 1937) Mech. Engr., G. M. Simonson, Cons. Engr., 74 New Montgomery St., San Francisco, and (for mail) 239 Park View Ave., Piedmont, Calif
GEE, William W., Jr. (*J* 1938) Designing Engr. (for mail) Graham and Gee, 133 Geary St., and 1670 Chestnut St., San Francisco, Calif.
GEIGER, Irvin H. (*M* 1919) Registered Prof. Engr. & Mfrs Repr. (for mail) 319 Telegraph Bldg., and 240 Maclay St., Harrisburg, Pa.
GELTZ, Ralph W. (*J* 1936) Air Cond. Engr., York Ice Machinery Corp., 2700 Washington Ave., Cleveland, and (for mail) 14213 Glenside Rd, Cleveland, O.

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GERHARD, David H. (*A* 1937) Power Sales Engr (for mail) Consumers Power Co., and 121 S. Higby, Jackson, Mich

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GERRISH, Grenville B. (*A* 1936, *J* 1930) Mgr (for mail) Fitzgibbons Boiler Co, Inc, 31 Main St, Cambridge, and Standish Rd, Melrose, Mass

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GIFFORD, Robert L. (*Life Member*; *M* 1908) Pres, Illinois Engineering Co, Cor. 21st St & Racine Ave, Chicago, Ill, and (for mail) 1231 S El Molino Ave, Pasadena, Calif

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GILFRIN, George F. (*M* 1932) Climas Artificiales, S A (for mail) Edificio "La Nacional" 608, and Esplanada No. 715 Lomas de Chapultepec, Mexico, D F.

GILL, Eric F. (*M* 1936) Chief Draftsman, Drayton Regulator & Instrument Co, Ltd, and (for mail) 30 Warwick Rd, West Drayton, Middlesex, England

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GILLETT, M. C. (*M* 1916) Dist. Sales Engr, Hoffman Specialty Co., Inc, and (for mail) 6600 Rising Sun Ave, Philadelphia, Pa

GILLHAM, Walter E. (*M* 1917) Consulting Engr. (for mail) 337 Law Bldg., and 3427 Bellfontain Ave, Kansas City, Mo

GILMAN, Franklin W. (*M* 1935) Plant Engr, Atwater Kent Mfg. Co, 4700 Wissahickon Ave, and (for mail) 514 W. Coulter St, Philadelphia, Pa.

GILMORE, John L. (*A* 1938) Owner, John L. Gilmore Htg-Vtg, 1602 Kay Ave, and (for mail) 1604 Union St, Brunswick, Ga

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GITTLESON, Harold (*A* 1936) Sales Mgr, Lariviere, Inc, 3715 St Lawrence Blvd, and (for mail) 1125 Lajoie Ave., Montreal, P Q, Canada

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GOENAGA, Roger C. (*M* 1931) Tech Director, Ateliers Ventil (for mail) 109 Cours Gambetta, Lyon, and 33 Avenue Valioud-Ste-Foy-les-Lyon, Rhone, France

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GONZALEZ, Rafael A. (*M* 1936) Mgr, Application Engrg Dept (for mail) Airtemp Div, Chrysler Corp, P O Box 1037, and 434 Delaware St, Dayton, O

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GORDON, Edward B., Jr. (*M* 1908) Pres, Pillsbury Engineering Co, 1200 Second Ave, S, and (for mail) 2450 West 24th St, Minneapolis, Minn

GORDON, Henry H. W. (*J* 1938) Asst Resident Engr (for mail) Carrier Engineering S A, 91 Smith St, Durban Natal, and 71 Pasadena Court, South Beach, Durban, South Africa.

GORDON, Peter B. (*A* 1938; *J* 1935) Treas (for mail) Wolff & Munier, Inc, 222 East 41st St, New York, N. Y, and 35 Park Ave, Bloomfield, N. J.

GORDON, William D. (*A* 1935) Air Cond and Sales Engr., Hart & Cooley Mfg Co of Canada, Ltd, Fort Erie, N., and (for mail) 71 Chilton Rd, Toronto, Ont, Canada

GORNSTON, Michael H. (*A* 1923) Custodian-Engr (for mail) Board of Education, Thomas Jefferson High School, Brooklyn, and 90-11-149th St., Jamaica, N Y

GOSS, Matthew H. (*M* 1921) Partner (for mail) M. H. Goss Co, 3536 Baldwin Ave, and 3417 Field Ave, Detroit, Mich

GOSSETT, Arthur L. (*A* 1938) Gage Bros, 617 Red River, Austin, Tex

GOSSETT, Earl J. (*M* 1923) Pres (for mail) Bell & Gossett Co, 3000 Wallace St, Chicago, and 314 Woodland Ave, Winnetka, Ill

GOTHARD, William W. (*A* 1936) Editorial Director (for mail) Domestic Engineering, 1900 Prairie Ave, Chicago, and 1027 Arlington Ave, LaGrange, Ill

GOTSCHALL, Harry C. (*M* 1935) Chairman, Air Cond. Dept, Lane Technical High School, 2501 W. Addison St., and (for mail) 2953 Eastwood Ave, Chicago, Ill

GOTTWALD, C. (*A* 1916) Pres (for mail) The Ric-wiL Co, 1563 Union Commerce Bldg, Cleveland, and 2225 Stillman Rd, Cleveland Heights, O.

GOUEDY, Kenneth E. (*A* 1935) Member of Firm and Engr (for mail) Modern Building Insulating Co, 411 Bona Allen Bldg, Atlanta, and 218 Columbia Drive, Decatur, Ga

GOULD, Henry E. (*J* 1936) Secy (for mail) Natkin & Co, 1800 Baltimore, and 6528 Summit, Kansas City, Mo

GOULDING, William (*A* 1933) Air Cond Engr, World Broadcasting System, 711 Fifth Ave, New York, and (for mail) 782 Westminster Rd., Brooklyn, N Y

GOUNDIE, Joseph K. (*M* 1938) Sales Engr, Fritch Coal Co, 116 River St, Bethlehem, and (for mail) 1426 Walnut St, Allentown, Pa.

GRABENSTEDER, Louis (*A* 1937, *J* 1935) 3206 Linnet Rd, Louisville, Ky.

GRABER, Ernst (*J* 1936) Engr, Minneapolis-Honeywell Regulator Co, 801 Second Ave, New York, and (for mail) 222 Hollywood Ave, Douglaston, L. I, N Y

GRABMAN, Henry B. (*S* 1938) Student, Carnegie Institute of Technology, Pittsburgh, and (for mail) 355 E Spring St, Zelienople, Pa.

GRAFF, William F. (*A* 1937) Salesman-Engr, Standard Sanitary Mfg Co, and (for mail) 940 Jefferson, S E., Grand Rapids, Mich.

GRAHAM, John M. (*A* 1937; *J* 1936) Sales Engr. (for mail) B F. Sturtevant Co, 528 Kentucky Home Life Bldg., and Puritan Apts, Louisville, Ky.

GRAHAM, William D. (*M* 1929; *A* 1925; *J* 1923) Sales Dept, Carrier Corp, and (for mail) 129 Circle Rd, Syracuse, N. Y.

GRANSTON, Ray O. (*J* 1935; *S* 1930) Secy. (for mail) University Plbg & Htg Co., 3939 University Way, and 4333-9 Ave., N. E., Seattle, Wash.

GRANT, Walter A. (*A* 1933; *J* 1928) Dist. Chief Engr, Carrier Corp, and (for mail) 236 Shotwell Park, Syracuse, N Y.

GRAVES, Willard B. (*Life Member*; *M* 1906) Pres (for mail) W. B Graves Heating Co., 162 North Desplaines St, and 5920 Addison St., Chicago, Ill

GRAY, Earle W. (*M* 1938, *A* 1934) In charge of Air Cond, Commercial Dept (for mail) Oklahoma Gas and Electric Co, Third and Harvey Sts, and 2125 N W 18th St, Oklahoma City, Okla

GRAY, Everett W. (*M* 1936) Mgr (for mail) The Trane Co, 1900 Euclid Ave, Cleveland, and 17545 Madison Ave., Lakewood, O

GRAY, George A. (*M* 1924) Branch Mgr (for mail) C A Dunham Co, Ltd, 404 Plaza Bldg, and 114 Belmont Ave, Ottawa, Ont, Canada

GRAY, John W. (*M* 1938) Member of Firm, The Gray-Henry Co, 614 N Water St, Bay City, Mich

GRAY, William E. (*M* 1922) Gray Engineering Co, and (for mail) Box 264, High Point, N C.

GREEN, Everett W. (*J* 1938) Junior Sales Engr (for mail) Green Furnace and Plumbing Co., 2815 North 48, and 5100 Leighton Ave, Lincoln, Nebr

GREEN, William C. (*Life Member*, *M* 1906) Dist Repr (for mail) Warren Webster & Co, 704 Race St, and 244 Erkenbrecher Ave, Cincinnati, O

GREENBERG, Irving (*S* 1937) Air Cond Engr, S Greenberg, 873 Columbus Ave., New York, and (for mail) 1615 Walton Ave, Bronx, N Y.

GREENBURG, Dr. Leonard* (*M* 1932) Executive Director, Div of Industrial Hygiene (for mail) N Y State Dept of Labor, 80 Centre St, and 173 West 78th St, New York, N Y.

GREENLAND, Sidney F. (*M* 1934) Htg & Vtg Engr, Gee Walker & Slater, 3, Fitzmaurice Place, London, W 1., and (for mail) 9, Anerley Court, Anerley Park, London, S E 20, England

GREENWOOD, Orrin J. (*M* 1938) Air Cond Engr, Walgreen Co, 744 Bowen Ave, and (for mail) 6357 S Homan Ave, Chicago, Ill

GREGG, Scranton H. (*A* 1936) Pres, Shellen-Berger-Gregg Co, 2203 N. Prospect Ave., and (for mail) 5134 N Woodburn St, Milwaukee, Wis

GREGG, Stephen L. (*J* 1936) Sales Engr (for mail) Potomac Electric Power Co, 10th & E Sts, N W, Washington, D C, and 4828 Edgemoor Lane, Bethesda, Md

GREINER, George E., Jr. (*J* 1938, *S* 1935) Engr, Wayne Crouse, Inc, 4647 Centre Ave, and (for mail) 5515 Claybourne St, Pittsburgh, Pa

GRIESS, Philip G. (*M* 1937) Mech Engr, Voorhees, Gmelin & Walker, 101 Park Ave, New York, N. Y, and (for mail) 189 Walnut Ave, Bogota, N J

GRIESSER, Charles E. (*A* 1936) Owner, Electric Contractor Dealer, Bryan, Tex

GRIEST, Kermit (*J* 1936) Sheet Metal Worker and Sales, Frank-Limbach & Co, 1722 E Ohio St, and (for mail) 437 Bausman St, Pittsburgh, Pa

GRIEVES, Thomas R. (*A* 1930) Branch Mgr (for mail) U S Radiator Corp., 303 Crosby Bldg., Buffalo, N Y.

GRIEWISCH, Alfred H. (*A* 1938) Pres (for mail) Bayley Heating Supply Co., 2045 W St Paul Ave, and 2557 N. 47th St, Milwaukee, Wis

GRIFFITH, Claude A. (*A* 1938) Assoc Member, Heating & Air Conditioning Service, 632 Hill Top Drive, Cumberland, Md.

GRIFFITH, Herbert T. (*M* 1938) Designing Engr (for mail) Lincoln Bouillon (Consult. Engr) 324-1411-4th Ave. Bldg, and 1909-3rd Ave, W., Seattle, Wash

GRIFFITH, Joseph B. (*J* 1938) Sales Engr., Drayer and Hanson, 738 E Pico Blvd, Los Angeles, and (for mail) 415 N. Sierra Vista Ave., Monterey Park, Calif

GRIMES, Fenner M. (*J* 1935) Junior Engr, War Dept., O Q M G. (Constr Div), Munitions Bldg, and (for mail) 2301 Cathedral Ave, N W, Washington, D C.

GROHS, Conrad E. (*J* 1938) Washington Repr. (for mail) American Gas Products, 4th & Channing Sts, N E, and 1349 Newton St, N. E, Washington, D. C.

GROOT, Harry W. (*M* 1937) Chief Engr, The Home Comfortable, Inc, 226 West Walnut St., and (for mail) 3728 N. Western Pkwy, Louisville, Ky.

GROSS, Lyman C. (*M* 1931) Sales Engr, Minneapolis-Honeywell Regulator Co, 2727 4th Ave, S., and (for mail) 5324 Oaklawn Ave., Linden Hills Station, Minneapolis, Minn

GROSSENBACHER, Henry E. (*A* 1938) Pres (for mail) Grossenbacher Steel Furnace & Mfg Co., 2626 Woodson Rd., and 9741 Lackland Rd., Overland, St Louis Co., Mo

GROSSMAN, Franklin A. (*J* 1938; *S* 1937) Experimental Engrg, Servel, Inc, Engrg Dept, 119 N. Morton Ave., and (for mail) 1161 E. Illinois St, Evansville, Ind

GROSSMAN, Harry E. (*A* 1933, *J* 1927) Sales Repr, Haynes Selling Co, Inc, Ridge and Spring Garden Sts, Philadelphia, and (for mail) 218 Parham Rd, Springfield, Pa

GROSSMANN, Harry A. (*M* 1931) Owner, H A Grossmann Co, 3138 Cass Ave, and (for mail) 3122 Geyer Ave, St Louis, Mo

GROSVOLD, Fred E. (*M* 1917) Owner (for mail) F E Grosvold Co, 417 Wisconsin St, and 711 Grand Ave, E, Eau Claire, Wis

GROVES, Samuel A. (*J* 1935) Sales Engr, American Radiator Co, 40 West 40th St, New York, and (for mail) 21 Cassilis Ave, Bronxville, N Y

GULER, George D. (*A* 1937) Sales Engr (for mail) Minneapolis-Honeywell Regulator Co, Wayne & Roberts St, and 334 E Allens Lane, Philadelphia, Pa

GUMAER, P. Wilcox (*M* 1937) Consulting Engr, Toxic Vapors & Dusts, 25 Garden St, West Englewood, N J

GUNNELL, George T (*M* 1937) Chief Htg Engr (for mail) Sunbeam Heating & Air Conditioning Co, 346 Peachtree St, N E, and 595 Ashby, S W, Atlanta, Ga

GURNEY, E. Holt (*M* 1929) (Pres, 1938, 1st Vice-Pres, 1937, 2nd Vice-Pres, 1936, Council, 1931-1938) Pres. (for mail) The Gurney Foundry Co, Ltd, 4 Junction Rd, and 347 Walmer Rd, Toronto, Ont, Canada

GURNEY, Edward R. (*J* 1937) Asst to Plant Supt (for mail) Gurney Foundry Co, Ltd, 4 Junction Rd, and 50 Eastbourne Ave, Toronto, Ont, Canada

GUSTAFSON, Carl A. (*M* 1938) Sales Engr (for mail) The Powers Regulator Co, 2720 Greenview Ave, and 5920 Ridge Ave, Chicago, Ill

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HACKETT, H. Berkeley (*M* 1921) Consulting Engr., 901 Architects Bldg, 17th and Sansom Sts, and (for mail) The Lenox, 13th and Spruce Sts, Philadelphia, Pa

HADDOCK, Isaac T. (*A* 1926) New England Gas & Electric Association, 719 Massachusetts Ave, Chicago, Ill

HADEN, G. Nelson (*M* 1934, *A* 1928; *J* 1922) Chairman and Managing Dir. (for mail) G N Haden & Sons, Ltd, 60 Kingsway, London, W C 2, and 36 Wildwood Rd, London, N W 11, England

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HADJISKY, Joseph N. (*M* 1930) Consulting Engr., 744 Bates St., Birmingham, Mich.

HAERLE, Robert A. (*A* 1938) Mech. Engr., Bayley Blower Co., 1817 S. 66th St, and (for mail) 1438 N. Humboldt Ave, Milwaukee, Wis.

HAGAN, William V. (*M* 1938; *A* 1933; *J* 1926) Secy., V. J. Hagan Co., and (for mail) 1811 Jones St, Sioux City, Ia

HAGEDON, Charles H. (*M* 1919) Partner (for mail) S E Fenstermaker & Co, 937 Archt & Bldrs Bldg, and 4156 Broadway, Indianapolis, Ind

HAHN, Roy F. (*J* 1936) Air Cond Engr (for mail) Advance Refrigeration, Inc., 350 Peachtree St., and 435-10th St, N E, Atlanta, Ga

HAINES, John J. (*M* 1915) Pres (for mail) The Haines Co, 1931 W Lake St, Chicago, and 623-17th Ave, Maywood, Ill

HAITMANEK, Louis M. (*A* 1938) Journeyman Sheet Metal Worker, 217 Rose St, Newark, N. J.

HAJEK, William J. (*M* 1932) 372 W Johnson St, Philadelphia, Pa

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HALE, Fred J. (*M* 1936) Mgr (for mail) Empire Sheet Metal Works, Ltd, 1606 West First Ave, and 3606 Point Grey Rd, Vancouver, B. C, Canada

HALE, John F. (*Life Member*, *M* 1902) (*Presidential Member*) (Pres, 1913, 1st Vice-Pres, 1912, Board of Governors, 1908-1910, 1912-1913) Dist Mgr (for mail) Aerofin Corp, Room 704, 111 W Washington St, Chicago, and 400 S. LaGrange Rd, LaGrange, Ill

HALEY, Harry S * (*M* 1914) Consulting Engr, Partner (for mail) Leland & Haley, 58 Sutter St., and 735-21st Ave, San Francisco, Calif

HALEY, Robert T. (*A* 1938) Supervisor of House Heating (for mail) Minneapolis Gas Co, 800 Hennipin Ave, and 5024-12th Ave, S, Minneapolis, Minn

HALL, George (*A* 1937) Secy-Treas and Mgr (for mail) Hyland, Hall & Co, 115 E Doty St, and 4201 Wanetah Trail, Nakoma, Madison, Wis

HALL, John R. (*M* 1937, *J* 1932) Mech Engr, 1416 Lakeview Ave, Minneapolis, Minn

HALL, Mora S (*M* 1934) Development Engr (for mail) Anthracite Industries Laboratory, Primos, Del Co, and 293 N Maple Ave, Lansdowne, Pa

HALL, Wilton L. (*M* 1938) Engr, Frank A Leon Co, 901 Girard St, N E, and (for mail) 5316 Dorsett Pl, N W, Washington, D C

HALLAR, Edgar V. (*A* 1937) Sales, Lane-White Electric Co, 204 West 4th St, Joplin, and (for mail) 1001 S Madison, Webb City, Mo

HALLECK, Leon P. (*A* 1937) Vice-Pres & Sales Mgr (for mail) The Allen Corporation, 9751 Erwin Ave, and 12049 Roselawn Ave, Detroit, Mich

HALLER, Arthur L. (*M* 1920) Pres-Treas (for mail) Haller Appliance Sales Co, Inc, 3321 Washington, St Louis, and 124 W Cedar Ave, Webster Groves, Mo

HALLSTEIN, Harry T. (*J* 1938) Dist Engr (for mail) Swett Bros Heating & Appliance Co, 559 State St, Springfield, Mass

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HAMAKER, Ambrose C (*A* 1937) Sales Engr. (for mail) Mayflower-Lewis Corp, 63 W Milwaukee Ave, and 18624 Santa Rosa Drive, Detroit, Mich

HAMERSKI, Francis D. (*J* 1934) Winona Coal Co, Winona, Minn

HAMIG, Louis L. (*J* 1935) Engr, 3514 Utah St, St Louis, Mo

HAMILTON, Lloyd L. (*J* 1938) Branch Office Mgr, Minneapolis-Honeywell Regulator Co, and (for mail) 208 Norwood Ave, N. E, Atlanta, Ga

HAMJE, Milton C. (*J* 1936) Engr, Syska & Hennessy, Consulting Engrs, 420 Lexington Ave, New York, and (for mail) 198 Hancock St, Brooklyn, N Y.

HAMLET, Francis A. (*A* 1936) Branch Mgr. (for mail) C. A. Dunham Co., Ltd., Room 931, Dominion Square Bldg., 1010 St. Catherine St., W., and 3550 Shuter St, Montreal, P Q, Canada.

HAMLET, Thomas F. (*M* 1938) Sales Engr (for mail) Taylor Forbes, Ltd, 1197 University St, Montreal, and 34 Burton Ave, Westmount, P Q, Canada.

HAMLIN, James B., Jr. (*A* 1937) Htg Engr, Crane Co., 14 W Broad St, and (for mail) 1530 East 51st St, Savannah, Ga

HANBURGER, Fred W. (*M* 1930) Consulting Engr, 252 West 76th St, New York, N Y

HANLEIN, Joseph H. (*M* 1937) Secy-Treas (for mail) Wilberding Co, Inc, 808-17th St, N W, Room 13, and 5420 Connecticut Ave, Washington, D C

HANLEY, Edward V. (*A* 1933) Pres (for mail) S V. Hanley Co, 1653 N Farwell Ave, Milwaukee, and 844 E Birch Ave, Whitefish Bay, Wis

HANLEY, Thomas F., Jr. (*M* 1933) Pres (for mail) Hanley & Co, 1503 S Michigan Ave, and 1640 E 50th St, Chicago, Ill

HANSLER, John E (*M* 1937) Zone Service Repr, Delco-Frigidaire Conditioning Div., 300 Taylor St, and (for mail) 2621 Hillview Ave, Dayton, O

HANSON, Leslie P. (*M* 1937, *A* 1936; *J* 1935, *S* 1933) Engr, U S Air Conditioning Corp, 2101 Kennedy, N E, and (for mail) 5027 Nokomis Ave, S, Minneapolis, Minn

HARBAUGH, Jacob W. (*M* 1937) Supt of Erection, Kupferle-Hicks Heating Co, 3974 Delmar Blvd, St Louis, and (for mail) 607 Lilac St, Webster Groves, Mo

HARBORDT, Otto E. (*A* 1936) Sales Mgr (for mail) U S Supply Co, 1315 West 12th St, and 303 Brush Creek Blvd, Kansas City, Mo

HARD, Amos L. (*A* 1938) Chief Engr, Thos. Emery & Sons Co, Carew Tower, and (for mail) 910 Kreis Lane, Cincinnati, O

HARDEN, J. Clinton (*M* 1938) Htg Engr, Round Oak Co, and (for mail) 106 Courtland St, Dowagiac, Mich

HARDING, Edward R. (*M* 1936) N C State Sales Engr (for mail) Kewanee Boiler Corp, P O Box 536, 704 Jefferson Bldg, Greensboro, and Guildford College, N C

HARDING, Louis A.* (*M* 1911) (*Presidential Member*) (Pres, 1930, 1st Vice-Pres, 1929, 2nd Vice-Pres, 1928, Council, 1922-1931) Commissioner of Public Works, City Hall, and (for mail) 85 Cleveland Ave, Buffalo, N Y

HARDY, Frank L. (*J* 1937) Secy (for mail) Gulf-York Co, 2300-3rd Ave, N, and 2312 Highland Ave, Birmingham, Ala

HARMONAY, William L. (*A* 1935) Mgr (for mail) M J Harmonay, Inc, 124 Elm St, and 34 Alida St, Yonkers, N Y

HARRIGAN, Edward M. (*M* 1915) Gen Mgr (for mail) Harrigan & Reid Co, 1365 Bagley Ave, and 7450 LaSalle Blvd, Detroit, Mich

HARRINGTON, Charles (*M* 1923) 43 Indian Grove, Toronto, Ont, Canada

HARRINGTON, Elliott* (*M* 1932, *A* 1930) Mgr, Commercial Engrg Div, Air Cond Dept (for mail) General Electric Co, 5 Lawrence St, Bloomfield, and 17 Wilson Terrace, West Caldwell, N J

HARRIS, Albert M. (*M* 1938) Sales Mgr, Air Cond. (for mail) Baker Ice Machine Co, 509 E 3rd St, and 2941 Glen Garden Drive, Fort Worth, Tex

HARRIS, Jesse B. (*M* 1918) Co-Partner (for mail) Rose & Harris Engineers, 416 Essex Bldg, and 3620 Colfax Ave, S, Minneapolis, Minn

HARRIS, John G. (*M* 1936) Dist Repr (for mail) Frigidaire Div, General Motors Sales Corp, Terminal Tower Bldg, Cleveland, and 14432 Delaware Ave, Lakewood, O

HARRISON, George G. (*M* 1937) Chief Engr, S T Johnson Co, 940 Arlington Ave, Oakland, and (for mail) 1428 Arch St, Berkeley, Calif

HARRISON, Julius C. (*M* 1938) Design Engr. (air conditioning) Texas Air Conditioning Co. (for mail) 407 Capps Bldg, and 919 W Cannon St, Fort Worth, Tex

HARROWER, William C. (*A* 1937) Air Cond. Draftsman, Gar Wood Industries, 409 Connecticut Ave., and (for mail) 12561 Third Ave, Highland Park, Mich

HARSCH, Richard J. (*M* 1936) Naval Archt., U S Government, Navy Yard, and (for mail) 142 Avenue O, Brooklyn, N. Y.

HART, F. Donald (*J* 1937) Air Cond. Engr. (for mail) E I DuPont de Nemours & Co, and 1301 Van Buren St, Wilmington, Del.

HART, Harry M.* (*M* 1912) (*Presidential Member*) (Pres, 1916, 1st Vice-Pres, 1915; Council, 1914-1917) Pres (for mail) L H. Prentice Co, 1048 Van Buren St, and 3730 Lakeshore Drive, Chicago, Ill

HART, Stanley (*M* 1938) Vice-Pres. (for mail) Tuttle & Bailey, Inc, and New Britain, Conn.

HART, Theodore S. (*M* 1938) Engr, Tuttle & Bailey, Inc, New Britain, Conn

HART-BAKER, Henry W. (*M* 1918) Hart Engineering Co, 451 Kiangse Rd, Shanghai, China.

HARTIN, William R, Jr. (*J* 1935) Htg. Engr., Vice-Pres-Secy. (for mail) W R Hartin & Son, Inc, 2123 Green St, and 212 S. Saluda Ave, Columbia, S C

HARTLINE, W. Raymond (*A* 1936) Sales Engr, 5216 Kansas Ave, N W, Washington, D C.

HARTMAN, John M. (*M* 1927) Engr (for mail) Kewanee Boiler Corp, and 719 Henry St., Kewanee, Ill

HARTON, A. J. (*A* 1935) Sales Engr, St. Joseph Railway, Light, Heat & Power Co, 601 Francis, and (for mail) 730 E Hyde Park Ave, St Joseph, Mo

HARTWEIN, Charles E. (*M* 1933) Supervisor, House Htg Dept, St Louis County Gas Co, 231 W Lockwood, Webster Groves, and (for mail) 135 Peeke Ave, Kirkwood, Mo

HARTWELL, Joseph C. (*M* 1922) Pres (for mail) Hartwell Co, Inc, 87 Weybosset St, and 16 Freeman Parkway, Providence, R I

HARVEY, Alexander D. (*A* 1928, *J* 1925) Sales Mgr (for mail) Nash Engineering Co, Wilson Rd, South Norwalk, and West St, New Canaan, Conn

HARVEY, Lyle C (*M* 1928) Pres (for mail) The Bryant Heater Co, 17825 St Clair Ave, Cleveland, and 2666 Leighton Rd, Shaker Heights, O

HASHAGEN, John B. (*M* 1930) Plant Engr, General Seafoods Corp, 1-15 Fish Pier, Boston, Mass

HATEAU, William M (*J* 1934) Draftsman & Designer, J O Ross Engineering Corp, 350 Madison Ave, and (for mail) 1530 Sheridan Ave, New York, N Y

HATHAWAY, Carl B (*A* 1938) Salesman (for mail) Advance Insulating Co, 714 Magee Bldg, and 5204 Woodlawn Place, Pittsburgh, Pa

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HAUCK, Elden L. (*J* 1936) Mgr (for mail) Hauck Bros, 232 S Center St, Springfield, and 1010 S Main St, Dayton, O

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HAUPT, Howard F. (*A* 1938) Htg Salesman, Kohler Co, 751 N Jefferson St, and (for mail) 614 E Beaumont Ave, Milwaukee, Wis

HAUS, Irvin J (*A* 1937, J 1935) Engr, Everett Smith Automatic Temperatures, Inc, 789 N Water St, and (for mail) Jackson Hotel, 926 N. Jackson St, Milwaukee, Wis

HAUSMAN, Louis M. (*M* 1935) Pres, L M. Hausman & Co, 440 Dasmarinas, and (for mail) P O Box 1729, Manila, P I

HAUSS, Charles F.* (*Life Member*) Via Gesu, No. 8, Milan, Italy

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HAWK, Joseph K. (*J* 1936) Engr (for mail) General Air Conditioning Co., 3096 Main St., and 150 Byron, Buffalo, N. Y

HAWKINSON, C. F. (*J* 1936) Mech. Engr., U. S Air Conditioning Corp., 2101 N E. Kennedy, and (for mail) 2833-38th Ave , S , Minneapolis, Minn.
HAYES, James J. (*M* 1920) Sales Engr. (for mail) Stannard Power Equipment Co., Room 925, 53 W. Jackson Blvd., and 7443 Jeffery Ave., Chicago, Ill.
HAYES, Joseph G. (*Life Member*; *M* 1908) Pres - Engr. (for mail) Hayes Brothers, Inc., 236 West Vermont St., and 2849 N. Capital Ave., Indianapolis, Ind.
HAYMAN, A. Eugene, Jr. (*J* 1935; *S* 1930) Draftsman, Moody & Hutchison, Consulting Engineers, 1701 Architects Bldg , Philadelphia, Pa., and (for mail) 2715 Washington St , Wilmington, Del.
HAYNES, Charles V. (*Life Member*; *M* 1917) (*Presidential Member*) (Pres , 1934; 1st Vice-Pres , 1933, 2nd Vice-Pres., 1932; Council, 1926-1929; 1932-1935) Vice-Pres , Hoffman Specialty Co , Inc , 500 Fifth Ave , New York, N Y , and Waterbury, Conn , and (for mail) 115 Llanfair Rd , Ardmore, Mont Co , Pa.
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HAZLETT, Dr. T. Lyle (*M* 1938) Medical Dir (for mail) Westinghouse Electric & Mfg Co , E. Pittsburgh, and 6634 Beacon St , Pittsburgh, Pa.
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HEBERLING, C. W. (*A* 1934) Box 115, Wayzata, Minn
HEBLEY, Henry F. J. (*M* 1934) Advisory Engr , Commerical Testing & Engineering Co , 307 North Michigan Ave , and (for mail) 636 Wrightwood Ave , Chicago, Ill
HECHLER, Samuel (*J* 1937) Engr (for mail) Westchester Square Plumbing Supply Co , Inc , 4617 White Plains Ave , and 3040 Cruger Ave , New York, N. Y
HECHT, Frank H. (*M* 1930) Sales Engr (for mail) B F. Sturtevant Co., 2635 Koppers Bldg , and 1467 Barnesdale St., Pittsburgh, Pa
HECKEL, Edmund P. (*M* 1918) Vice-Pres , Carrier Corp , 222 West North Bank Drive, Chicago, and (for mail) 314 Cuttriss Place, Park Ridge, Ill
HEDDEN, Willard M. (*A* 1937) Treas , The Hedden Co. (for mail) 17-25 South Warren St , and 7 Reservoir Ave., Dover, N J
HEDEEN, Laurel E. (*J* 1938) Sales Engr , The Trane Co (for mail) 818 Hubbell Bldg , and 1709 9th, Des Moines, Ia
HEDGES, H. Berkley (*M* 1919) Mgr , Industrial Sales (for mail) John H. Nesbitt, Inc , Holmesburg, Philadelphia, and 114 Waverly Rd., Wyncote, Pa
HEDLEY, Park S. (*M* 1923) Park S. Hedley Co , 361 Delaware Ave., Buffalo, N. Y.
HEDLUND, Richard A. (*J* 1938; *S* 1937) Sales Engr. (for mail) The Trane Co , 1513 N. Cameron, and 906 N. Third, Harrisburg, Pa
HEEBNER, Walter M. (*M* 1922) Sales Engr , Warren Webster & Co , 20 Washington Place, Newark, and (for mail) 282 Highwood Ave , Teaneck, N J.
HEIBEL, Walter E. (*M* 1917) Dist. Mgr (for mail) Aerofin Corp , 11 West 42nd St., New York, N. Y., and Old Greenwich, Conn
HEILMAN, Russell H.* (*M* 1923) Senior Industrial Fellow (for mail) Mellon Institute, 4400 5th Ave., and 2303 Beechwood Blvd., Pittsburgh, Pa.
HEINKEL, Charles E. (*J* 1938) Sales Engr., Control Equipment Co., 304 Selling Bldg., and (for mail) 1431 S. W. Park Ave., Portland, Ore.
HEISTERKAMP, Herbert W. (*J* 1937) Sales Engr., The Bryant Heater Co , 17825 St. Clair Ave , and (for mail) 10801 St. Mark St , Cleveland, O.
HELBURN, I. B., (*M* 1929; *J* 1927) Jr. Assoc. (for mail) Wyman Engineering, 1306 Chamber of Commerce Bldg., and 3815 Winding Way, Cincinnati, O
HELLER, Joseph A. (*A* 1938) Sales (for mail) Air Conditioning Utilities Co., 8 West 40th St., and 150 West 82nd St., New York, N Y.
HELLMERS, Charles C., Jr. (*J* 1937) Gas Htg. Engr., Iowa-Nebraska Light & Power Co , 1401 O St., and (for mail) 2554 Woodsdale Blvd., Lincoln, Nebr.
HELLSTROM, John (*A* 1929) Vice-Pres (for mail) American Air Filter Co , Inc , 215 Central Ave., and 423 Lightfoot Rd , Louisville, Ky.
HELMRICH, G. Bernard (*M* 1936) Detroit Edison Co., 2000 Second Ave , Room 750 Detroit, and (for mail) 26590 Dundee Rd., Huntington Woods, Royal Oak, Mich.
HELSTROM, Clifford W. (*M* 1938) Mgr , Htg , Plbg. & Air Cond. Dept , Globe Machinery and Supply Co , 205-11 Court Ave., and (for mail) 1614 Thompson Ave , Des Moines, Ia
HELSTROM, Herman G. (*M* 1928) Fire-box Boilers and Stoker Div (for mail) Wm Bros Boiler and Mfg Co , Nicollet Island, and 4608 Arden Ave., S , Minneapolis, Minn.
HENDERSON, Alexander S. (*S* 1938) Jr Engr., Unit Air Conditioners Pty , Ltd , 300 Pitt St , Sydney, and (for mail) 500 Blaxland Rd., Eastwood, N S W , Australia
HENDRICKSON, Harold M. (*M* 1933) Asst. Branch Engr (for mail) York Ice Machinery Corp , 5051 Santa Fe Ave , Los Angeles, and 3901 Liberty Blvd , South Gate, Calif
HENDRICKSON, Ralph L. (*M* 1938) Chief Engr , Utilities Engineering Inst., 404 N Wells St , and (for mail) 6125 Kenwood Ave , Chicago, Ill
HENDRIKSEN, Leonard (*A* 1938) Prop , Hendriksen Sheet Metal & Heating Service, 1919 Vernon Ave , Flint, Mich
HENION, Hudson D. (*A* 1923) Sales Mgr (for mail) C A Dunham Co , Ltd , 1523 Davenport Rd , and 45 Ridge Drive, Toronto, Ont , Canada
HENNESSY, William J. (*M* 1937) Engr , Green Foundry & Furnace Works, Des Moines, Ia , and (for mail) 1826 S. 23rd St , Lincoln, Nebr.
HENRY, Alexander S., Jr. (*M* 1930) 300 Central Park West, New York, N Y
HENRY, Ernest C. (*M* 1938) Member of Firm, The Gray-Henry Co , 614 N. Water St , Bay City, Mich
HENSZEY, William P. (*J* 1935) Chief Engr (for mail) "Carrier-Egypt, S. A E", 37, Shareh Kasr El Nil, Cairo, Egypt, and 320 Hamilton Ave , State College, Pa., U S A
HERBACEK, Edward E. (*M* 1938) Chief Engr. & Secy , Spencer Air Conditioning Co , 515 Essex Bldg , and (for mail) 4624 Upton Ave , S , Minneapolis, Minn
HERBERT, Richard M. (*J* 1938) Jr Engr (for mail) Major Appliance Co , 2558 Farnam St , and 418 North 39th St , Omaha, Nebr.
HERING, Alfred (*M* 1935) Pres , Hering Heating Co , Inc , 304 East 78th St , New York, N Y.
HERKIMER, Herbert (*M* 1934) Dir. (for mail) Herkimer Institute, 1819 Broadway, and 25 Central Park West, New York, N Y.
HERLIHY, Jeremiah J. (*Life Member*; *M* 1914) 3751 Eddy St , Chicago, Ill.
HERMAN, Neil B. (*J* 1937; *S* 1936) Engr (for mail) Riggs Distler & Co , Inc , 713 Maritime Bldg , New Orleans, La , and 4217 Garfield Ave , S , Minneapolis, Minn
HERRING, Edgar (*Life Member*; *M* 1919) Chairman and Governing Dir (for mail) J. Jeffreys & Co., St George's House, 195-203, Waterloo Rd., London, S E. 1 , and "Kenia", Keswick Rd , Putney, London, S. W., England
HERSH, Franklin C. (*A* 1933; *J* 1930) Specialty Engr. (Htg , Vtg. & Air Cond) Pennsylvania Power & Light Co., 901 Hamilton St , and (for mail) 47 S. St. Cloud St , Allentown, Pa.
HERSKE, Arthur R. (*M* 1926) Vice-Pres , Gen. Mgr. Sales (for mail) American Radiator Co., 40 West 40th St , New York, and 101 Brookfield Rd., Mt. Vernon, N. Y.

HERTY, Frank B. (*M* 1933) Industrial Sales Mgr (for mail) The Brooklyn Union Gas Co, 176 Remsen St, Brooklyn, and 106 Pinehurst Ave, New York, N Y.

HERTZLER, John R.* (*M* 1936; *J* 1928) Gen. Repr, Refrig & Air Cond (for mail) York Ice Machinery Corp, and 863 S George St, York, Pa.

HESS, Arthur J. (*M* 1937) Engr, English & Lauer, Inc, 1978 S Los Angeles St, and (for mail) 2616 West 70th St, Los Angeles, Calif

HESS, David K. (*J* 1936; *S* 1932) 5824 Harper Ave, Chicago, Ill

HESSELSCHWERDT, August L., Jr. (*J* 1937) Instructor, Mech Engrg, Wayne University, 4841 Cass Ave, and (for mail) 15722 Kentucky Ave, Detroit, Mich

HESSLER, Lester W. (*M* 1936) Branch Mgr, The Trane Co, 1835 N 3rd St, and (for mail) 6034 N Bayridge, Milwaukee, Wis

HESTER, Thomas J. (*M* 1919) Vice-Pres-Treas. (for mail) Hester Bradley Co, 2835 Washington Ave, and 67 Aberdeen Place, St Louis, Mo

HEWETT, John B. (*M* 1937, *A* 1935) Anemostat Corp of America, 10 East 39th St, New York, and (for mail) Sussex Hall, Dobbs Ferry, N Y

HEXAMER, Harry D. (*M* 1931) 163 E Delavan Ave, Buffalo, N Y

HEYDON, Charles G. (*A* 1923) Mgr Sales of Western Div, Wright Austin Co, 315 West Woodbridge St, and (for mail) 2681 Nebraska, Detroit, Mich

HIBBS, Frank C. (*M* 1917) Htg Engr, The H B Smith Co, Inc, 2209 Chestnut St, and (for mail) 846 North 65th St, Philadelphia, Pa

HICKEY, Daniel W. (*A* 1931) Pres, D W Hickey & Co, Inc (for mail) 1631 University Ave, and 1874 Highland Pkwy, St Paul, Minn

HICKMAN, Herbert V. (*A* 1938) Sales Engr, Birchfield Boiler, Inc, 1129 Folsom St, San Francisco, Calif

HIERS, Charles R. (*M* 1929, *J* 1927) Sales Engr, Minneapolis-Honeywell Regulator Co, 801 Second Ave, New York, and (for mail) 19 Westminster Rd, Great Neck, L I, N Y

HIGDON, Harry S. (*A* 1937) Sales, Andrews Heater Co, 2231 Market, and (for mail) 231 Byxbee, San Francisco, Calif

HIGH, John M. (*A* 1938) Mgr, The Ruberoid Co, 500 Fifth Ave, New York, N Y

HILDER, Frederick L. (*M* 1937) Chief Engr, Electric Furnace-Man, Inc, 780 E 138th St, New York, N Y, and (for mail) 162 Trenton Ave, Clifton, N J

HILDRETH, Egbert S. (*A* 1936) Air Cond Promotion (for mail) Indianapolis Power & Light Co, 17 N Meridian St, and 5626 E Michigan St, Indianapolis, Ind

HILDRETH, Lane W. (*M* 1935) Secy (for mail) Anthracite Institute, 19 Rector St, New York, N Y, and 243 W Tulpehocken St, Philadelphia, Pa

HILL, Charles F. (*J* 1936) Carrier Air Cond Dept Mgr, United Engineers, Ltd, River Valley Rd, Singapore, Straits Settlements

HILL, Dr. E. Vernon* (*M* 1914, *A* 1912) (*Presidential Member*) (Pres, 1920, 1st Vice-Pres, 1919, 2nd Vice-Pres, 1918, Council, 1915-1921) Owner (for mail) 179 W. Washington St, and 6826 Newell Ave, Chicago, Ill

HILL, Fred M. (*M* 1930) 225 East Avenue 39, Los Angeles, Calif

HILL, Harold G. (*J* 1938) Sales Engr, Gurney Foundry Co, Ltd., Junction Rd, and (for mail) Apt 5, 29 Jane St, Toronto, Ont, Canada

HILL, Harold H. (*M* 1935) Branch Mgr (for mail) American Blower Corp, 1211 Commercial Bank Bldg, and 1705 East Blvd, Charlotte, N C

HILL, Jared A. (*M* 1937) Htg & Air Cond Engr, Pacific Gas & Electric Co, 245 Market St, San Francisco, and (for mail) 717 Laurel Ave, Burlingame, Calif

HILL, Vaughn H. (*J* 1938) Engr, Nash-Kelvinator Corp, Automatic Dept, Plymouth Rd., Detroit, and (for mail) 205 Lathrop St., Lansing, Mich

HILLIARD, Charles E. (*M* 1932; *J* 1927) Htg.-Vtg Engr. (for mail) E C Hilliard Corp., 27 B St, South Boston, and No. 6 Arcade, Framingham, and 341 Hunnewell St, Needham Heights, Mass.

HILLS, Arthur H. (*M* 1924) Mgr (for mail) Sarco Canada, Ltd, 85 Richmond St, W., and 100 Nealon Ave, Toronto, Ont, Canada.

HINCKLEY, Harlan B. (*A* 1934) Engr-Custodian, Chicago Board of Education, 8510 S. Green St, and (for mail) 6933 Princeton Ave, Chicago, Ill

HINES, Guy M. (*A* 1937) Chief Engr., Texas Agricultural & Mechanical College Power Dept., College Station, Tex.

HINES, John C. (*M* 1937) Vice-Pres-Treas (for mail) R B Hayward Co, 1714 Sheffield Ave, Chicago, and 6629 Ramona Ave, Lincolnwood, Ill

HINKLE, Edwin C. (*Life Member*; *M* 1911) 170 N. Franklin St, Hempstead, L I, N Y

HINNANT, Clarence H., Jr. (*J* 1938) Chief Engr, Hungerford Coal Co, 717 E Grace St, and (for mail) R F D No 9, Box 156, Richmond, Va

HINRICHSEN, Arthur F. (*M* 1928) Pres & Treas (for mail) A F. Hinrichsen, inc, 50 Church St, New York, N Y, and Mountain Lakes, N J

HINTON, Robert P. (*A* 1938) Sales Engr (for mail) Crane, Ltd, 95 Lombard St, and 112 Balmoral St, Winnipeg, Man, Canada

HIRSCH, Martin H. (*M* 1938) Sales Engr, Hoffman Specialty Co, Inc, 500 Fifth Ave, and (for mail) 1478 Walton Ave, New York, N Y

HIRSCHMAN, William F. (*M* 1929) Pres and Chief Engr, W F Hirschman Co, Inc, 220 Delaware Ave, and (for mail) 165 Le Brun Circle, Buffalo, N Y

HITCHCOCK, Paul C. (*M* 1931) Vice-Pres (for mail) Burlingame, Hitchcock & Estabrook, Inc, 521 Sexton Bldg, and 5130 Harriet Ave, Minneapolis, Minn

HITT, John C. (*A* 1936) Branch Mgr (for mail) Holland Furnace Co, 34-17th St, and 301 Valley View Ave, Wheeling, W Va

HOBBIE, Edward H (*A* 1937) Mgr Sales Promotion and Research (for mail) Mississippi Glass Co, 220 Fifth Ave, New York, N. Y, and Ridgedale Ave, Florham Park, N J.

HOBBS, J. Clarence (*M* 1920) Vice-Pres, Diamond Alkali Co, Painesville, O

HOBBS, William S. (*A* 1936) (for mail) Wm S Hobbs, P O Box 269, and 327 Park Ave, Swarthmore, Pa

HOCHMAN, Eugene (*S* 1938) 286 Chestnut St, Chelsea, Mass

HOCKENSMITH, Francis E. (*M* 1936) Chief Engr (for mail) Lennox Furnace Co, Inc, 400 N Midler Ave, and 124 Ludington St, Syracuse, N Y

HODGE, William B. (*M* 1934) Vice-Pres & Engr (for mail) Parks-Cramer Co, P O Box 1234, and 301 Hawthorne Lane, Charlotte, N C

HOEHL, Edward R (*J* 1935) Sales Engr (for mail) Long Island Air Conditioning Co, Inc, 541 Franklin Ave, and 821 Franklin Ave, Garden City, L I, N Y.

HOEY, James K. (*A* 1938) Pres-Mgr (Registered Mech. Engr) (for mail) Crater Metal & Engineering, Inc, 142 N Front St, and 119 Lincoln St, Medford, Ore

HOFFBERGER, John P. (*A* 1938) Merchandising, Electrical Corp, 6400 Plymouth, and (for mail) 5838 Enright, St Louis, Mo

HOFFMAN, Angelo (*A* 1938) Vice-Pres (for mail) Louis Hoffman Co, 117 W Pittsburgh Ave, and 4850 N Oakland Ave., Milwaukee, Wis

HOFFMAN, Charles S. (*M* 1924) Pres (for mail) Baker Smith & Co, Inc, 576 Greenwich St, and 108 East 38th St, New York, N Y

HOGAN, Edward L.* (*M* 1911) General Consulting Engr (for mail) American Blower Corp, 6000 Russell St, and 700 Seward Ave, Detroit, Mich

HOGUE, William M. (*A* 1935) Sales Engr. (for mail) U. S. Electrical Motors, Inc., 200 E. Slauson Ave, and 4839 Keniston Ave., Los Angeles, Calif.

HOLLAND, George R. (*S* 1938) Asst. Chemist, American Sugar Refining Co, 49 S 2nd St., and (for mail) 19 East 98th St, New York, N. Y.
HOLLAND, Robert B. (*M* 1937) Sales Engr. (for mail) York Ice Machinery Corp, 1275 Folsom St, and 3820 Scott St, San Francisco, Calif.
HOLLISTER, E. Wallace (*M* 1936; *J* 1931) Owner (for mail) Hollister's, 31 Ridge St., and 115 Grant Ave, Glens Falls, N. Y.
HOLLISTER, Norman A.* (*M* 1933) 7101 Colonial Rd, Brooklyn, N. Y.
HOLMES, Arthur D. (*M* 1935) Vice-Pres (for mail) Plumbers Supply Co, 323 W. First, and 1848 East 18th St, Tulsa, Okla
HOLMES, Paul B. (*A* 1936) Branch Mgr. (for mail) National Radiator Corp., 600 W. St, N. E., and 4525 Fessendon St., N W., Washington, D. C.
HOLMES, Richard E. (*A* 1938; *J* 1934) Air Cond. Design Engr, Westinghouse Elec & Mfg. Co., E. Springfield, and (for mail) 258 Redlands St., Springfield, Mass
HOLT, James (*M* 1933) Assoc Prof. of Mech Engrg (for mail) Massachusetts Institute of Technology, Cambridge, and 1062 Massachusetts Ave, Lexington, Mass
HOLT, Walter H. (*J* 1938) Engr. (for mail) Buffalo Forge Co, 490 Broadway, and 149 Highland Ave, Buffalo, N. Y
HOLUBA, Henry J. (*J* 1938) Sales Engr (for mail) A-J Manufacturing Co, 2119 Washington St, and 501 W 31st, Kansas City, Mo
HOLYFIELD, Earl F. (*A* 1937) Air Cond Engr, Oklahoma Electrical Supply, and (for mail) 121 E Park Ave, Oklahoma City, Okla.
HOMAN, John D. (*M* 1938) Chief Elec Engr, Amalgamated Phosphate Co., P. O. Box 172, Brewster, Fla.
HONERKAMP, Fritz (*M* 1937) Chief Engr. (for mail) Anemostat Corp of America, 10 East 39th St, New York, and 67-12 50th Ave, Woodside, L I, N. Y
HOOK, Frank W. (*M* 1937) Branch Mgr (for mail) Johnson Service Co, 814 Rialto Bldg, 2363 Larkin St., San Francisco, Calif.
HOPPE, Albert A. (*M* 1935) Design and Application Engr, Carrier Corp, 213 West 1st St, and (for mail) 1941 N. W. 17th St., Oklahoma City, Okla
HOPPE, Marcel F. (*M* 1938) Consulting Engr. (for mail) 1621 Connecticut Ave, N. W., Washington, D C, and P O B 531, Falls Church, Va
HOPPER, Garnet H. (*M* 1923) Engr., Taylor-Forbes, Ltd, 1088 King St., W, and (for mail) 19 Brummell Ave, Toronto, Ont, Canada.
HOPPER, John S. (*M* 1938) Instructor, Texas A. & M. College, Mech. Engrg. Dept, College Station, Tex
HOPSON, William T. (*Life Member*; *M* 1915) The Hopson & Chapin Mfg Co, New London, Conn
HORNER, Samuel D. (*J* 1937) c/o Irrawaddy Flotilla Co, Ltd, Sales Dept., 510 Merchant St, Rangoon, Burma, India.
HORNUNG, J. C. (*M* 1914) 854 Bluff St, Glencoe, Ill
HOSHALL, Robert Houston (*M* 1930) Associate (for mail) Thos H. Allen, Consulting Engr., 65 McCall Place, and 1844 Cowden Ave, Memphis, Tenn.
HOSKING, Homer L. (*M* 1930) Sales Mgr, Pacific Steel Boiler Corp, 101 Park Ave, New York, and 208 Madison Rd, Scarsdale, N Y.
HOSTERMAN, Charles O. (*M* 1924) Supt., The McMurrer Co., 303 Congress St., Boston, and (for mail) 25 Bateswell Rd., Dorchester, Mass.
HOTCHKISS, Charles H. B. (*M* 1927) Editor, Heating and Ventilating, 148 Lafayette St., New York, N. Y.
HOTOP, Herbert C. (*A* 1938) Vice-Pres and Mgr., Fred J. Hotop & Co., 315 N. Church St., Kalamazoo, Mich.
HOUGHTEN, Ferry C.* (*M* 1921) Dir. (for mail) Research Lab., A S H.V.E., 4800 Forbes St., and 1136 Murrayhill Ave, Pittsburgh, Pa.
HOULIS, Louis D. (*M* 1935) Chief Engr., Master Baker Ovens, and (for mail) 655 Pedretti Rd., West Price Hill, Cincinnati, O.
HOULISTON, G. Baillie (*A* 1928) Secy. (for mail) The W C. Green Co., 704 Race St, Cincinnati, O., and 33 Tremont Ave, Fort Thomas, Ky.
HOUSKA, Arthur D. (*J* 1937) Sales Engr (for mail) Clowe & Cowan, Inc., and 1417 Harrison St., Amarillo, Tex.
HOWARD, Fenton L. (*M* 1937) Chief Engr. (for mail) Refrigeration & Air Conditioning Inst., 2150 Lawrence Ave., and 6619 N. Rockwell St., Chicago, Ill.
HOWATT, John* (*M* 1915) (*Presidential Member*) (Pres., 1935; 1st Vice-Pres, 1934; 2nd Vice-Pres, 1933; Council, 1927-1936) Chief Engr (for mail) Board of Education, 228 N. LaSalle St., and 4940 East End Ave, Chicago, Ill.
HOWE, Willis W. (*M* 1936; *A* 1917) Sales Engr., Pacific Gas & Electric Co., and (for mail) 68 Central Ave, Sausalito, Calif.
HOWELL, Lloyd (*M* 1915) Engr., Industrial Dept (for mail) Peoples Gas Light and Coke Co, 122 S. Michigan Ave., and 7605 Yates Ave, Chicago, Ill.
HOWLETT, Ira G. (*M* 1935; *S* 1934) Consulting Engr. (for mail) I G Howlett Co, 120 E Main St., and 2123 N. Fonshill Ave., Oklahoma City, Okla.
HOYT, Charles W. (*A* 1931) Pres-Treas. (for mail) Wolverine Equipment Co, 31 Main St, Cambridge, and 45 Thaxter Rd., Newtonville, Mass
HOYT, Leroy W. (*M* 1930) N. Stamford Ave., Stamford, Conn
HUBBARD, George W.* (*M* 1911) Consulting Engr. (for mail) 1406 Railway Exchange, Chicago, and 710 Bonnie Brae, River Forest, Ill
HUBBARD, Nelson B. (*M* 1919) Partner, Hubbard & Wagschal, Engineers, 243 Congress St., W, and (for mail) 2985 Blaine Ave., Detroit, Mich
HUBBUCH, Nicholas J., Jr. (*S* 1939) Student (for mail) University of Dayton, Dayton, O, and 510 Breckenridge Lane, Louisville, Ky.
HUBER, Enrique (*M* 1938) Mech Engr, Bucaroli No. 128, P O Box 10633, Mexico, D. F
HUCH, A. J. (*M* 1919) Secy & Treas. (for mail) Central Supply Co, 312 S 3rd St, and 4037 Harriet Ave, Minneapolis, Minn
HUCKER, Joseph H. (*M* 1921) Partner, Hucker-Pryibil Co., 1700 Walnut St, Philadelphia, and (for mail) 715 Stanbridge St., Norristown, Pa.
HUDEPOHL, Louis F. (*M* 1936) Pres (for mail) T J Conner, Inc., 3290 Spring Grove Ave, and 4395 Haight Ave, Cincinnati, O
HUDSON, Robert A. (*M* 1934) Partner (for mail) Hunter & Hudson, Consulting Engrs, 41 Sutter St., Room 710, San Francisco, and Route 2, Box 51, Cordilleras Rd, Redwood City, Calif
HUFF, James M. (*M* 1936) Air Cond Engr. (for mail) Kribs & Landauer, Inc, 404 Dallas Gas Bldg, and 839 N. Bishop, Apt 4, Dallas, Tex.
HUGHES, Harold R. (*A* 1938) Mgr, Fuel Oil Div (for mail) McColl Frontenac Oil Co, Ltd., 1010 St Catherine St, W, and 4396 Mayfair Ave, Montreal, P. Q, Canada.
HUGHES, L. K. (*J* 1936) Vice-Pres. in Charge of Sales, Howard Air Conditioning, Ltd, 881 Yonge St., and (for mail) 43 Rivercourt Blvd, Toronto, Ont, Canada.
HUGHES, William U. (*M* 1936) Pres. (for mail) The Lewis-Brown Co, Ltd, 1411 Crescent St., Room 206, and 1610 Sherbrooke St., West, Apt. 36, Montreal, P. Q., Canada.
HUGHEY, Thomas M. (*A* 1935) Sales Engr. (for mail) Westerlin & Campbell Co, 906 North 4th St., and 2350 North 58th St, Milwaukee, Wis.
HUGHSON, Harry H. (*M* 1927) Sales Engr. (for mail) The Coon-DeVisser Co, 2051 W. Lafayette St., Detroit, and 58 Florence Ave, Highland Park, Mich.
HUGONIOT, Victor E. (*M* 1935) Zone Engrg. Mgr. (for mail) Airtemp Sales Corp., 3717 Washington Ave., and 1347 Kingsland Ave., St. Louis, Mo.
HULL, Harry B. (*M* 1931) Research Engr. (for mail) Frigidaire Div. of General Motors Corp., and 1454 Glendale Ave., Dayton, O.

HUMMEL, George W. (*M* 1937) Field Engr (for mail) The Trane Co, Drawer 679, and 327 E. McDowell Rd, Phoenix, Ariz
HUMPHREY, Dwight E. (*M* 1921) Htg. & Vtg Engr, Goodyear Tire & Rubber Co, Akron, and (for mail) 2499 Six St, Cuyahoga Falls, O
HUMPHREY, Leonard G., Jr. (*J* 1938) Engr (for mail) Buffalo Forge Co, 490 Broadway, and 261 Crestwood Ave, Buffalo, N Y
HUMPHREYS, Clark M. (*M* 1931) (Council, 1938) Asst Prof of Mech Engrg (for mail) Carnegie Institute of Technology, Schenley Park, and 1934 Remington Dr, Pittsburgh, Pa
HUNGER, Robert F. (*M* 1927) Associate Dist Mgr (for mail) Davidson & Hunger, 220 South 16th St, Philadelphia, and 100 Avon Rd, Narberth, Pa
HUNGERFORD, Leo (*M* 1930) Sales Mgr, Utility Fan Corp, 2528 Santa Fe Ave, and (for mail) 900 N Stanley Ave, Los Angeles, Calif
HUNT, MacDonald (*A* 1936) Mfrs Agent (for mail) McDonnell Miller Co, 12 W Madison St, and Windsor Court Apts, Baltimore, Md
HUNT, Noel P. (*M* 1934) Managing Director (for mail) Carrier Australasia, Ltd, 41-49 Forbes St, and 52 Lang Rd, Centennial Park, Sydney, N S W, Australia.
HUNTER, Louis N. (*M* 1936) Mgr of Research (for mail) National Radiator Corp, 221 Central Ave, and 839 Luzerne St, Johnstown, Pa
HUNZIKER, Chester E. (*A* 1934) Branch Mgr (for mail) American Blower Corp, 331 State St, Schenectady, and 422 Reynolds St, Scotia, N Y
HUSKY, S. T. (*J* 1936, *S* 1934) Engr, Zenith Gas System, Box 397, Alva, Okla
HUST, Carl E. (*M* 1932) Supervisor Heating Engr (for mail) Cincinnati Gas & Electric Co, 4th & Main Sts, and 15 Mason St, Cincinnati, O
HUSTOEL, Arnold M. (*A* 1930) 2414 N Kedzie Blvd, Chicago, Ill
HUTCHEON, Clifford R. (*J* 1938) Engr (for mail) Anemostat Corp of America, 10 E 39th St, New York, and 3131-41st St, Astoria, L I, N Y
HUTCHINGS, Robert L. (*J* 1937) Hughes Heating & Air Conditioning Co, 125 N Jefferson St, Dayton, O
HUTCHINS, William H. (*M* 1934) Chief Engr, Delco Appliance Div, General Motors Corp, and (for mail) 88 Magee St, Rochester, N Y
HUTCHINSON, Frank W. (*A* 1937) 114 Engineering Bldg, University of California, Berkeley, Calif
HUTZEL, Hugo F. (*M* 1918) Air Cond Applications Dept, Kelvinator Corp, and (for mail) 2635 Woodstock Dr, Detroit, Mich
HUYBERT, Leslie E., Jr. (*J* 1938) Draftsman, U. S Naval Operating Base, and (for mail) 604 Carolina Ave, Norfolk, Va
HVOSLEF, Frederick W. (*M* 1931, *A* 1921) Heating Research Engr (for mail) Kohler Co, and 523 Audubon Rd, Kohler, Wis
HYDE, Elmer H. (*A* 1937) Tech Repr, Koppers Co., Tar & Chemical Div, 501 Flannery Bldg, and (for mail) 442 Sulgrave Rd, Pittsburgh (11) Pa
HYDE, Eric F. (*M* 1937) Consulting Engr, 512 Free Press Bldg, Detroit, and (for mail) 708 Oakland Ave, Birmingham, Mich
HYMAN, Wallace M. (*M* 1920) Pres (for mail) Reis & O'Donovan, Inc, 12 W 21st St, and 23 W 73rd St, New York, N Y
HYNES, Lee P. (*M* 1919) Pres and Chief Engr (for mail) Hynes Electric Heating Co, 240 Cherry St, Philadelphia, Pa, and 127 West End Ave, Haddonfield, N J

I

IBISON, James L. (*J* 1938) Sales Engr, Ralph B Johnson & Co, 201 Petroleum Bldg, Houston, Tex
ICKERINGILL, John C. (*M* 1923) Sales Engr, Spencer Heater Co, 2020 N Broad St, and (for mail) 477 Flamingo St, Roxborough, Philadelphia, Pa
ILLIG, Ernest E. (*J* 1938) Sales Engr, Walter R Illig, 1 Cushing St, and (for mail) 242 Blossom St, Fitchburg, Mass
ILLIG, Walter R. (*M* 1935; *A* 1927) Owner, Heating, Plumbing, Air Conditioning, 1 Cushing St, and (for mail) 242 Blossom St, Fitchburg, Mass
INGALLS, Frederick D. B. (*M* 1906) Consulting Htg and Air Cond Engr, 1 Hopkins St, Reading, Mass
INGELS, Margaret* (*M* 1923, *J* 1918) Mech Engr (for mail) Carrier Corp, and 412 University Place, Syracuse, N. Y
IRWIN, Robert R. (*J* 1937) Air Cond Engr (for mail) York Ice Machinery Corp, Mayo Building, and 811 North Rockford St, Tulsa, Okla
ISETT, William M. (*A* 1936) Pres (for mail) C B Isett & Son, Inc, 3035 N Rockwell St, and 4236 N Drake Ave, Chicago, Ill
d'ISSERTELLE, Henry G.* (*Life Member*, *M* 1913, *A* 1912) Consulting Engineer, 31 Park Terrace W, Apt A-8, New York, N Y
IVERSON, Henry R (*M* 1936) Co-Mgr (for mail) The Trane Co, 1772 Columbia Rd, N W, and 1601 Argonne Place, N W, Washington, D C.

J

JACKES, Herman D. (*M* 1915) Consulting Engr, 1 Clinton Rd, Glen Ridge, N J
JACKSON, Charles H. (*M* 1923) Vice-Pres (for mail) Blower Application Co, 918 N Fourth St, and 2706 N Farwell Ave, Milwaukee, Wis
JACKSON, Gilbert R. (*M* 1938) Mgr, Boiler & Radiator Dept (for mail) Crane Ltd, 45-51 Leman St, London, E 1, and 174 Chiltern Court, Baker St, London, N W 1, England
JACKSON, Marshall S. (*M* 1919) Repr (for mail) Powers Regulator Co, 250 Delaware Ave, and 108 Larchmont Rd, Buffalo, N Y
JACOBSEN, K C. S. (*A* 1939) Sales Repr, Imperial Electric Co, and (for mail) 309 South 22nd St, Philadelphia, Pa
JACOBUS, Dr. David S. (*Life Member*, *M* 1916) Advisory Engr (for mail) The Babcock & Wilcox Co, 85 Liberty St, New York, N Y, and 93 Harrison Ave, Montclair, N J
JAKOBY, Albert C. (*A* 1938) Estimator & Designer, Sears Roebuck & Co, Dept. 405, 4640 Roosevelt Blvd, and (for mail) 1913 E Clearfield St, Philadelphia, Pa
JALONACK, Irwin G. (*A* 1933, *S* 1930) Chief Engr (for mail) Alfred L. Hart, Inc, 315 Vanderbilt Ave, Brooklyn, and 62-30 Saunders St, Rego Park, N Y
JAMES, Hamilton R. (*M* 1931) Service Equip. Engr, United Engineers & Constructors, Inc, 1401 Arch St, Philadelphia, and (for mail) 55 W Drexel Ave, Lansdowne, Pa
JAMES, John W.* (*M* 1937, *J* 1933) Tech Secy., American Society of Heating & Ventilating Engineers, 51 Madison Ave, New York, N Y
JAMES, Richard E. (*M* 1936) Mgr, Htg Dept, Harry Cooper Supply Co, and (for mail) 597 E Elm St, Springfield, Mo
JANET, Harry L. (*M* 1920) Mech Engr, Buensod-Stacey Air Cond, 60 East 42nd St, New York, and (for mail) 688 Decatur St, Brooklyn, N Y
JARCHO, Martin D. (*J* 1936) Vice-Pres. (for mail) Jarcho Bros, 215 East 37th St, New York, and 941 Washington Ave, Brooklyn, N Y
JARDINE, Douglas C. (*M* 1929, *A* 1926) Pres (for mail) Jardine & Knight Plumbing & Heating Co, 516 S Tejon St, and 1512 E Platte Ave, Colorado Springs, Colo
JARDINE, William H., Jr. (*A* 1938) Pres (for mail) Iona Ventilator Co, Inc, 2821-29 W Dauphin St, and 3552 Shelmire St, Philadelphia, Pa
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JELINEK, Frank R. (*J* 1937) Sales Engr (for mail) Johnson Service Co, 2505 Commerce St, Dallas, and 1214 Banks St, Houston, Tex

JENKINS, Frank H. (*S* 1938) Student Engr (for mail) Buffalo Forge Co , 490 Broadway, and 149 Highland St , Buffalo, N Y

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JENNINGS, Irving C. (*M* 1924) Pres (for mail) Nash Engineering Co , and 138 Flax Hill Rd., South Norwalk, Conn

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JOHNSON, Allen J.* (*M* 1935) Dir Anthracite Industries Laboratory, Primos, Delaware Co , Pa.

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JONES, Charles R. (*A* 1928) Pres , Jones Supply Co , Siloam Springs, Ark.

JONES, David J. (*M* 1936) Control Engr , Vapor Car Heating Co , Inc , Railway Exchange Bldg , Chicago, and (for mail) 391 Poplar Ave , Elmhurst, Ill.

JONES, Edwin (*M* 1933; *J* 1924) Engr and Estimator (for mail) Watt Plumbing, Heating & Supply Co., 608 S Cincinnati, and 1436 East 17th Place, Tulsa, Okla

JONES, Edwin A. (*M* 1919) Chief Engr (for mail) L J Mueller Furnace Co , 2005 W. Oklahoma Ave , and 4381 N. Alpine Ave , Milwaukee, Wis

JONES, Edwin F. (*M* 1923) Utilities Engr , City of St. Paul, 216 Courthouse, and (for mail) 220 Montrose Place, St Paul, Minn.

JONES, Harold L. (*M* 1920) Supt (for mail) W W. Farrier Co , 44 Montgomery St., Jersey City, and 11 Cambridge Rd , Glen Ridge, N J

JONES, Hubert L. (*A* 1938) Zone Engr , Delco-Frigidaire Cond Div , General Motors Sales Corp , and (for mail) 519 W Norman Ave , Dayton, O.

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JONES, Sprague (*M* 1936) Pres (for mail) Sprague Jones, Inc , 1116 Madison Ave., and 3769 S Lockwood Ave , Toledo, O

JONES, William T. (*M* 1915) (*Presidential Member*) (Pres., 1933, 1st Vice-Pres , 1932; 2nd Vice-Pres , 1931; Council, 1925-1933) Treas., Barnes & Jones, 128 Brookside Ave , Jamaica Plain, and (for mail) 16 Harvard St , Newtonville, Mass

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KARAKASH, Theodore J. (*J* 1936) Head Engr (for mail) Air Cond Branch, G & A Baker Co , P. O. Box 468, and Engin Apt , Feruzaga, Istanbul, Turkey
KARCHMER, Jacob H. (*A* 1936) Mgr (for mail) Karchmer Co , 600-14 N Jefferson Ave , and 1316 Roanoke Ave , Springfield, Mo
KARGES, Albert (*A* 1935) Mgr , The James Stewart Mfg Co , Ltd , and (for mail) 37 Perry St , Woodstock, Ont , Canada
KARLSON, Alfred F. (*M* 1918) Chief Engr (for mail) Parks-Cramer Co , 970 Main St , Fitchburg, and 186 Prospect St , North Leominster, Mass
KARLSTEEN, Gustav H. (*M* 1935) Plant Engr , Dunlop Tire & Rubber Corp , Buffalo, and (for mail) Box 55, Route 1, Tonawanda, N Y.
KARTORIE, V. T. (*J* 1935; *S* 1933) Sales Engr (for mail) York Ice Machinery Corp , and 1513 Third Ave , York, Pa
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KELLY, Wilbur C. (*M* 1935) Field Engr (for mail) Iron Fireman Mfg Co of Canada Ltd , 602 King St , W , and 58 Elmsthorpe Ave , Toronto, Ont , Canada
KENDALL, Edwin H. (*M* 1930) Sales Engr , English & Lauer, Inc , 1978 S Los Angeles St , Los Angeles, Calif
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KENT, Laurence F. (*A* 1927; *J* 1924) Pres (for mail) Moncrief Furnace Co , P O Box 1673, and 1515 Morningside Drive, N E , Atlanta, Ga
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KERN, Joseph F., Jr. (*A* 1937) Asst Editor, Heating & Ventilating, 148 Lafayette St , New York, and (for mail) 88-24-166th St , Jamaica, L. I , N Y
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KERSHAW, Melville G. (*M* 1932; *A* 1926; *J* 1921) Vtg and Air Cond Engr (for mail) E I DuPont de Nemours & Co , Wilmington, Del , and 7313 North 21st St , Philadelphia, Pa
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KESSLER, Maurice E. (*M* 1937) Mgr , Pioneer Heating-Cooling Co , 901 Niagara St , and (for mail) Falls Station, P O Box 664, Niagara Falls, N Y
KETTER, Jack W. (*J* 1937) Design Engr , Krenz & Co , 5114 W Center St , and (for mail) 3042 N 2nd St , Milwaukee, Wis.
KEYES, Robert E. (*M* 1913) Chief Engr , Cooling & Air Cond Div , B F Sturtevant Co , Hyde Park, Boston, Mass

KEYSER, Herman M. (*A* 1937) Sales Engr, Murray W Sales & Co, Detroit, and (for mail) 10703 Hart Huntington Woods, Royal Oak, Mich

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KIDD, Charles R. (*A* 1938) Mgr, Commercial Kelvinator Dept, and (for mail) 611 N W 28th St, Oklahoma City, Okla

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KIEFER, E. J., Jr. (*A* 1932, *J* 1928) Mgr, H C Archibald Co, 406 Main St, and (for mail) 108 N. 6th St, Stroudsburg, Pa

KIESLING, Justin A. (*M* 1930) Pres (for mail) Robischung-Kiesling Contracting Corp, 4848 Main St., and 1602 Stuart St, Houston, Tex

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KILLIAN, Thomas J. (*A* 1937) Htg Contractor, 118 Belvidere St, Waukegan, Ill

KILLIAN, Vic. J. (*A* 1937) Pres (for mail) V J Killian Co, 907 Linden Ave, and 1348 Edgewood Lane, Winnetka, Ill

KILLOUGH, Robert E (*A* 1938) Engr, Standard Oil Co of Pennsylvania, 1618 N Broad St, and (for mail) 2108 E Chelten Ave, Germantown, Philadelphia, Pa

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KIMBLE, Carl W. (*J* 1938) Mgr Htg Dept, A Y McDonald Mfg Co, 929 S W. 9th St, and (for mail) 3125 S W 13th St Place, Des Moines, Ia

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KING, Lon D. (*A* 1937) Air Cond and Htg (for mail) Sidles Co, Airtemp Div, 425 Stuart Bldg, and 2325 R St, Lincoln, Nebr

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LANDES, Bates E. (*M* 1938) Mech. Engr. (for mail) 915 Hubbell Bldg, and 1603-47th St., Des Moines, Ia.

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LANGENBERG, Everett B. (*M* 1914) Owner (for mail) Langenberg Heating Co., 3800 West Pine Blvd, St. Louis, and 223 E. Adams St., Kirkwood, Mo

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LARSON, Carl W. (*M* 1936) Sales Engr, Barnes & Jones, Inc, Rm 901, Industrial Trust Bldg., Providence, R. I, and (for mail) 641 Hyde Park Ave, Roslindale, Mass.

LARSON, Clifford P. (*J* 1936) (for mail) The Insulite Co., 205 W. Wacker Drive, and 30 W. Chicago, Ave., Chicago, Ill

LARSON, Gustus L.* (*M* 1923) (*Presidential Member*) (Pres, 1936; 1st Vice-Pres, 1935; 2nd Vice-Pres, 1934; Council, 1929-1937) Prof., Steam and Gas Engrg, and Chairman of Dept of Mech. Engrg. (for mail) University of Wisconsin, Mech. Engrg. Bldg., and 1213 Sweetbriar Rd., Shorewood Hills, Madison, Wis.

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LEEK, Charles W. (*M* 1938) Managing Director, Leek & Co, Ltd, 1111 Homer St, and (for mail) 4682 West 6th Ave, Vancouver, B C, Canada

LEEK, Walter (*Life Member*; *M* 1903) Pres (for mail) Leek & Co, Ltd, 1111 Homer St, and 4769 W Second Ave., Vancouver, B C, Canada

LEFEBVRE, Eugene J. (*M* 1937) Engr, Warden King, Ltd., 2 Bennett Ave, Montreal, and (for mail) 38 Third St, St Lambert, P. Q, Canada.

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LEHMANN, Matt (*J* 1937) Engr., Mech & Elec. Consulting, 603 Architects Bldg, and (for mail) 1569 Midvale Ave., Westwood, Los Angeles, Calif.

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LEIGH, Robert L. (*A* 1938) Engr, Hart & Cooley Mfg. Co., and (for mail) 78 E. 12th St., Holland, Mich

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LEILICH, Roger L. (*M* 1922) Pres. (for mail) Baltimore Heat Corp., 2000 W. Pratt St., and 2810 Elsinor Ave, Baltimore, Md

LEINROTH, J. Paul (*M* 1929) Gen Industrial Fuel Repr. (for mail) Public Service Electric & Gas Co., 80 Park Place, Newark, and 37 The Fairway, Montclair, N. J

LEITCH, Arthur S. (*M* 1908) Pres and Managing Dir. (for mail) The Arthur S Leitch Co, Ltd, 1123 Bay St, and 421 Russell Hill Rd, Toronto, Ont, Canada.

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LELAND, William E. (*M* 1915) Consulting Engr (for mail) Leland & Haley, 58 Sutter St, San Francisco, and 704 The Alameda, Berkeley, Calif.

LENIHAN, William O. (*A* 1936) Vice-Pres (for mail) Laverack & Haines, Inc., 718 White Bldg, and 703 W. Ferry St, Buffalo, N Y.

LENONE, Jose M. (*M* 1919) Designing Engr (for mail) Wilson & Co, Inc, 4100 S Ashland Ave, and 1358 East 48th St, Chicago, Ill

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LEUPOLD, George L. (*A* 1937) Sales Engr, Minneapolis-Honeywell Regulator Co, 561 Reading Rd, and (for mail) 1715 Stonybrook Drive, Cincinnati, O

LEUTHESSER, Fred W., Jr. (*M* 1937) Secy (for mail) National Metal Products Corp, 21 N Loomis St, Chicago, and 1640 Wesley Ave, Berwyn, Ill

LEVENTHAL, Bernard (*J* 1937, *S* 1935) 3913-13th Ave, Brooklyn, N Y.

LEVY, Marion I. (*M* 1938; *A* 1936; *J* 1931) Pres, Viking Air Cond Corp, Main & Center Sts, and (for mail) 3156 Ludlow Rd, Cleveland, O

LEWIS, Carroll E. (*M* 1930) Sales Mgr, Delco-Frigidaire Conditioning Div, General Motors Sales Corp, 300 Taylor St, and (for mail) 2724 Fairmont, Dayton, O

LEWIS, Clyde A. (*J* 1937) Htg-Vtg & Air Cond Engr, A Edward Johnson, Consulting Engr, 132 East 58th St, and (for mail) 23-27-28th St, Astoria, L I, N Y.

LEWIS, George M. (*M* 1937) Chief Engr Penobscot Bldg, Simon J Murphy Co, 1366 Penobscot Bldg, and (for mail) 14414 Grandmont Rd, Detroit, Mich

LEWIS, H. Frederick (*A* 1937) Vice-Pres (for mail) Dwight Oil Heat, 147 Dongan Ave, Albany, and Sweet's Crossing, Nassau, N Y

LEWIS, J. C. (*J* 1938) Salesman (for mail) York Ice Machinery Corp, 5051 Santa Fe Ave, and 6327-A Middleton St, Los Angeles, Calif

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LEWIS, L. Logan* (*M* 1918) Vice-Pres, Chief Engr. (for mail) Carrier Corp, 300 S Geddes St, and 207 Sedgewick Drive, Syracuse, N Y

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LEWIS, Thornton* (*M* 1919) (*Presidential Member*) (Pres, 1929; 1st Vice-Pres, 1928; 2nd Vice-Pres, 1927; Council, 1923-1930) Pres., Pulp Products Co, Inc., 60 East 42nd St, New York, N. Y, and (for mail) Holiday Hill, R. D. No. 2, Newtown, Pa

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LIEBRECHT, Walter J. (*J* 1936) Sales Engr. (for mail) American Radiator Co, Fourth and Channing Sts, N E, and 3032 Rodman St., N W, Washington, D C

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LOCKHART, Harold A. (*A* 1936, *J* 1935) Chief Engr, Bell & Gossett Co, 3000 Wallace St, and (for mail) 11749 Hale Ave, Chicago, Ill
LOCKHART, William R. (*J* 1936) Dist Sales Mgr. (for mail) York Ice Machinery Corp, 215 Investment Bldg, and 3430-30th St, N W, Washington, D C
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LOFTE, John A. (*J* 1936, *S* 1933) Engr, Pflugradt Co, 215 W Kilbourn Ave, and (for mail) 3117 W Highland Blvd, Milwaukee, Wis
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LOUCKS, David W. (*A* 1930) Supvr, Commercial Electric & Steam Sales (for mail) Duquesne Light Co, 435 Sixth Ave, and 535 Shelbourne Ave (Wilkinsburg) Pittsburgh, Pa
LOUGHRAN, Patrick H., Jr. (*J* 1937) Lab Engr, Washington Gas Light Co, 411-10th St, N W, and (for mail) 4513-49th St, Washington, D C
LOVE, Clarence H. (*M* 1919) Mfrs Agent, Nash Engineering Co, 317 Chamber of Commerce, and (for mail) 289 Norwalk Ave, Buffalo, N Y
LOVING, William H. (*J* 1936) Laboratory Supvr, Washington Gas Light Co, 411-10th St, N. W, and (for mail) 3901 Fulton St, N W, Washington, D C
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LYNN, Richard G. (*J* 1938) Htg & Air Cond Engr, Frigidaire Div, General Motors Sales Corp, 2446 University Ave, and (for mail) 2284 Highland Pkwy, St Paul, Minn
LYON, P. S. (*M* 1929) Pres and Gen Mgr (for mail) Cochrane Corp, 17th St below Allegheny Ave, and 3416 Warden Drive, Philadelphia, Pa
LYONS, Cornelius J. (*A* 1932) Sales Engr (for mail) Nash Engineering Co, Wilson Ave, and 5 Olmstead Pl, South Norwalk, Conn

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MACCUBBIN, Howard A (*M* 1934) Buyer, Montgomery Ward & Co, Chicago, and (for mail) 2135 Ridge Ave, Evanston, Ill
MACDONALD, Donald B. (*M* 1930) Sales Engr, Donald B Macdonald Co, 101 E. Walnut St, Kingston, Pa.
MacDONALD, Douglas J. (*M* 1935) Vice-Pres. (for mail) Dominion Radiator & Boiler Co, Ltd, Royce & Lansdowne Ave, and 96 Hudson Drive, Toronto, Ont, Canada
MacEACHIN, Graham C. (*M* 1938) Dist Engr, Frigidaire Div, General Motors Sales Corp, Air Conditioning Dept, 2615 West 7th St, and (for mail) 4613 El Campo Ave, Fort Worth, Tex
MACHEN, James T. (*A* 1938, *J* 1934) Chicago Branch Mgr (for mail) The Ric-wiL Co, 111 W Monroe St, and 420 Diversey Pkwy, Chicago, Ill
MACHIN, Donald W. (*J* 1935) Fuel Engr, The Pittsburgh & Midway Coal Mining Co, 816 Dwight Bldg, Kansas City, Mo, and (for mail) 2112 Vermont St, Lawrence, Kan
MACK, Emil H. (*A* 1938) Asst Sales Mgr., The Vilter Mfg Co, 2217 S First St, and (for mail) 2225 N Booth St, Milwaukee, Wis
MACK, Ludwig (*M* 1935) Dist Mgr, Cooling & Air Cond Div, B F Sturtevant Co, Cresmont & Haddon Aves, Camden, N J, and (for mail) 246 W Upsal St, Germantown, Philadelphia, Pa.

MacLACHLAN, Victor D. (*J* 1938) Sales Engr (for mail) Minneapolis-Honeywell Regulator Co, 637 Craig St, W, and 495 Prince Arthur, Apt 16, Montreal, P Q, Canada
MacMILLAN, Alexander R. (*M* 1936) Mgr, Educational Dept, Delco-Frigidaire Conditioning Div, General Motors Sales Corp, and (for mail) 130 Beverly Place, Dayton, O
MACRAE, Robert B. (*J* 1935) Air Cond Engr, E J Nell Co, Manila, P I
MACROW, Lawrence (*J* 1936) Branch Engr, Carrier Corp, 1201 Statler Bldg, Boston, and (for mail) 27 Arborough Rd, Roslindale, Mass
MacWATT, Donald A. (*M* 1938) Sales Engr, Powers Regulator Co, 231 East 46th St, New York, and (for mail) 4611-258th St, Great Neck, L I, N Y
MADDEN, John J. (*A* 1937) Owner (for mail) The Madden Co, 339 Warren St, Roxbury, and 16 Brown Ave, Roslindale, Mass
MADDUX, O. Lloyd (*M* 1935, *A* 1933) Owner, O Lloyd Maddux, 53 Park Place, New York, N Y, and (for mail) 95 Washington St, East Orange, N J
MADELY, Frederick J. (*A* 1936) Chief Estimator, Eastern Steel Products, Ltd, 1335 Delorimier Ave, and (for mail) 6370 Louis-Hemon St, Montreal, P Q, Canada
MADISON, Richard D (*M* 1926) Research Engr (for mail) Buffalo Forge Co, 490 Broadway, Buffalo, and 218 Brantwood Rd, Snyder, N Y
MAEHLING, Leon S (*M* 1932) Supt, Equitable Gas Co, 6304 Penn Ave, and (for mail) 414 Sulgrave Rd, Pittsburgh, Pa
MAGEE, Kevin B (*A* 1938) Air Cond Engr, P R Moses & Associates, 11 Park Place, and (for mail) 2308 Newtown Ave, Astoria, L I, N Y
MaGIRL, Willis J (*M* 1934, *A* 1931, *J* 1927) Chief Engr (for mail) P H MaGirl Foundry & Furnace Works, 401-13 E Oakland Ave, and 108 Warner Ave, Bloomington, Ill
MAGNUSSON, Nicholas (*A* 1938) Estimator-Designer, Montgomery Ward & Co, 150-15 Jamaica Ave, and (for mail) 138-05 Linden Blvd, Jamaica, L I, N Y
MAHER, Thomas F., Jr. (*A* 1937) Salesman, Kewanee Boiler Corp, 37 West 39th St, New York, and (for mail) 116-48 218th St, St Albans, L I, N Y
MAHON, B. B (*M* 1935) Principal of School of Air Cond (for mail) International Correspondence Schools, Wyoming Ave, and 433 Fig St, Scranton, Pa
MAHON, Clarence A. (*A* 1938) Pres-Mgr (for mail) Air Control Equipment Co, 1712 Main St, and 6123 Kenwood Ave, Kansas City, Mo
MAHON, Frank B. (*M* 1937) Air Cond Promotion, Duquesne Light Co, 435 Sixth Ave, and (for mail) 1241 Illinois Ave, Pittsburgh, Pa
MAHONEY, David J. (*M* 1930, *A* 1926) Branch Mgr (for mail) Johnson Service Co, 503 Franklin St, and 140 Linwood Ave, Buffalo, N Y
MAIER, George M. (*M* 1921) Asst to Vice-Pres and Gen Mgr of Mfg (for mail) American Radiator Co, 8007 Jos Campau, Detroit, Mich
MAIER, Herman F. (*M* 1926) Chief Engr-Secy, The New York Blower Co, 3155 Shields Ave, and (for mail) 7121 S Morgan St, Chicago, Ill
MAILLARD, Albert L. (*M* 1934) Consulting Engr, 3740 Washington St, Kansas City, Mo
MAKIN, Henry T., Jr. (*M* 1939) Engr, H B Smith Co, 2209 Chestnut St, and (for mail) 48 W Gowen Ave, Philadelphia, Pa
MALCOLM, Bernard L. (*J* 1937) Sales Engr (for mail) Sidles Co, Airtemp Div, 502 South 19th, and 4960 Military Ave, Apt 7, Omaha, Nebr
MALLIS, William (*M* 1914) Owner (for mail) 330 Lyon Bldg, and 723 Federal Ave, Seattle, Wash
MALLY, Chester F. (*A* 1938) General Mgr, Mally & Co, 307 Stormfeltz Loveley Bldg, Detroit, and (for mail) 2034 Central Ave, Ferndale, Mich
MALONE, Dayle G. (*M* 1929, *A* 1925) Vice-Pres (for mail) Petroleum Heat and Power Co, 1725 S Michigan Ave, and 7337 Merrill Ave, Chicago, Ill
MALONE, James S. (*A* 1936) Dist. Repr. (for mail) Hoffman Specialty Co, 411 N Tenth St., and 7124 Waterman Ave, St. Louis, Mo.
MALVIN, Ray C. (*M* 1929) Pres (for mail) Malvin & May, Inc, 2427 S. Michigan Ave., and 8220 Dante Ave, Chicago, Ill
MANDELL, Thomas P. (*A* 1937) Salesman, Carrier Corp, 1201 Statler Office Bldg, Boston, and (for mail) Walnut Rd, South Hamilton, Mass
MANN, Arthur R (*M* 1930) Partner (for mail) Mann & Co, Archts, 902 Wiley Bldg, and 122 W 15th St, Hutchinson, Kan
MANNING, Charles E. (*J* 1937) Sales Engr., Refrig and Air Cond, c/o Bond & Bond, Ltd, Auckland, New Zealand
MANNING, Walter M. (*M* 1930) Traveling Engr, Baker Mfg Co, Omaha, and (for mail) P O Box 112, Clarks, Nebr
MANNY, J Harvey (*A* 1936) Vice-Pres & Secy (for mail) Robinson Furnace Co, 213 W Hubbard St, and 5950 Midway Park, Chicago, Ill
MARCHIO, Emilio, Jr. (*S* 1938) 212 S Monroe, Kansas City, Mo
MARCONETT, Vernon G. (*A* 1936) Engr & Factory Supt, The Farquhar Furnace Co, and (for mail) Wilmington, O
MARIN, Axel* (*M* 1935) Assoc Prof Mech Engrg (for mail) University of Michigan, 241 W. Engineering Bldg, and P O Box 175, Ann Arbor, Mich
MARKLAND, Charles E. (*M* 1939) Mech Engr (for mail) 110 Power Plant, University of Illinois, Urbana, and 408 N Prairie, Champaign, Ill
MARKS, Alexander A. (*A* 1930) Chief Engr, Richmond Radiator Co, and (for mail) 818 Fayette Title & Trust Bldg, Uniontown, Pa.
MARKUSH, Emery U. (*M* 1931) Consulting Engr (for mail) 225 East 21st St, New York, and 8442-85th Rd, Woodhaven, L I, N Y.
MAROTTA, John A. (*S* 1936) 11116 Tuscora Ave, Cleveland, O
MARRINER, John M S (*M* 1934) Vice-Pres (for mail) Taylor Engineering & Construction Co, Ltd, 80 Richmond St, W, and 111½ Balsam Ave, Toronto, Ont, Canada
MARSCHALL, Peter J. (*M* 1930, *J* 1927) Engr, Kroeschell Engineering Co, 215 W Ontario St, and (for mail) 6434 N Seeley Ave, Chicago, Ill
MARSHALL, Albert W (*M* 1937) Asst Supt & Resident Engr, Soho Public Baths, 2410 Fifth Ave, Pittsburgh, Pa
MARSHALL, Alexander G. (*A* 1936) Sales Engr (for mail) Trane Co of Canada, Ltd, 660 St Catherine St, W, and 2353 Wellington St, Montreal, P Q, Canada
MARSHALL, Orville D. (*A* 1931) Mfrs Agent (for mail) 514 Anderson Bldg, and 1350 Calvin Ave, Grand Rapids, Mich
MARSHALL, R. Douglas (*A* 1938) Partner, Delavan Engineering Co, 414 12th St, and (for mail) 2719 Moyer Ave, Des Moines, Ia
MARSHALL, Stanley C. (*M* 1939) Chief Engr, Mayflower-Lewis Corp, Duluth Ave & East 7th St, St Paul, and (for mail) Oak Grove Hotel, Minneapolis, Minn
MARSHALL, Thomas A. (*J* 1937) Sales Engr, York Ice Machinery Corp, 1275 Folsom St, San Francisco, Calif
MARSHALL, William D. (*M* 1935) Branch Mgr (for mail) Noland Co, Inc, 1823 N Arlington Ridge Rd, and 1307 N Wakefield St, Arlington, Va
MARSTON, Anson D.* (*A* 1937) Industrial Engr, In Charge of Air Cond (for mail) Kansas City Power & Light Co, 1330 Baltimore, and 4943 Central, Kansas City, Mo
MARTEL, Charles L., Jr. (*J* 1937) Pres, Martel Heating Co, 13534 Cedargrove Ave, Detroit, Mich
MARTENS, Edward D. (*M* 1937) Mech Engr (for mail) Thompson Starrett Co, Inc, 444 Madison Ave, New York, and 89 Eldridge Ave., Hempstead, L. I, N Y

MARTIN, Albert B. (*M* 1917) Chicago Branch Mgr. (for mail) Kewanee Boiler Corp., 1858 S. Western Ave, Chicago, and 997 Vine St, Winnetka, Ill.

MARTIN, George W.* (*M* 1911) Supervising Engr. (for mail) U. S Realty & Improvement Co., 111 Broadway, New York, N. Y., and 340 Prospect St, Ridgewood, N. J

MARTIN, Leonard (*J* 1936) Sales Engr, H L. Peiler & Co, Ltd., 620 Cathcart St., Montreal, P. Q, Canada

MARTIN, Raymond (*A* 1937) Sales Engr (for mail) Vapor Car Heating Co of Canada, Ltd, 65 Dalhousie St, Montreal, and 825 Moffat Ave., Verdun, P Q., Canada.

MARTINEZ, Juan J. (*J* 1929) Research and Rate Engr, The Mexican Light & Power Co., Ltd, Gante 20, and (for mail) Paseo de la Reforma 183, Mexico, D F, Mexico.

MARTINKA, Paul D. (*J* 1937; *S* 1934) 13703 Chautauqua Ave., Cleveland, O

MARTOCELLO, Joseph A. (*M* 1934) Pres, Jos A. Martocello & Co, 229 North 13th St, Philadelphia, Pa

MARTY, Edgar O. (*M* 1916) Pres and Gen Mgr, Indian Head Anthracite, Inc., Thompson Bldg, and (for mail) 1775 Howard Ave, Pottsville, Pa.

MARTYN, Henry J. (*A* 1937) Pres. (for mail) Martyn Brothers, Inc, 911 Camp St, and 5306 Ridgedale St, Dallas, Tex.

MARZOLF, Frank X. (*A* 1937) Sales Engr, Minneapolis-Honeywell Regulator Co., 415 Brainard St., and (for mail) 15046 Mettetal, Detroit, Mich.

MARZORATI, Giuseppe (*M* 1938) Consulting Engr, S i N.C. (for mail) Giacomo Jucker, 28 Mauro Macchi, and Via Baldissera 9, Milano, Italy

MASON, Gail C. (*M* 1939; *A* 1937) Air Cond Engr (for mail) The Williamson Heater Co, 337 W Fifth St, and Hotel Sinton, Cincinnati, O

MASTERS, Robert B. (*A* 1938) Inspector, Pacific Greyhound Lines, 401 Kansas St., San Francisco, and (for mail) General Delivery, Oakland, Calif

MATCHETT, James C. (*M* 1923) Vice-Pres. and Gen Mgr. (for mail) Illinois Engineering Co, Racine Ave and 21st St., and 9936 South Winchester Ave., Chicago, Ill.

MATHER, Harry H. (*A* 1929) Industrial Promotion (for mail) Philadelphia Electric Co., 1000 Chestnut St, Philadelphia, and 373 Lakeview Ave., Drexel Hill, Pa.

MATHEWSON, Marvin E. (*M* 1937) Secy (for mail) A M. Kinney, Inc., 1820 Carew Tower, and 2156 Alpine Place, Cincinnati, O.

MATHIS, Eugene* (*M* 1922) Vice-Pres & Treas (for mail) The New York Blower Co., 32nd Street & Shields Ave., and 9151 S. Hoyne Ave, Chicago, Ill.

MATHIS, Henry (*M* 1921) New York Blower Co, 32nd St. and Shields Ave, and (for mail) 10317 Oakley Ave, Chicago, Ill

MATHIS, John (*A* 1938) Engr., Standard Furnace & Supply Co, 407 South 10th St, and (for mail) 109 South 42nd St, Omaha, Nebr.

MATHIS, Julian W. (*A* 1921) New York Blower Co, 32nd and Shields Ave, Chicago, Ill

MATHISON, Russell S. (*A* 1938) Asst. Gen. Mgr. (for mail) Weathermakers (Canada) Ltd., 593 Adelaide St, W, and 44 Strathgowan Ave, Toronto, Ont., Canada.

MATOUSEK, A. G. (*M* 1937) Air Cond Engr, York Ice Machinery Corp., 117 S. 11th St., and (for mail) 1528 Locust St., St. Louis, Mo.

MATTHEWS, John E. (*M* 1934) Dist. Mgr, B. F. Sturtevant Co., 1106 Commerce Bldg., and (for mail) 5642 Lydia St., Kansas City, Mo.

MATTHEWS, Wesley M. (*J* 1937) Sales Engr, Sidles Co., Airtemp Div., 425 Stuart Bldg, Lincoln, and (for mail) P. O. Box 685, Scottsbluff, Nebr

MATZ, George N. (*M* 1938) Mech. Engr, A. Ernest D'Ambly, 901 Architects Bldg., Philadelphia, and (for mail) 649 Ferne Ave., Drexel Hill, Pa.

MAUTSCH, Robert (*A* 1928) Engr., Managing Dir, Compagnie Belge des Freins Westinghouse, 97 Avenue Louise, Brussels, Belgium.

MAWBY, Pensyl (*M* 1934) Dist. Sales Mgr., Lehigh Navigation Coal Co., 1421 Chestnut St., Philadelphia, and (for mail) 15 Golf Rd., Lansdowne, Pa.

MAXWELL, George W. (*M* 1935; *S* 1932) Engr, Kenealy & Maxwell, Main St., and (for mail) Lower County Rd., Harwich Port, Mass

MAXWELL, Robert S. (*M* 1937) Gen. Mgr (for mail) Bennett & Wright, Ltd., 72 Queen St, E., and 560 Briar Hill Ave., Toronto, Ont, Canada.

MAY, Arthur O. (*A* 1938; *J* 1928) Sales Engr. (for mail) Stannard Power Equipment Co, 53 W. Jackson Blvd, and 5736 N. Bernard St, Chicago, Ill.

MAY, Clarence W. (*M* 1933) Consulting Engr. (for mail) 1201 Smith Tower, and 6056 4th, N. E., Seattle, Wash

MAY, Edward M. (*M* 1931) Branch Mgr, Steel Products Engineering Co, 1601 S Michigan Ave, Chicago, and (for mail) 848 N Ridgeland Ave, Oak Park, Ill

MAY, George E. (*M* 1933) Utilization Engr. (for mail) New Orleans Public Service, Inc., 317 Baronne St, and 2031 Short St, New Orleans, La.

MAY, James W. (*M* 1938; *J* 1935) Assoc Prof of Htg-Vtg (for mail) College of Engrg., University of Kentucky, and 261 Lyndhurst Place, Lexington, Ky

MAY, Maxwell F. (*M* 1929) Secy-Treas (for mail) Malvin & May, Inc, 332 S Michigan Ave, Chicago, and Palos Park, Ill

MAYER, Robert L. (*A* 1938) Sales Engr., Smedley & Mehl Co, 200 W. Montgomery Ave, Ardmore, and (for mail) 43 W Albemarle Ave, Lansdowne, Pa

MAYER, Robert W. (*A* 1937) Dist. Mgr. (for mail) Minneapolis-Honeywell Regulator Co, 561 Reading Rd, and 3980 Rose Hill Ave, Cincinnati, O

MAYETTE, Charles E. (*M* 1926) Consulting Engr, Room 1417, Graybar Bldg, 420 Lexington Ave, New York, N. Y

MAYNARD, J. Earle (*M* 1931) Chief Htg Engr., Fox Furnace Co, Woodford St, and (for mail) 324 Fifth St, Elyria, O.

MAYNE, Walter L. (*M* 1937) Brance Mgr (for mail) U S Radiator Corp, Cor Wayne and C L. & N. R R and 1239 Delta Ave, Cincinnati, O.

McCAFFERTY, Joseph E. (*A* 1937) Dist Engr, Petroleum Heat and Power Co, 419 Boylston St, Boston, and (for mail) 196 Manthorne Rd., West Roxbury, Mass.

McCAFFRAY, Charles E. (*M* 1938) Engr, Henry Adams, Inc, Consulting Engineers, 1263 Calvert Bldg, and (for mail) 1 Hamilton Court, Hamilton St, Baltimore, Md

McCAIN, H. King (*A* 1938; *J* 1937) Sales Engr, Delco-Frigidaire Div, General Motors Sales Corp., Dayton, O, and (for mail) 406 Emoriland Blvd, Knoxville, Tenn.

McCARTHY, John J. (*A* 1937) Chief Engr. (for mail) Providence Public School Dept., 20 Summer St, and 318 Academy Ave., Providence, R I.

McCARTHY, Thomas F. (*M* 1938) Dist Mgr. (for mail) Frigidaire Corp, 2031 S Calumet Ave, and 6948 Calumet Ave, Chicago, Ill

McCAULEY, James H. (*M* 1921) Pres (for mail) J. H. McCauley, Inc, 5558 West 65th St., Chicago, and 707 William St, River Forest, Ill.

McCLAIN, Clifford H. (*M* 1937) Htg. Engr, Upper Darby Plumbing & Heating Co, Inc., 7127 Marshall Rd, and (for mail) 1600 Darby Rd, Brookline, Upper Darby, Pa.

McCLELLAN, James E. (*M* 1922) Mgr., Chicago Office (for mail) American Blower Corp, 228 N. LaSalle St., Chicago, and 738 Marion Ave., Highland Park, Ill.

McCLINTOCK, Alexander, Jr. (*M* 1928; *J* 1920) Htg. Contractor (for mail) A. McClintock's Sons, 1937 Ridge Ave., and Rochelle Ave., Philadelphia, Pa.

McCLINTOCK, William (*M* 1935) Supervising Engr, Engrg. Design Section, Administrative Staff, U S. W P A., 70 Columbus Ave, and (for mail) 643 East 232nd St, New York, N Y

McCONACHIE, Lorne L. (*A* 1928) Htg and Plbg., 1003 Maryland Ave, and (for mail) 1379 Maryland Ave, Detroit, Mich

McCONNER, Charles R. (*A* 1925; *J* 1922) Gen Sales Mgr, Clarage Fan Co, Kalamazoo, Mich.

McCORMACK, Denis (*M* 1933) Mgr, Air Cond. Instruments and Controls Dept (for mail) Julien P. Friez & Sons, Inc, 4 N. Central Ave., Baltimore, and Ruxton Post Office, Baltimore County, Md

McCOY, C. E. (*M* 1936) Partner (for mail) Turner-McCoy, 210 W Second St, and 3922 S Lookout Ave, Little Rock, Ark

McCOY, Thomas F. (*M* 1924) Mgr (for mail) The Powers Regulator Co, 125 St Botolph St, Boston, and Glen Rd, Wellesley Farms, Mass

McCRAE, George W. (*A* 1936) Chief Engr, John McCrae Machine & Foundry Co, 77-85 William St, N, and (for mail) 51 Bond St., Lindsay, Ont, Canada

McCREA, Joseph B. (*M* 1937) Owner, Heating & Ventilating, 3039 Coplin Ave, Detroit, Mich

McCREERY, Hugh J. (*M* 1922) Owner (for mail) 335 Burrard St, and 1617-49th Ave, W, Vancouver, B C

McCULLOUGH, Henry G. (*M* 1936) Mgr, Commercial Dept, S S Fretz, Jr, Inc, 1902 Chestnut St, Philadelphia, and (for mail) 328 Glen Echo Rd., Germantown, Philadelphia, Pa

McCUNE, Byron V. (*M* 1928) 2310 W Yakima Ave, and (for mail) 101 W. Yakima Ave, Yakima, Wash

McDONALD, Anthony K. (*A* 1936) Sales Engr, Standard Oil Co of New Jersey, 261 Constitution Ave, N W, and (for mail) 3035 Rodman St, N W, Washington, D C

McDONALD, Ivan (*A* 1938) Dist Repr. (for mail) Minneapolis-Honeywell Regulator Co, Ltd, 791 Erin St, and Ste 16 Bayview Apt, Winnipeg, Man, Canada

McDONALD, James J. (*J* 1938) Research Engr, Nash Kelvinator, and (for mail) 3974 Commonwealth Ave, Detroit, Mich

McDONALD, Thomas (*A* 1931) Vice-Pres, Minneapolis-Honeywell Regulator Co, 2747 Fourth Ave, S, Minneapolis, Minn

McDONNELL, Everett N. (*M* 1923) Pres (for mail) McDonnell & Miller, 400 N. Michigan Ave, and Drake Hotel, Chicago, Ill

McDONNELL, John E. (*A* 1936) Vice-Pres (for mail) McDonnell & Miller, 400 N Michigan Ave, Chicago, and 2421 Central Park, Evanston, Ill

McDOWELL, Harry L. (*J* 1939) Htg Engr., Syska & Hennessey, Consulting Engrs. (for mail) 111 Corcoran St, and 310 Holloway St., Durham, N. C

McELGIN, John W.* (*A* 1937; *J* 1931) Limekiln and Butler Pikes, Ambler, Pa.

McELHANEY, Gerald W. (*A* 1938; *J* 1936) Air Cond Engr. (for mail) Ohio Edison Co, Akron, and 1924 Tenth St, Cuyahoga Falls, O.

McEWAN, Eugene E. (*M* 1936) N Y. Mgr Air Cond Div. (for mail) Frigidaire Div., General Motors Sales Corp, 224 W. 57th St, and Salisbury Hotel, 123 W. 57th St, New York, N. Y.

McGAUGHEY, Harold M. (*M* 1937) Sales Mgr, Commercial Air Conditioning & Automatic Htg, Nash Kelvinator Corp, and (for mail) 300 Whitmore Rd, Detroit, Mich.

McGEORGE, Richard H. (*M* 1927) Mgr, Htg. & Air Cond. Dept, McCord Radiator & Mfg Co, 2587 E. Grand Blvd., and (for mail) 14565 Glastonbury Rd., Detroit, Mich

McGONAGLE, Arthur (*M* 1932) Consulting Engr. (for mail) 1913 Fulton Bldg., Pittsburgh, and 6815 Prospect Ave., Ben Avon, Pa.

McGRAIL, Thomas E. (*M* 1926) Local Repr, Canadian Sirocco Co., Ltd., P. O. Box 555 Station B., Ottawa, Ont., Canada.

McILVAINE, John H.* (*M* 1929) Vice-Pres. and Treas, Landwehr Heating Corp., Sixth and Cuyuga Sts., Philadelphia, Pa.

McINTIRE, James F. (*M* 1915; *A* 1914) (1st Vice-Pres., 1938; 2nd Vice-Pres., 1937; Council, 1926-1928; 1932-1938) Vice-Pres. (for mail) U. S. Radiator Corp, 1056-44 Cadillac Square, P. O. Box 686, and 3261 Sherbourne Rd, Detroit, Mich

McINTIRE, James L. (*J* 1938) Sales Engr., (for mail) York Ice Machinery Corp., 117-121 South 11th St, and 5164 Washington St, St Louis, Mo.

McINTOSH, Fabian C (*M* 1921; *J* 1917) (Council, 1929-1931; 1933-1935) Branch Mgr (for mail) Johnson Service Co, 1238 Brighton Rd, and 302 Marshall Ave, Pittsburgh, Pa.

McKEE, James W. (*A* 1938) Branch Mgr. (for mail) U S Radiator Corp., 532 E Corcoran Ave., and 6713 W Bluemound Rd., Milwaukee, Wis.

McKEEMAN, Clyde A.* (*M* 1936) Asst. Prof. of Mech Engrg (for mail) Case School of Applied Science, Cleveland, and 1359 Lynn Park Drive, Cleveland Heights, O

McKENZIE, Murdock C., Jr. (*M* 1938) Htg. Engr, Southern California Gas Co, 810 S. Flower St, and (for mail) 3806 Boyce Ave, Los Angeles, Calif

McKERLIE, Jardine (*M* 1938) 15 Glenarden Rd., Toronto, Ont, Canada.

McKINLEY, Carroll B. (*J* 1936; *S* 1934) Sales Engr (Consulting) General Refrigeration Corp., and (for mail) 1123 Harrison, Beloit, Wis.

McKINNEY, Carl A. (*J* 1937) Air Cond. Engr. (for mail) United Gas Corp, 1018 Rusk Bldg., and 1904 Brun St, Houston, Tex.

McKINNEY, William J. (*M* 1938; *A* 1934) Mgr., Atlanta Dist (for mail) American Blower Corp., 716-101 Marietta St. Bldg, and 3363 Mathieson Drive, Atlanta, Ga.

McKITRICK, Walter D. (*M* 1936) Htg-Vtg Engr (for mail) Mills, Rhines, Bellman & Nordhoff, Archts. & Engrs, 518 Jefferson Ave., and 3038 Gunckel Blvd, Toledo, O

McKITTRICK, Percy A. (*A* 1934) Treas.-Gen. Mgr (for mail) Parks-Cramer Co., 970 Main St., and 219 Blossom St, Fitchburg, Mass.

McLAREN, Fred S. (*J* 1935) Air Cond. Sales Engr (for mail) Frigidaire Div., General Motors Sales Corp, 4436 Toulouse, and 905 Fern St., New Orleans, La

McLAREN, T. H. (*A* 1938) Gen. Sales Mgr. (for mail) The James Morrison Brass Mfg. Co., Ltd., 276 King St, W, and 2084 Girard St., E., Toronto, Ont, Canada

McLARNEY, Harry W. (*M* 1933) Air Cond Engr. (for mail) Union Electric Co. of Missouri, 315 N. 12th Blvd, and 5038 Bancroft Ave., St. Louis, Mo

McLAUGHLIN, Joseph D. (*A* 1930; *J* 1928) Htg. Contractor (for mail) Braley & McLaughlin, 166 Aborn St, and 45 Roslyn Ave, Providence, R. I.

McLEAN, Dermid (*M* 1917) Member of Firm (for mail) Snyder & McLean, 2308 Penobscot Bldg., and 12651 Birwood Ave, Detroit, Mich.

McLEAN, James E. (*M* 1936) 520 Bigham Rd., Pittsburgh, Pa

McLEISH, William S. (*A* 1932; *J* 1928) Sales Engr (for mail) The Ric-wiL Co, Room 1838, 101 Park Ave, New York, and 6446-184th St., Flushing, N. Y.

McLENEGAN, David W.* (*M* 1933) Asst. Engr., Comm. Engr. Div, Air Cond. Dept (for mail) General Electric Co., 5 Lawrence St., Bloomfield, and 73 Arlington Ave., Caldwell, N. J.

McLOUTH, Bruce F. (*M* 1936; *J* 1934) Chief Engr, Heater Div. (for mail) Dail Steel Products Co, Lansing, and 135 Gunson, East Lansing, Mich.

McMAHON, Thomas W. (*M* 1928) Dist. Mgr. (for mail) American Blower Corp., 1711 Railway Exchange Bldg, and 6173 Waterman Blvd., St. Louis, Mo

McMULLEN, Earle W. (*M* 1938) Dir. of Research (for mail) The Eagle-Picher Lead Co., and 626 Jaccard Place, Joplin, Mo.

McNAMARA, William (*A* 1930) Mgr (for mail) The Trane Co, 2694 University Ave, and 1355 Como Ave, W, St Paul, Minn

McNEVIN, Joseph E. (*M* 1937) Mgr (for mail) Colorado Heating Co, 950 Cherokee St, and 225 E Dakota Ave, Denver, Colo

McPHERSON, William A. (*M* 1929) Chief, Htg-Vtg Div, Dept of School Bldgs, 26 Norman St, Boston, and (for mail) 86 Dwinnell St, West Roxbury, Mass

McQUAID, Daniel J. (*M* 1934) Owner (for mail) Daniel J McQuaid, Engineering Service, 614 Cooper Bldg, and 1565 Milwaukee St, Denver, Colo

McREYNOLDS, C. V. (*A* 1938) Engr, Natkin & Co, 212 Iowa Bldg, and (for mail) 3905-8th St, Des Moines, Ia

MEAD, Edward A. (*M* 1926) Asst Sales Mgr (for mail) Nash Engineering Co, South Norwalk, and 5 Thames St, Norwalk, Conn

MEAGHER, Arthur T. (*M* 1938) Dir and Sales Mgr, Plbg & Htg Dept, Wm Stairs, Son & Morrow, Ltd, 174-190 Lower Water St, and (for mail) 83 Seymour St, Halifax, Nova Scotia, Canada

MEAKIN, John B. (*J* 1935) Sales Engr (for mail) Foxboro Co, Neponset Ave, and 12 Baker St, Foxboro, Mass

MEARS, Leon A. (*A* 1938, *J* 1935) 721 Alice St, and (for mail) 975 Sunnyhills Rd, Oakland, Calif

MEDOW, Jules (*J* 1937) Designing Engr, Ilg Electric Ventilating Co, 2850 N Crawford Ave, and (for mail) 147 S Springfield Ave, Chicago, Ill

MEHL, Oscar H. (*J* 1935) Engr (for mail) Carrier Corp, 2022 Bryan St, and 4841 Tremont St, Dallas, Tex

MEHNE, Carl A. (*M* 1929) Htg-Vtg Expert, C A Mehne, Room 821, 101 Park Ave, New York, and (for mail) 35 Livingston St, Valhalla, N Y

MEINHOLTZ, Herbert W. (*M* 1936) 608 Mayo Bldg, Tulsa, Okla

MEINKE, Howard G. (*M* 1933) Div Engr (for mail) Consolidated Edison Co of New York, Inc, 4 Irving Place, New York, and 41 Harte St, Baldwin, L I, N Y

MELLON, James T. J. (*M* 1911) Pres (for mail) Mellon Co, 4419 Ludlow St, and 431 N 63rd St, Philadelphia, Pa

MELONEY, Edward J. (*M* 1937) Vice-Pres (for mail) Bowers Bros Co, 2015 Sansom St, Philadelphia, and 100 E Stewart Ave, Lansdowne, Pa

MENDEN, Peter J (*M* 1935) Htg Engr, Advance Heating Co, 910 Herrick Ave, and (for mail) 1509 Arthur Ave, Racine, Wis

MENSING, Frederick D. (*M* 1920) (Treas, 1931-1932) Consulting Engr, Mensing & Co, 2845 Frankford Ave, Philadelphia, Pa

MERCER, Charles F. (*M* 1937) Prof Physics (for mail) University of South Carolina, and 219 S Waccamaw, Columbia, S C

MERENS, Seymour H. (*A* 1939) Sales Engr & Estimator, T H Litvin Plumbing & Heating Co, 610 W Randolph, and (for mail) 3355 Eastwood Ave, Chicago, Ill

MERLE, André (*M* 1934) Engr (Air Cond & Refrig), Office of the Q M G., War Dept, U S A, and (for mail) 1501 Massachusetts Ave, N W, Washington, D C

MERRILL, Carle J. (*M* 1919) Treas (for mail) C J Merrill, Inc, 54 St. John St, and 15 Longfellow St., Portland, Me

MERRILL, Frank A. (*M* 1934) Consulting Engr (for mail) Office of Hollis French, Consulting Engrs, 210 South St, Boston, and 19 Auburndale Rd, Marblehead, Mass

MERTZ, Walter A. (*M* 1919) Secy (for mail) Kehm Bros Co, 51 E Grand Ave, and 3753 N. Keeler Ave, Chicago, Ill.

MERWIN, Gile E. (*M* 1924; *J* 1923) Secy-Treas, Rockford Plumbing Supply Co, 700 S Main St, and (for mail) 1536 Myott Ave, Rockford, Ill.

MESSENGER, Theodore I. (*A* 1936) Power Engr (for mail) Buffalo Niagara & Eastern Power Corp, 1107 Electric Bldg, and 263 Highland Ave, Buffalo, N Y

METCALFE, Curtis (*A* 1937) Engr, House Htg. Dept, Detroit City Gas Co, and (for mail) 8575 Dumbarton Rd, Detroit, Mich

METZGER, H. J. (*A* 1937) Vice-Pres and Htg Supt, Wheeler-Blaney Co, 249 N. Burdick St, and (for mail) 706 Locust St, Kalamazoo, Mich.

MEYER, Charles L. (*M* 1930) Mech Engr., American Welfare League, 86-97 Palo Alto Ave, Hollis, L I, N Y.

MEYER, Frank L. (*M* 1932, *J* 1928) Vice-Pres, The Meyer Furnace Co, and (for mail) 9 Cole Court, Peoria, Ill

MEYER, Henry C., Jr.* (*Life Member, M* 1898) (Council, 1915-1916) Pres (for mail) Meyer, Strong & Jones, Inc, 101 Park Ave, New York, N Y, and 25 Highland Ave, Montclair, N J

MEYER, Karl A. (*M* 1938) Design Engr (Fan) L J Mueller Furnace Co, 2005 W Oklahoma Ave, and (for mail) 3171 North 15th St, Milwaukee, Wis

MEYERS, John (*M* 1937) Branch Mgr, Temperature Regulation, Johnson Service Co, Bond Bldg, 14th & New York Ave, N E, Washington, D C

MICHIE, D. Fraser (*M* 1938, *A* 1930) Engrg Sales Dept (for mail) Crane Ltd, 93 Lombard St, and Ste 26, 75 Kennedy St, Winnipeg, Man, Canada

MIDDLETON, David K. (*J* 1936) Salesman, Johnson Service Co, 1100 N W 38th St, Oklahoma City, Okla

MIDDLETON, Howard A (*A* 1935) Engr, 525 E Armour, Kansas City, Mo

MIDEKE, Joseph M. (*A* 1937) Vice-Pres, Mideke Supply Co, 100 E Main St, and (for mail) 2003 N W, 13th St, Oklahoma City, Okla

MILENER, Eugene D. (*M* 1936) Secy, Industrial Gas Section, American Gas Association, 420 Lexington Ave, New York, N Y

MILES, Clarence N (*A* 1937) Foreman, Assembly Dept, Kohlenberger Engineering Corp, 805 S Spadra Rd, and (for mail) Rte 1, Box 174 A, Fullerton, Calif

MILLARD, E. L. (*A* 1938) Chief Engr (for mail) A Y McDonald Mfg Co, 1201 Dodge St, and 4238 Larimore Ave, Omaha, Nebr

MILLARD, Junuis W. (*M* 1929) Dist Mgr, Carrier Corp, 1201 Statler Bldg, Boston, and (for mail) 7 Tappan Rd, Wellesley, Mass

MILLEN, Ralph J (*A* 1938) Estimating and Layout Engr, Haried Home Appliance, 121 Downer Place, and (for mail) 933 Harriett Ave, Aurora, Ill

MILLER, Archibald T. (*M* 1938) Mgr, Insulation Sales, The Barrett Co, 40 Rector St, New York, N Y, and (for mail) 125 Godwin Ave, Ridgewood, N J

MILLER, Bruce R. (*M* 1935, *A* 1930) Mech Engr, 1533 Northwest 25th St, Oklahoma City, Okla

MILLER, Chas A. (*A* 1917) Salesman, The H B Smith Co, Inc, 36-46-33rd St, Long Island City, and (for mail) 2870 Marion Ave, New York, N Y

MILLER, Charles W. (*M* 1919, *J* 1908) Pres (for mail) The Rado Co, 759 N Milwaukee St, Rm 405, Milwaukee, and R-1, Box 42, Menomonee Falls, Wis

MILLER, Edgar R. (*A* 1935) Chief Engr (for mail) Winnipeg Cold Storage, Cor Jarvis and Salter, and Ste 0, Bexley Court, Winnipeg, Man, Canada.

MILLER, Floyd A. (*M* 1911) Inspection Engr. (for mail) U S Treasury Dept, 377 U S Court House, and 944 Montrose Ave, Chicago, Ill

MILLER, George F. (*M* 1936) Owner (for mail) Geo F Miller Sales Engr, 1625 K St, N W, Washington, D C, and 209 Connecticut Ave, Kensington, Md.

MILLER, Glen (*A* 1937) Htg-Vtg Engr (for mail) Southern Counties Gas Co, 810 S Flower St, Los Angeles, and 685 Luton Drive, Glendale, Calif

MILLER, Jack E. (*J* 1938) Engr (for mail) Fairbanks, Morse & Co, 217 S 8th St, St Louis, Mo, and 7325 Phillips Ave, Chicago, Ill

MILLER, Jacob (*M* 1936) Pres (for mail) Universal Heating Co, Inc, 121 St Marks Place, New York, and 435 East 92nd St, Brooklyn, N Y

MILLER, James E. (*M* 1914, *J* 1912) Htg Contractor, 2210 Colfax St, Evanston, Ill

MILLER, John F G. (*M* 1916) Vice-Pres (for mail) B F Sturtevant Co, Hyde Park, Boston, and 20 Chapel St, Brookline, Mass

MILLER, Leo B. (*M* 1926) Mgr, Refrigeration and Air Cond Div (for mail) Minneapolis-Honeywell Regulator Co, 2753 Fourth Ave, S, and 4255 E Lake Harriet Blvd, Minneapolis, Minn

MILLER, Lorin G.* (*M* 1933) Head, Mech Engrg Dept (for mail) Michigan State College, and 525 Albert St, East Lansing, Mich

MILLER, Merl W. (*M* 1932, *J* 1926) Plant Engr, The Trane Co, and (for mail) 333 North 23rd St, LaCrosse, Wis

MILLER, Robert A.* (*M* 1931) Tech Sales Engr (for mail) Pittsburgh Plate Glass Co, 2200 Grant Bldg, Pittsburgh, and 1211 Carlisle St, Tarentum, Pa

MILLER, Robert E (*J* 1935) Sales Engr (for mail) American Radiator Co, 1344 Broadway, and 18264 Birchcrest Drive, Detroit, Mich

MILLER, Robert T. (*A* 1927) Chief Engr, Sales Dept (for mail) Masonite Corp, 111 W Washington St, Chicago, and Flossmoor, Ill

MILLER, Tolbert G (*A* 1929, *J* 1921) Supt and Engr, Herre Bros, Seventh and Emeralds St, Harrisburg, and (for mail) 11 N Second St, Wormleysburg, Pa

MILLER, William T. (*M* 1938) Prof Htg-Vtg (for mail) Purdue University, and 525 Hayes St, West Lafayette, Ind

MILLHAM, Franklyn B. (*M* 1937) Installation Mgr, S S Fretz, Jr, Inc, 1902 Chestnut St, and (for mail) 532 Ellet St, Philadelphia, Pa

MILLIKEN, J. H* (*M* 1923) Repr (for mail) American Air Filter Co, Inc, 20 N Wacker Drive, Chicago, and 1021 Ridge Court, Evanston, Ill

MILLIS, Linn W (*Life Member, M* 1918) Secy, Security Stove & Mfg Co, 1630 Oakland, and (for mail) 3534 Wabash Ave, Kansas City, Mo

MILLS, Clarence A.* (*M* 1936) Prof of Experimental Medicine (for mail) University of Cincinnati, Cincinnati General Hospital, and 5016 Oberlin Blvd, Cincinnati, O

MILLS, Hartzell C (*A* 1935) Salesman, Minneapolis Gas Light Co, 800 Hennepin Ave, and (for mail) 4137 Tenth Ave, S, Minneapolis, Minn

MILNE, Arthur H. (*M* 1938) Dir Dept of Bldgs, Protestant Board of School Commissioners of the City of Montreal (for mail) 3460 McTavish St, and 4786 Grosvenor Ave, Montreal, P Q, Canada

MILWARD, Robert K. (*A* 1920) Mgr (for mail) U S Radiator Corp, 127 Campbell Ave, and 2411 Calvert Ave, Detroit, Mich

MIRABILE, Jasper J. (*A* 1938) Sales Engr, Htg Div, Asbestos Insulating Co, Astor & Main Sts, Philadelphia, and (for mail) 714 W Marshall St, Norristown, Pa

MITCHELL, Charles H. (*M* 1924) Engr, The Fels Co, 42 Union St, Portland, and (for mail) 25 Everett Ave, South Portland, Me

MITCHELL, Jack (*M* 1938, *J* 1930) Mgr, Air Cond Dept (for mail) Straus-Frank Co, and 2222 Nebraska, Houston, Tex

MITCHELL, John A. (*J* 1938) Sales Engr, Air Cond and Refrigeration Systems (for mail) 202 Waterloo Bldg, and 1301 Jefferson St, Waterloo, Ia.

MITCHELL, John G. (*J* 1937; *S* 1936) Sales Engr. (for mail) Fairbanks, Morse & Co, 220 E 5th St., St Paul, and 704 Delaware, S E., Minneapolis, Minn

MITTENDORFF, Edward M. (*M* 1932) Asst. Engr (for mail) Sarco Co, Inc, Merchandise Mart, Chicago, and 956 Greenwood Ave, Winnetka, Ill

MODIANO, Rene (*M* 1925) Managing Dir., Carrier Continentale, 4, Rue d'Aguesseau, Paris (8s) and (for mail) 55 Boulevard Beauséjour, Paris (16s), France

MOFFAT, Ormond G. (*A* 1937) Application Engr, Canadian Westinghouse Co, Sanford Ave, and (for mail) 141 George St, Hamilton, Ont, Canada

MOFFITT, Lloyd C. (*J* 1937) Branch Engr (for mail) Sidles Co, Airtemp Div, 509 S 19th, and 3109 Mason, Omaha, Nebr

MOHN, H. Leroy (*M* 1937) Development Engr, Fitzgibbons Boiler Co, Inc, E 10th & Mercer St, and (for mail) 136 E 4th St, Oswego, N Y

MOHRFELD, Herbert H. (*J* 1935) Air Cond Engr (for mail) C P Mohrfeld, Inc, 24 Lees Ave, Collingswood, and 131 Chestnut St, Haddonfield, N J

MOLER, William H (*M* 1927, *J* 1923) Vice-Pres (for mail) Kribs & Landauer, 200 Houseman Bldg, Dallas, and Box 69 A, R F D No 1, Irving, Tex

MOLFINO, Philip (*M* 1938) Mech Engr (for mail) Leland & Haley, 58 Sutter St, and 125 Clayton St, San Francisco, Calif

MOLLENBERG, Harold J. (*M* 1936) Vice-Pres, Mollenberg-Betz Machinery Co, 22 Henry St, Buffalo, and (for mail) 172 Westgate Rd., Kenmore, N Y

MOLONEY, Roger R (*M* 1937) Design Eng, Dept of Interior, Commonwealth Govt of Australia, Canberra, A F C, and (for mail) 26 Bonner Ave, Manley, Sydney, Australia

MONICK, Fred R (*A* 1936) Mgr (for mail) Cochran-Sargent Co, 605 E 8th St, and 1114 S 6th Ave, Sioux Falls, S D

MONTGOMERY, Edward G (*A* 1938) Special Repr, Steel Co of Canada, Ltd, 525 Dominion St, Montreal, and (for mail) 20 Finchley Rd, Hampstead, P Q, Canada

MONTGOMERY, John R. (*A* 1937) Mgr, Standards and Research (for mail) Truscon Steel Co, Albert St, and 296 Granada Ave, Youngstown, O

MONTGOMERY, Ora C. (*M* 1933) Asst Supt of Power (for mail) New York Central Railroad, Grand Central Terminal, Room 1842, 70 East 45th St, and 255 West 84th St, New York, N Y

MOODY, Lawrence E. (*M* 1919) Partner (for mail) Moody & Hutchison, Consulting Engineers, 1701 Architects Bldg, Philadelphia, Pa, and 237 Jefferson Ave, Haddonfield, N J

MOON, L Walter (*M* 1915) Pres (for mail) Bradley Heating Co, 3834 Olive St, and 5006 N Kings Highway, St Louis, Mo

MOORE, Bill J, Jr (*J* 1937) Pres, U S Air Conditioning Sales Corp, 1701 Grand Ave, and (for mail) 1305 Valentine Rd, Kansas City, Mo

MOORE, Don R. (*S* 1936) 402 W Penn St, Hoopeston, Ill

MOORE, Frank C. (*A* 1938) Canadian Mgr (for mail) Aerofin Corp, 67 Yonge St, and 323 Manor Rd, Toronto, Ont, Canada

MOORE, H Carlton* (*M* 1935) Asst Prof Mech Engrg (for mail) Massachusetts Institute of Technology, Room 1-202, Cambridge, and 145 Beaumont Ave, Newtonville, Mass

MOORE, H. Lee (*M* 1919) (Council, 1927-1928) Repr (for mail) Buffalo Forge Co, 431 Fulton Bldg, Pittsburgh, and Flaccus Rd, Ben Avon, Pittsburgh, Pa

MOORE, Henry W. (*M* 1935) Mgr Air Cond Engrg Dept (for mail) The Bimel Co, 305 Walnut St, and Cincinnati Club, Cincinnati, O.

MOORE, Herbert S (*A* 1923) Dist Repr, Iron Fireman Mfg Co of Canada, Ltd, 602 King St and (for mail) 107 Clendenan Ave, Toronto, Ont, Canada.

MOORE, R. Edwin (*A* 1928) Vice-Pres., Bell & Gossett Co., 3000 Wallace St., Chicago, and (for mail) 425 Merrill Ave , Park Ridge, Ill.
MOORE, Wesley R. (*M* 1937) Branch Mgr. (for mail) Minneapolis-Honeywell Regulator Co , 4501 Prospect Ave., Cleveland, and 14211 Ashwood Rd., Shaker Heights, O
MOREHOUSE, H. Preston (*M* 1933) Gen Air Cond. Repr (for mail) Public Service Electric & Gas Co , 80 Park Place, Newark, and 85 Halsted St., East Orange, N J
MOREHOUSE, J. Stanley (*M* 1938) Prof. Mech Engrg., Villanova College, Villanova, and (for mail) 102 Llandaff Rd , Upper Darby, Pa
MORGAN, Arthur S. (*M* 1938) Mgr , Fess Oil Burners of Canada, Ltd , 85 King St , W , and (for mail) 156 Glenmanow Drive, Toronto, Ont , Canada
MORGAN, Glenn C. (*M* 1911) Partner (for mail) Morgan-Gerrish Co , 307 Essex Bldg , 84 S. Tenth St , and 4308 Fremont Ave., S , Minneapolis, Minn
MORGAN, Robert C. (*M* 1915) Pres , Stewart A Jellett Co , 1200 Locust St., and (for mail) 314 W Seymour St , Philadelphia, Pa
MORGAN, Robert W. (*M* 1938) Research Engr., Air Cond & Commercial Refrigeration, Nash-Kelvinator Corp , and (for mail) 12739 Hubbell Ave , Detroit, Mich.
MORIARTY, John M. (*M* 1937) Owner (for mail) Consolidated Heating & Ventilating Co , 1709 West 8th St , and 2525 Burnside Ave , Los Angeles, Calif.
MORIN, A. R. (*A* 1938) Mgr , Refrigeration Dept , Macklanburg Brass & Copper Products, Inc , 111 N W 23rd, and (for mail) 925 S W. 28th, Oklahoma City, Okla
MORRIS, Arnold M. (*J* 1934) Sheet Metal Worker, Philadelphia Navy Yard, Sheet Metal Shop Bldg (Shop No 17) and (for mail) 3022 Baltz St., Philadelphia, Pa
MORRIS, C. Raymond (*M* 1921) Pres , Power & Heating Equipment Sales, Inc , 14 Burnett Place, Nutley, N J.
MORRIS, John A. (*J* 1936) Htg Dept , James Robertson Co , Ltd , 946 William St , and (for mail) 4134 Marlowe Ave , Montreal, P Q , Canada
MORRISON, Chester B. (*M* 1931) Mgr (for mail) York Shipley, Inc , 81 Jinkee Rd., and 347 Route Cohen, Shanghai, China.
MORRISON, Wayne L. (*A* 1938) Owner (for mail) Fair Plumbing & Heating Co., 1908 Broadway St , and 3422 16th St , Great Bend, Kan.
MORROW, J. DeWitt (*A* 1938) Secy -Treas and Gen Mgr. (for mail) The Warren Co , Inc., 614 Walker Ave , and 5503 La Branch, Houston, Tex.
MORSE, Clark T. (*M* 1913) Pres (for mail) American Blower Corp., 6000 Russell, and 8120 E. Jefferson, Detroit, Mich
MORSE, Floyd W. (*A* 1934) Asst Gen Sales Mgr (for mail) Chamberlin Metal Weather Strip Co , 52 Vanderbilt Ave , New York, and 132 Villa St , Mt Vernon, N Y
MORSE, Louis S., Jr. (*A* 1938, *J* 1936) Air Cond. Sales Engr (for mail) Westerlin & Campbell Co., 5924 Second Blvd , and 19480 Canterbury Rd., Detroit, Mich.
MORSE, Robert D. (*M* 1936) Mfrs Repr (for mail) R. D. Morse Agency, 1534-1st Ave , S , and 4316 E. 43rd St., Seattle, Wash.
MORTON, Charles H. (*A* 1931) Sales Repr , Kewanee Boiler Corp., Warren Webster & Co , 228 Ottawa Ave., N W., and (for mail) 1106 Sherman St , S E , Grand Rapids, Mich
MORTON, Harold S. (*M* 1931) Sales Engr., Sutherland Air Conditioning Corp , 385 Minnesota St., St. Paul, and (for mail) 4330 Wooddale Ave., Minneapolis, Minn.
MOSES, Walter B., Jr. (*S* 1936) Student, Tulane University (for mail) 425 S. Peters St , and 1516 Dufossat St , New Orleans, La.
MOSHER, Clarence H. (*A* 1919) C. H. Mosher Co., 423 Ashland Ave., Buffalo, N. Y.
MOSS, Edward (*M* 1920) Htg -Vtg. Engr. (for mail) New York Rapid Transit Corp , 385 Flatbush Ave Extension, Brooklyn, and 9053-204th St , Hollis, L I , N Y.
MOTZ, O. Wayne (*M* 1932) Consulting Engr , 234 Paramount Bldg , Cincinnati, and (for mail) 2524 Moundview Drive, Norwood, O.
MOULD, Delmar E. (*M* 1936) Mgr. (for mail) J. W. Mould & Son, Ltd , 10708 Jasper Ave , and 8619-108 A St , Edmonton, Alta , Canada.
MOULDER, Albert W.* (*M* 1917) Vice-Pres (for mail) Grinnell Co , Inc , 260 W. Exchange St , Providence, and Barrington, Providence R I
MUELLER, Harold C. (*M* 1936; *A* 1930) Mgr , Contract Div (for mail) Powers Regulator Co., 2720 Greenview Ave , Chicago, and 2720 Lawndale Ave , Evanston, Ill
MUELLER, Harold P. (*M* 1936) Pres (for mail) L. J. Mueller Furnace Co , 2005 W Oklahoma Ave , and 4721 N. Larkin St , Milwaukee, Wis.
MUELLER, John E. (*M* 1937) Mgr of Commercial Sales, West Penn Power Co , 14 Wood St , Pittsburgh, Pa
MUESSIG, James W. (*M* 1938) Sales Engr , Clarage Fan Co., Kalamazoo, Mich (for mail) 333 N. Michigan Ave , Chicago, and 442 Lodge Lane, Lombard, Ill
MUIRHEID, John G. (*J* 1937) Sales Engr , Baker Ice Machine Co , 2311 Hopedale Ave , Charlotte, N C
MULCEY, Paul A. (*J* 1938) Asst Dir , Anthracite Industries Laboratory, Primos, Delaware Co , and (for mail) 300 Springfield Ave , Aldan, Pa
MULLEN, Thomas J., Jr. (*J* 1935) Sales Engr., B. F Sturtevant Co , Hyde Park, Boston, Mass.
MUNIER, Leon L. (*M* 1919; *J* 1915) Pres. (for mail) Wolff & Munier, Inc , 222 E 41st St., New York, and 63 Columbia Ave , Hartsdale, N. Y.
MUNKELT, Frederick H. (*M* 1938) Vice-Pres , Consolidated Air Cond Div (for mail) W B Connor Engineering Corp , 114 East 32nd St., New York, and 1388 East 24th St., Brooklyn, N Y
MUNN, E. Fitz, (*M* 1935) Partner (for mail) Over & Munn, 903 McArthur Bldg , and 65 Berrydale Ave , Winnipeg, Man , Canada
MUNRO, George A. (*M* 1937) Member of Firm and Gen Mgr , Hugh F Munro & Sons, 2404 N Mascher St , and (for mail) 173 W Godfrey Ave , Philadelphia, Pa
MURDOCH, John P. (*M* 1937) Pres (for mail) John P Murdoch Co , S W Cor 30th & Oakford Sts , and 735 Beechwood Drive, Beechwood, Pa.
MURNIN, Edward A., Jr. (*A* 1937) Supt of Development & Assembly, Sarco Mfg Co , Clewell & Itaska Sts , and (for mail) 802 Broadway, Bethlehem, Pa
MURPHREE, Robert L. (*J* 1936) Promotional, Standard Sanitary Mfg. Co , 503 South 22nd St., and (for mail) 1509 North 21st Place, Birmingham, Ala.
MURPHY, Edward T.* (*M* 1915) Vice-Pres. (for mail) Carrier Corp., Merchandise Mart, and 230 E Delaware Place, Chicago, Ill
MURPHY, Howard C.* (*M* 1923) Vice-Pres (for mail) American Air Filter Co , Inc , 215 Central Ave , and Lightfoot Rd , Louisville, Ky
MURPHY, Joseph R. (*M* 1934, *A* 1925) Vice-Pres (for mail) Taco Heaters, Inc , 342 Madison Ave , New York, N. Y , and The Terrace, Riverside, Conn
MURPHY, William A. (*M* 1926) Gen Sales Mgr., Watts Regulator Co , 417 W Ohio St , and (for mail) 6214 N Richmond Ave , Chicago, Ill.
MURPHY, William W. (*M* 1930) Treas (for mail) W. W. Murphy Co , 424 Worthington St., and 25 Mansfield St., Springfield, Mass
MURRAY, Hayward G. S. (*J* 1936) Sales Engr. Refrig. & Air Cond. Div. (for mail) Canadian Comstock Co , Ltd , 1008 New Birks Bldg , and Apt. 6, 3727 de l'Oratoire, Montreal, P. Q., Canada.
MURRAY, John J. (*A* 1933) Salesman-Vice-Pres., Pierce Perry Co , 236 Congress St , Boston, and (for mail) 60 Commonwealth Park West, Newton Centre, Mass.

MURRAY, Thomas F. (*M* 1923) State Archt, and (for mail) 14 S. Lake Ave, Albany, N. Y

MUSGRAVE, Merrill N. (*A* 1935) Pres, Harrison Sales Co., 314-9th Ave, N., Seattle, Wash

MYERS, George W. F. (*M* 1930; *A* 1928; *J* 1923) Myers Engineering Equipment Co, 3736 W Pine Blvd, St Louis, and (for mail) 476 Pasadena Ave, Webster Groves, Mo.

MYLER, William M., Jr. (*M* 1937) Chief Engr, Space Htg Engrg. Dept (for mail) Surface Combustion Corp, 400 Dublin Ave, and 1120 Northwest Blvd, Columbus, O

MYTINGER, Kenneth L. (*M* 1936) Owner, Kenneth L. Mytinger, 105 Monmouth St, and (for mail) 119 East Bergen Place, Red Bank, N. J.

N

NACHMAN, George P. (*M* 1938) Secy & Treas (for mail) Spohn Heating & Ventilating Co, 1775 E 45th St, and 2870 Meadowbrook Blvd, Cleveland, O

NAROWETZ, Louis L., Jr. (*M* 1929; *A* 1912) Active Head, Narowetz Heating & Ventilating Co, 1711-1717 Maypole Ave, Chicago, Ill.

NASS, Arthur F. (*M* 1927) Pres (for mail) McGinness Smith & McGinness Co, 527 First Ave, and R. D No 8, Crafton P. O, Pittsburgh, Pa

NATHAN, Percival V. (*A* 1938) Chief Draftsman, Linde Canadian Ref Co, Ltd, 355 St Peter St, Montreal, P Q, Canada

NATKIN, Benjamin (*M* 1909, *J* 1907) Pres (for mail) Natkin & Co., 1800 Baltimore, and 5211 Rockhill Rd, Kansas City, Mo

NEARINGBURG, Arthur (*A* 1938) Sales Engr (for mail) Sheldons, Ltd, 1221 Bay St, and 130 Floyd Ave, Toronto, Ont, Canada

NEE, Raymond M. (*M* 1936) Head, Steam Service Section (for mail) Boston Edison Co, 39 Boylston St, Boston, and 10 Orkney Rd, Brookline, Mass

NEILER, Samuel G. (*Life Member*, *M* 1898) Owner (for mail) Neiler, Rich & Co, 431 S Dearborn St, Chicago, and 737 N Oak Park Ave, Oak Park, Ill

NELSON, Arthur W. (*A* 1936) Mgr, Brockton Oil Heat, Inc, 27 Legion Pkwy, Brockton, and (for mail) 12 Sylvan Rd, Sharon, Mass

NELSON, C. L. (*A* 1937; *J* 1929) Chief Air Cond Engr., Sears and Piou, 814 S Vandeventer, St Louis, and (for mail) 1731 Princeton Place, Richmond Heights, Mo

NELSON, D. W.* (*M* 1928) Assoc Prof of Mech. Engrg (for mail) University of Wisconsin, Mech. Engrg Bldg, and 3906 Council Crest, Madison, Wis

NELSON, Edwin L. (*A* 1936) Engrg Dept. (for mail) Union Ice Co, 1315 E Seventh St, and 4313 Victoria Ave, Los Angeles, Calif

NELSON, George C. (*M* 1923) Engr, Carstens Brothers, Ackley, Ia

NELSON, Harold M. (*M* 1937) Pres (for mail) H M. Nelson & Co, Inc, 1223 Connecticut Ave, and Rear 2208 Que St, N. W, Washington, D C.

NELSON, Herman W. (*M* 1909) Pres & Gen Mgr, Herman Nelson Corp, 1824 Third Ave, and (for mail) Le Claire Hotel, Apt. 1202, Moline, Ill.

NELSON, Richard H. (*A* 1933; *J* 1928) Secy-Treas, Herman Nelson Corp, 1824 Third Ave, and (for mail) 1303-30th St, Moline, Ill

NELSON, Roy O. (*M* 1938) Sales Engr (for mail) Fedders Mfg Co, 112 N. Green St, and 6419 N. Richmond St, Chicago, Ill

NESBITT, A. J.* (*M* 1921) Secy. and Treas (for mail) John J. Nesbitt, Inc, State Rd and Rhawn St, Philadelphia, and Rockfield Farm, Ambler, Pa.

NESBITT, J. J. (*Life Member*; *M* 1923) Pres (for mail) John J. Nesbitt, Inc., State Rd and Rhawn St., Philadelphia, and Rockfield Farm, Ambler, Pa

NESMITH, Oliver E. (*A* 1928) Engr., Williams Oil-O-Matic Heating Corp, Bell & Hanna, and (for mail) 107 Warner, Bloomington, Ill.

NESS, William H. C. (*M* 1931) Gen. Mgr. (for mail) Master Fan Corp, 1323 Channing St., and 215 N Kingsley Drive, Los Angeles, Calif.

NESSELL, Clarence W. (*M* 1937) Field Application Engr, Minneapolis-Honeywell Regulator Co, 1024 Third National Bldg., Dayton, O.

NESSI, André (*M* 1930) Ingr des Arts et Mfrs., Expert pres le Tribunal Civil de la Seine (for mail) 1 Avenue du President Wilson, Paris, XVI, France

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NEU, Henri J. E. (*M* 1933) Pres, Etablissements Neu, 47-49 Rue Fourier, Lille (Nord) France.

NEWCOMB, Lionel B. (*A* 1936, *J* 1933) Junior Engr, Philadelphia Electric Co, and (for mail) 6056 Walton Ave, Philadelphia, Pa

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NICHOLLS, Percy (*M* 1920) Supervising Engr, Fuels Section (for mail) Bureau of Mines, 4800 Forbes St, and 5251 Forbes St, Pittsburgh, Pa

NICKLE, Arthur J. (*A* 1936) Sales Engr. (for mail) Darling Brothers, Ltd, 140 Prince St., and 4356 Marcil Ave, Montreal, P. Q, Canada.

NIESSE, Joe H. (*M* 1937) Indiana Dist Mgr (for mail) Ilg Electric Ventilating Co, 836 Architects & Builders Bldg, and 5837 Winthrop Ave., Indianapolis, Ind

NIGHTINGALE, George F. (*A* 1931) Sales Mgr. (for mail) Tuttle & Bailey, Inc, P O Box 1313, and 290 Corbin Ave, New Britain, Conn.

NININGER, Christian H. (*A* 1938) Engr. and Dist Repr (for mail) H C Baker Co, Inc., 29 Franklin Rd, and 638 Northumberland Ave., Roanoke, Va

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NOBIS H. M. (*M* 1914) Chief Engr, Dresser-Nobis, Inc, 305-306 Caxton Bldg, Cleveland, and (for mail) 1827 Stanwood Rd, East Cleveland, O.

NOBLE, James P. (*A* 1937) Mgr, The Refrigeration & Htg Co, 21 N Limestone St, and (for mail) 113 E Cassilly St, Springfield, O

NOLAN, Ralph E. (*A* 1938) Owner, Ralph Nolan, Mfrs Repr (for mail) 429 Citizens & Southern Natl Bank Bldg, and 3384 Mathieson Rd., N E, Atlanta, Ga

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NOYES, Richard R. (*J* 1938) Sales Engr (for mail) Canadian Sirocco Co, Ltd, 630 Dorchester St, W, and 2010 Mansfield, Apt 12, Montreal, P Q, Canada

NUSBAUM, Lee* (*M* 1915) Owner (for mail) Pennsylvania Engineering Co, 1119-21 N Howard St, and 315 Carpenter Lane, Germantown, Philadelphia, Pa

NUTTING, H. G. D. (*M* 1938) Sales Engr, Air Cond & Refrigeration, Vilter Mfg Co, 604 Donovan Bldg, and (for mail) 1461 Calvert Ave, Detroit, Mich

NYE, L. Bert, Jr. (*J* 1936) Htg Engr, Washington Gas Light Co, 411 Tenth St, N W, Washington, D C, and (for mail) 309 Piedmont St, Arlington, Va

O

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OBERG, H C. (*A* 1933) Mgr, Engrg Dept, Crane Co of Minnesota, Fifth and Broadway, and (for mail) 1362 W Minnehaha St, St Paul, Minn

OBERSCHULTE, Richard H. (*J* 1938) Sales Engr (for mail) D J Randall Co, 404 Stormfeltz-Loveley Bldg, and 13999 Mark Twain Ave, Detroit, Mich

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OELGOETZ, J. F. (*M* 1938) Sole Owner (for mail) J F Oelgoetz Co, 3365 N High St, and 279 E North Broadway, Columbus, O

OFFEN, Ben (*M* 1928) Owner (for mail) B Offen & Co, 608 S Dearborn St, and 502 W Briar Place, Chicago, Ill

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O'GORMAN, J. S. (*A* 1934) Mgr Detroit Office (for mail) Johnson Service Co, 427 Brainard St, Detroit, and 147 Abbey Rd, Birmingham, Mich

OKE, William C. (*M* 1938, *J* 1934) Air Cond Engr, Weathermakers (Can) Ltd, 593 Adelaide St, W, and 460 Merton St, Toronto, Ont, Canada

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OLDES, Willard E. (*J* 1936) Piping and Incinerator Designer, Standard Oil Co, Elizabeth, N J, and (for mail) 610 West 204th St, Apt D-3, New York, N Y

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OLSON, Gilbert E. (*M* 1930) Chief Engr (for mail) Kelley Manufacturing Co, and 5411 Austin, Houston, Tex

OLSON, Milton J. (*J* 1937) Vice-Pres, Olson Bros, 2612 Leavenworth St, and (for mail) 5627 Williams St, Omaha, Nebr

OLSON, Robert G. (*M* 1923) Eastern Mgr (for mail) Hydraulic Coupling Div, American Blower Corp, 50 West 40th St, and 22 East 38th St, New York, N Y

OLVANY, William J. (*M* 1912) Pres (for mail) Wm J Olvany, Inc, 100 Charles St, New York, and 109-40-71st Rd, Forest Hills, N Y

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OPPERMAN, Everett F. (*J* 1935, *S* 1933) Estimator, Frederick Opperman, Railroad Ave, and (for mail) 7 West Elm St, Greenwich, Conn

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PEXTON, Frank S. (*A* 1936) Sales Engr (for mail) Kansas City Gas Co., 824 Grand, and 43 West 73rd Terrace, Kansas City, Mo

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PIETSCH, James A. (*M* 1936) Consulting Engr. (for mail) James A. Pietsch, Inc., 155 Prospect Ave, and 101 Cebra Ave., New Brighton, S. I., N. Y.

PIHLMAN, Arthur A. (*M* 1928) Service Engr. (for mail) Consolidated Edison Co. of New York, Inc., 4 Irving Place, New York, N. Y., and 98 Sherman Place, Jersey City, N. J.

PIKE, Wallace H. (*M* 1935) Design Engr., Newcomb David Co., 5779-81 Russell St., and (for mail) 4708 Buckingham Rd., Detroit, Mich

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PINTO, Chester B. (*A* 1937) Div Head, Montgomery Ward and Co, Plbg and Htg, 150-18 Jamaica Ave, Jamaica, and (for mail) 11 Buena Vista Ave, Lawrence, L. I, N Y.

PISTLER, Willard C. (*M* 1934) Mech Engr in Charge of Design, Carl J. Kiefer, Consulting Engr, 918 Schmidt Bldg., and (for mail) Orchard Lane & Crestview Ave, Pleasant Ridge, Cincinnati, O

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POGALIES, Louis H. (*M* 1931) Mech Engr, Wilbur Watson & Associates, 4614 Prospect Ave, and (for mail) 4102 Archwood Ave, Cleveland, O.

POHLE, K. F. (*A* 1930) Vice-Pres (for mail) W. F Hirschman Co, Inc, 202 E 44th St, New York, and 32-39-80th St, Jackson Heights, N. Y

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PRESDEE, Cliff W. (*A* 1926) S65 Div. Mgr , S R. Dresser Mfg Co , Bradford, Pa

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PRICE, Charles F. (*J* 1937) Htg and Air Cond Engr , The Knapp Supply Co , Ohio Ave and Dudley St., and (for mail) 1015 W Washington St., Muncie, Ind

PRICE, D. O. (*M* 1934) Htg & Air Cond Engr , General Steel Wares, Ltd , 199 River St , and (for mail) 131 St Germain Ave , Toronto, Ont , Canada.

PRICE, Ernest H. (*A* 1937; *J* 1934; *S* 1932) Engr. (Htg.), Wm Worton, Consulting Engineer, 504 Scott Block, and (for mail) 170 Harbison Ave , Winnipeg, Man , Canada.

PRIESTER, Gayle B. (*J* 1935; *S* 1934) Air Cond. Engr. (for mail) Carrier Corp., Merchandise Mart, and 5737 N. Kenmore, Chicago, Ill.

PRINCE, Raymond F. (*J* 1936) Htg. & Sales Engr., R B. Dunning & Co., Broad St., and (for mail) 27 McKinley St , Bangor, Me

PRITCHARD, William J. (*J* 1937) 3727-79th St., Apt. 21, Jackson Heights, L I., N. Y.

PROEBSTLE, Leonard (*J* 1938) Air Cond. Engr., Frigidaire Div., General Motors Sales Corp., 2446 University Ave , St. Paul, and (for mail) 800 S E. Superior St., Minneapolis, Minn

PROIE, John (*M* 1936) Gen. Mgr. (for mail) Proie Brothers, 856 W North Ave., and 101 Dilworth St , Pittsburgh, Pa.

PRUDDEN, O. D. (*J* 1938; *S* 1936) Engr , General Plastics, Inc , North Tonawanda, and (for mail) 37 Park Place, Lockport, N Y.

PRUDEN, Bradlee (*M* 1936) Engr , Barber-Colman Co , 150 Loomis St , and 1509 Grant Ave , Rockford, Ill.

PRYIBIL, Paul L. (*A* 1932) Partner, Hucker-Pryibil Co , 1700 Walnut St , and (for mail) 328 E Philellena St , Philadelphia, Pa

PRYKE, John K. M. (*A* 1937) Director, Lipscombe Air Conditioning Co., Dacre House, Dacre St , London, S W 1, and (for mail) 216 Clive Court, Maida Vale, London, W 9, England.

PRYOR, Frederick L. (*M* 1913) 5 Colt St , Paterson, N J.

PUGH, Daniel C. (*S* 1939) Student, Carnegie Institute of Technology, Pittsburgh, and (for mail) 267 Virginia Ave., Rochester, Pa.

PULLEN, Royal R. (*M* 1935) Chief Mech. Engr , Homestake Mining Co , and (for mail) 109 East Hill St , Lead, S D

PURCELL, Frederick C. (*M* 1926) Sales Engr. (for mail) Minneapolis-Honeywell Regulator Co , 415 Brainard St , and 4711 Second Blvd , Detroit, Mich

PURINTON, Dexter J. (*A* 1923) Vice-Pres , Mahoney-Troast Construction Co , 657 Main Ave , Passaic, N J , and (for mail) 148 E 53rd St , New York, N Y

PURSELL, H. E. (*M* 1919) Special Repr , Kewanee Boiler Corp , Kewanee, Ill

PUTNAM, Norman J. (*J* 1938) Sales Engr (for mail) Robischung-Kiesling, 4848 Main, Houston, Tex , and 902 S. Darlington, Tulsa, Okla.

Q

QUALL, Clarence O. (*A* 1937) Owner, Quall Plumbing & Heating Co , 65 Ninth St , and (for mail) 54 Pearl St , Clintonville, Wis

QUEER, Elmer R. (*M* 1933) Instructor in Engrg Research (for mail) Pennsylvania State College, Engrg Experiment Station, and 338 Arbor Way, State College, Pa

QUICK, Blair A. (*A* 1938) Sales Mgr , The Independent Register Co , 3747 E 93rd St , and (for mail) Fenway Hall, Cleveland, O

QUIRK, C. H. (*M* 1916; *J* 1915) Eastern Repr. (for mail) The Trane Co , 250 East 43rd St , New York, and 465 Front St., Hempstead, N Y.

R

RABE, Albert E. (*M* 1938) Pres (for mail) Condition-Aire Corp , 39 Sheridan Ave., and 97 Homestead Ave , Albany, N Y.

RABER, Benedict F. (*M* 1937) Prof of Mech Engrg (for mail) University of California, Room 114, Engrg Bldg , and 1124 Arch St., Berkeley, Calif.

RACHAL, John M. (*A* 1936; *J* 1930) Air Cond. Section, Andersen Meyer Co , Ltd., P O. Box 265, Shanghai, China

RAGATZ, Theodore E. (*A* 1937) Sales, Sidles Co , Airtemp Div , 14th St , and (for mail) 2222-14th St , Columbus, Nebr.

RAINE, John J. (*M* 1912) Vice-Pres (for mail) G. S Blodgett Co., 190 Bank St , Burlington, and Essex Junction, Vt.

RAINGER, Wallace F. (*A* 1930; *J* 1924) Jaros, Baum & Bolles, 415 Lexington Ave , and (for mail) 441 Hawthorne Ave , Yonkers, N. Y.

RAISLER, Robert K. (*A* 1933; *J* 1930) Treas. (for mail) Raisler Corp , 129 Amsterdam Ave., and 38 East 85th St , New York, N. Y.

RALPH, David S. (*J* 1938) Engr (for mail) General Roofing & Air Cond Co, Charleston, W Va, and 620 Kenilworth Ave, Dayton, O.

RAMSAY, James W. (*A* 1936) Sales Engr. (for mail) King & Shepherd, 50 Church St, New York, and 8415 Fourth Ave, Brooklyn, N Y.

RAND, Fred R. (*M* 1938) Htg Engr. & Sales Mgr, Enamel & Heating Products, Ltd, and (for mail) P. O Box 521, Sackville, N B, Canada

RANDALL, Robert D. (*A* 1930) Partner (for mail) D T Randall & Co, 7310 Woodward Ave, No 404, and 340 E Grand Blvd, Detroit, Mich.

RANDALL, W. Clifton* (*M* 1928) Chief Engr. (for mail) Detroit Steel Products Co, 2250 E Grand Blvd, Detroit, and 770 Shirley Drive, Birmingham, Mich

RANDOLPH, C. H. (*M* 1930, *A* 1928, *J* 1926) Air Cond Engr, Wisconsin Electric Power Co, 231 W Michigan St, and (for mail) 1614 E Royall Place, Milwaukee, Wis

RANK, Arthur I. (*A* 1936) Pres (for mail) Universal Insulation Co, 2429 South St, and 6308 Ross St, Philadelphia, Pa

RASMUSSEN, Robert P. (*M* 1931) Pres, Economy Equipment Co, 223 N Wolcott Ave, and (for mail) 1243 E 46th St, Chicago, Ill

RATHER, Max F. (*M* 1919) Mgr Eastern Territory, Johnson Service Co, 28 East 29th St, New York, N Y

RATHKE, Arthur C. (*J* 1937) Architectural Draftsman, Board of Education (Arch Dept) Toledo, and (for mail) 1025 Wayne St, Sandusky, O

RAVEN, Andrew H. (*M* 1938) Htg Engr, Wigman Co, 316 Perry St, and (for mail) 2119 George St, Sioux City, Ia

RAY, Lewis B. (*M* 1932) Mech Engr (for mail) Ray Engineering Co, Inc, 800 Broad St, Newark, and 151 Augusta St, Irvington, N J

RAYMER, W. F., Jr. (*A* 1936, *J* 1934) Sales Engr (for mail) American Blower Corp, 249 High St, Newark, and 266-4th Ave, East Orange, N J

RAYMOND, Fred I.* (*A* 1929) Owner (for mail) F I Raymond Co, 629 W Washington Blvd, Chicago, and 547 N Keystone Ave, River Forest, Ill

RAYNIS, Theodore (*J* 1934) Asst Naval Archt, Brooklyn Navy Yard, Design Section Building No 5, and (for mail) 8528-118th St, Richmond Hill, N Y

READER, Joseph T. (*A* 1938) Partner (for mail) Kerr Machinery Co, 608 Kerr Bldg, and 8120 E Jefferson Ave, Detroit, Mich

REAMER, William S., Jr. (*M* 1937) Vice-Pres, Treas, Reamer Industries, Inc, Seaboard Park, and (for mail) 2400 Blossom St, Columbia, S C

RECK, William E. (*M* 1927) Civil Engr (for mail) Reck Heating Co, Ltd, Copenhagen N, Esromgade 15, and Sundvej 16, Hellerup, Denmark

REDRUP, Will D (*M* 1936) Pres (for mail) Majestic Co, and 310 Randolph St, Huntington, Ind

REDSTONE, Arthur L. (*M* 1931) Research Engr, Proctor & Schwartz, 7th & Tabor Rd, and (for mail) 402 Park Towers, Kemble & Ogontz Ave, Philadelphia, Pa

REED, Irving G. (*A* 1937, *J* 1934) Asst Supt and Chief Engr, Grant Building, Inc, 420 Grant Bldg, and (for mail) 3227 Middletown Rd, Sheradan, Pittsburgh, Pa

REED, Van A., Jr (*M* 1930) Mech Engr (for mail) Federal Engineering Co, 239-4th Ave, Pittsburgh, and 114 Water St, Elizabeth, Pa

REED, Virgil C. (*M* 1938) Estimator and Engr (for mail) James H Pinkerton Co, 927 Howard St, and 1234 Second Ave, San Francisco, Calif

REED, William H., III (*A* 1938) Sales Engr (for mail) Dravo Corp, Carrier Dept, 302 Penn Ave, Pittsburgh, and 7815 Westmoreland, Swissvale, Pa

REGER, Henry P. (*M* 1934) Pres-Treas (for mail) H. P Reger & Co, 1501 East 72nd Place, and 6939 Bennett Ave., Chicago, Ill.

REID, Henry P. (*M* 1931, *A* 1927) Operating Engr. (for mail) Universal Atlas Cement Co, Rm 1527, 208 S LaSalle St, Chicago, and 3507 Oak Park Ave, Berwyn, Ill

REID, Herbert F. (*A* 1932) Reid-Graff Plbg Co, 1417 Peck St., Muskegon Heights, Mich

REIF, Allan F. (*M* 1937) Pres (for mail) Reif-Rexoil, Inc, 37-43 Carroll St, Buffalo, and 10 Livingston Pkwy, Snyder, N Y

REIF, Charles A. (*M* 1937) Vice-Pres, Treas (for mail) Reif-Rexoil, Inc, 37-41 Carroll St, and 115 Larchmont Rd, Buffalo, N Y

REIFSCHNEIDER, Jake (*A* 1938) Supt of Maintenance (for mail) Eppley Hotels Co, and 4637 Douglas St, Omaha, Nebr

REILLY, Bertram B. (*J* 1938) Engr, Trilling & Montague, 24th & Walnut St, and (for mail) 3259 Sansom St, Philadelphia, Pa

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REILLY, J. Harry (*M* 1931, *J* 1929) Sales Engr, American Radiator Co, 528 Ferry St, Newark, and (for mail) 14 Watson Ave, East Orange, N J.

REINKE, Alfred G. (*J* 1933) Secy, Gus Reinke Machinery & Tool Co, 63 Dickerson St, Newark, and (for mail) 321 Park Place, Irvington, N J

REINKE, Louis F. (*A* 1937) Owner (for mail) Reinke Sheet Metal Works, 534 S Fifth St, and 1535 W Walker St, Milwaukee, Wis

REINOLDI, Charles (*J* 1937) Cadet Gas Engr, Washington Gas Light Co, 411 Tenth St, Washington, D C, and (for mail) 3965 Wilsby Ave, Baltimore, Md

RENOUF, E. Prince (*M* 1933) Air Cond Supvr, Westinghouse Elec & Mfg Co, 1005 Insurance Bldg, and (for mail) 3431 Rankin St, Dallas, Tex

RENTE, Harry W (*M* 1931) Owner, Oil Burner Engr and Contractor, 114 Morris Ave, Buffalo, N Y.

RESS, Otto J. (*J* 1937) Gas Htg Engr, Iowa-Nebraska Light & Power Co, 1401 O St, and (for mail) 1900 South 17th, Lincoln, Nebr

RETTEW, Harvey F. (*M* 1929) Chief Engr, Board of Education, 21st St and the Parkway, and (for mail) Chelton Courts, Apt B-29, 17th & Chelton Ave, Philadelphia, Pa

REYNOLDS, Thurlow W. (*M* 1922) Asst Mech. Engr, New York World's Fair 1939, Inc, Flushing, L I, and (for mail) 100 Pinecrest Drive, Hastings-on-Hudson, N Y

REYNOLDS, W. V. (*A* 1928) Pres (for mail) Walter Reynolds, Inc, 861 Third Ave, and 444 East 52nd St, New York, N Y

RHINE, George R. (*A* 1938) Air Cond Engr (for mail) San Antonio Public Service Co, 201 N St Mary's St, and 155 Harrison Al Hts, San Antonio, Tex

RHOTON, W. R (*M* 1936) Pres, The W R. Rhoton Co, 1305 East 107th, Cleveland, and (for mail) 1728 Lee Rd, Cleveland Heights, O

RICE, Clarence J. (*A* 1923) Pres (for mail) Sterling Engineering Co, 3738 N Holton St, and Route 6, Box 374, Milwaukee, Wis

RICE, Robert B (*M* 1934) Prof of Experimental Engrg (for mail) North Carolina State College, State College Station, and 700 W Morgan St., Raleigh, N C

RICHARD, Edwin J. (*M* 1933) Owner, Edwin J Richard Equipment Co, 528 Chamber of Commerce Bldg, Cincinnati, O

RICHARDSON, Henry G (*M* 1934) Partner, Williams & Richardson, 204 Dooly Bldg, and (for mail) 1433 Harvard Ave, Salt Lake City, Utah

RICHARDSON, Robert D (*J* 1938) Htg Draftsman, Messrs Hope's Heating & Lighting, Ltd, Smethwick, and (for mail) 85 Silhill Hall Rd, Solihull, Birmingham, England

RICHFIELD, Nicholas H. (*M* 1937) Tech Head, Oil Burner Div, American Radiator Co, 40 West 40th St, New York, and (for mail) 173 N Tyson Ave, Floral Park, L I, N Y

RICHTMANN, W. M.* (*A* 1932, *J* 1926) Assoc Prof. of Mech Engrg (for mail) Texas College of Arts and Industries, and 600 W. Richards St., Kingsville, Tex.

RIES, Lester S. (*M* 1929) Supt. of Bldgs and Grounds (for mail) Oberlin College, 32 E College St, and 221 Woodland Ave, Oberlin, O

RIESMEYER, Edward H., Jr. (*A* 1936, *J* 1930) Engr, Schaffer Heating Co, 231 Water St, and (for mail) 4702 Stanton Ave, Pittsburgh, Pa

RIETZ, Elmer W.* (*M* 1923) Mgr Specialty Div (for mail) The Powers Regulator Co, 2720 Greenview Ave, Chicago, and 2250 S Sheridan Rd, Highland Park, Ill

RIGBY, Robert A. (*A* 1937) Sales Engr, Air Conditioning, and (for mail) 3325 North 48th Ave, Omaha, Nebr

RIST, Lawrence M. (*J* 1937) Sales Engr (for mail) Sidles Co, Airtemp Div, 502 South 19th St, and 3326 Harney, Omaha, Nebr

RITCHIE, A. G. (*M* 1933) Pres (for mail) John Ritchie, Ltd, 102 Adelaide St, E, and 41 Garfield Ave, Toronto, Ont, Canada

RITCHIE, E. J. (*M* 1923) Vice-Pres, Sales, Sarco Co, Inc, 183 Madison Ave, New York, and (for mail) 2 Grace Court, Brooklyn, N Y

RITCHIE, William (*M* 1909) 17 Van Reipen Ave, Jersey City, N J

RITTER, Arthur (*M* 1911) Dist Mgr (for mail) American Blower Corp, 50 West 40th St, New York, and 29 Edgemont Rd, Scarsdale, N Y

RIVARD, Melvin M (*M* 1935) Mgr, Rivard Sales Co, 4550 Main St, and (for mail) 1805 West 49th St Terrace, Kansas City, Mo

ROBB, Joseph E. (*A* 1936) Sales Engr, Minneapolis-Honeywell Regulator Co, 2753-4th Ave, S, Minneapolis, Minn, and (for mail) 6020 Maple Ave, Overland Park, Kan

ROBERTS, Henry L. (*M* 1916) Htg Engr and Contractor, Henry L Roberts (for mail) 228 North 16th St, Philadelphia, and 1014 Allston Rd, Brookline, Del Co (Upper Darby P O) Pa

ROBERTS, Henry P. (*A* 1936) Secy (for mail) Roberts-Hamilton Co, 713 South 3rd St, and 1901 James Ave, S, Minneapolis, Minn

ROBERTS, James R (*A* 1937, *J* 1934) Engrg Mgr (for mail) Sutherland Air Cond Corp, 15 N Eighth St, and 5705-11th Ave, S, Minneapolis, Minn

ROBERTSON, James A M (*A* 1936) Vice-Pres (for mail) The James Robertson Co, Ltd, 946 William St, Montreal, and 109 Sunnyside Ave, Westmount, P Q, Canada

ROBINSON, Arthur S (*M* 1936) Engrg Dept, E I duPont de Nemours Co, c/o Dye Works, Carney's Point, N J, and (for mail) 730 Ogden Ave, Swarthmore, Pa

ROBINSON, Donald M (*A* 1936) Sales Engr (for mail) Buffalo Forge Co, 820 Woodward Bldg., Washington, D C, and 16 Cedar St, Hyattsville, Md

ROBINSON, Earl T. (*A* 1938) Sales Repr, Crane Co, 201 Church St, and (for mail) 138 Wesley Ave, Buffalo, N Y

ROBINSON, Edgar R (*A* 1938) Development Engr, May Oil Burner Corp, and (for mail) 5434 Jonquil Ave, Baltimore, Md

ROBINSON, George L (*A* 1935) Draftsman and Designer, E I duPont de Nemours (for mail) 210 West 28th St, Apt 1, Wilmington, Del

ROBINSON, Jack A. (*J* 1936) Air Cond Engr, Australian Gas Light Co, Parker St, Sydney (for mail) Box 481 AA, G P O, Sydney, and 595 New South Head Rd, Rose Bay, N. S W, Australia

ROCHE, I. F (*A* 1936) Mgr (for mail) Fess Oil Burners of Canada, Ltd, 1405 Drummond St, and 4709 Cole St Catherine Rd, Montreal, P Q, Canada

ROCK, George A. (*M* 1937) Partner, Forbes & Co, 216 Southwest 12th Ave, and (for mail) 336 Southwest 13th Ave, Miami, Fla

ROCKWELL, Theodore F (*M* 1933, *J* 1932) Instructor (for mail) Carnegie Institute of Technology, Schenley Park, and Glenover Place, Aspinwall, Pittsburgh, Pa

RODEE, E. John (*M* 1936) Chief Engr (for mail) John B Pierce Foundation, 290 Congress Ave, New Haven, and 130 Bellevue Ave, West Haven, Conn

RODENHEISER, George B. (*M* 1933) Asst. Dir (for mail) David Ranken Jr. School of Mechanical Trades, 4431 Finney Ave, and 3639a Dover Place, St Louis, Mo

RODGERS, F. A. (*A* 1934) Branch Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 1916 Cedar Springs Rd, and 3421 St Johns Drive, Highland Park, Dallas, Tex

RODGERS, Joseph S (*A* 1937; *J* 1934) Engrg Draftsman, U S Government, Edgewood Arsenal, Edgewood, and (for mail) 1 Third Ave, Brooklyn Park, Md

RODMAN, R W. (*M* 1922) Supt of Plant Operation (for mail) Board of Education, City of New York, 500 Park Ave, and 175 West 73rd St, New York, N Y

ROEBUCK, William, Jr. (*M* 1917) Mfrs Agent (for mail) 220 Delaware Ave, and 154 Sanders Rd, Buffalo, N Y

ROGERS, Robert C (*A* 1937) House Htg Engr, Community Natural Gas Co, and (for mail) 4020 Dickason St, Dallas, Tex

ROHLIN, Karl W. (*M* 1930) Engr, Warren Webster & Co, 17th and Federal Sts, Camden, and (for mail) 4453 Terrace Ave, Merchantville, N J

ROLLAND, Sverre L. (*A* 1934) Design Engr (for mail) Oklahoma Gas & Electric Co, 321 N Harvey Ave, and 2131 Northwest 20th St, Oklahoma City, Okla

ROLLOSSON, John A. (*J* 1938) Pres (for mail) Rollosson-Keeland Co, 3714 Main St, and 1920 McGregor Ave, Houston, Tex

RONSICK, Edward H (*M* 1937) Industrial Engr (for mail) The St Louis County Gas Co, 231 W Lockwood, Webster Groves, and 7615 Marion Court, Maplewood, Mo

ROOS, Erik B. J (*J* 1935) 23, Alwiyah, Baghdad, Iraq

ROOT, Edwin B. (*M* 1936) Mgr Htg & Air Cond Dept, Nelson Co, 2604 Fourth Ave, Detroit, and (for mail) 964 Pierce St, Birmingham, Mich

ROSE, Harold J (*M* 1937) Sr Industrial Fellow (for mail) Mellon Institute, 4400 Fifth Ave, and 219 Lytton Ave, Pittsburgh, Pa

ROSE, Howard J (*M* 1934) Mgr of Engrg Sales, Suburban Air Conditioning Corp, 7 Depot Plaza, White Plains, and (for mail) 100 Siebrecht Place, New Rochelle, N Y

ROSE, Jerome C. (*M* 1937) Air Cond Engr, Buensod-Stacey Air Conditioning, Inc, 60 East 42nd St, New York, and (for mail) 8831 Fort Hamilton Pkwy, Brooklyn, N Y

ROSE, William H., Jr. (*J* 1938) Sales Engr (for mail) B F Sturtevant Co, 812 Michigan Theatre Bldg, and 5442 2nd Blvd, Detroit, Mich

ROSEBROUGH, J. Stoddard (*A* 1937) Sales Engr, L J Mueller Furance Co, 4246 Forest Park Blvd, and (for mail) 5917 Washington Ave, St Louis, Mo

ROSEBROUGH, Robert M. (*M* 1920) Branch Mgr (for mail) L J Mueller Furance Co, 4246 Forest Park Blvd, St Louis, and 204 S Maple Ave, Webster Groves, Mo

ROSELL, Axel F. (*M* 1935) Engr, A B Svenska Flaktfabriken, Kungsgatan 8, Stockholm, and (for mail) Kv Atlas 3, Lidingo, Sweden

ROSENBERG, Philip (*A* 1928) Secy-Treas, Universal Fixture Corp, 137 West 23rd St, and (for mail) 250 West 104th St, New York, N Y

ROSENBLATT, Arthur M. (*M* 1938) Partner (for mail) Rosenblatt & Hunt, P O. Box 828, 923 Virginia St, E, and 1250 Edgewood Drive, Charleston, W Va

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ROSS, Roderick (*M* 1937) Consulting Engr. (for mail) Nicholas Bldg., 37 Swanston St., Melbourne, C 1, P. O Box 1381 M, and 5 Burns St, Elwood, Melbourne, S 3, Australia

ROTH, Charles F. (*A* 1930) Pres (for mail) International Exposition Co., Grand Central Palace, and 141 East 36th St., New York, N Y

ROTH, Harold R. (*M* 1935) Sales Engr (for mail) Canadian Sirocco Co, Ltd., 57 Bloor St., W., and 5 Castleview, Toronto, Ont, Canada.

ROTHMANN, S. C. (*M* 1936) Industrial Hygiene Engr. (for mail) West Virginia Compensation Commission, State Capitol Bldg, and 2008½ Kanawha St, Charleston, W Va

ROTTMAYER, Samuel I. (*A* 1933; *J* 1928) Mech. Engr (for mail) Samuel R. Lewis, Consulting Engineer, 407 S Dearborn St, and 6041 S. St. Lawrence Ave, Chicago, Ill

ROWE, Irving E. (*A* 1936) Engr, Etie Sheet Metal Works, 1704 Houston Ave, and (for mail) 512 Bishop St, Houston, Tex

ROWE, William A.* (*M* 1921) (Council, 1929-1931) Mech Engr (for mail) The Trane Co, LaCrosse, Wis, and 718 Longfellow Ave, Detroit, Mich

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ROWLEY, Frank B.* (*M* 1918) (*Presidential Member*) (Pres, 1932; 1st Vice-Pres, 1931; 2nd Vice-Pres., 1930; Council, 1927-1933) Prof of Mech Engrg and Director of Experimental Engrg Lab, University of Minnesota, and (for mail) 4801 E Lake Harriet Blvd, Minneapolis, Minn

ROY, Arthur C. (*A* 1937) Metropolitan Sales Mgr, Hoffman Specialty Co, Inc, 500 Fifth Ave, New York, N Y, and (for mail) Box 507, Morristown, N J.

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ROYER, E. B. (*M* 1928) Designing Engr, Fosdick & Hilmer, Consulting Engrs, 1703 Union Trust Bldg, and (for mail) 6635 Iris Ave, Cincinnati, O

RUDD, Dann J. (*M* 1937) Htg & Vtg Engrg, N Y. City Board of Education, 49 Flatbush Ave Extension, Brooklyn, and (for mail) 369 Deer Park Ave, Babylon, N Y

RUDIO, H. M. (*M* 1921) Regional Engr, Airtemp Sales Corp, 631 Investment Bldg, Washington, D C, and (for mail) 2704 N Lexington St, Arlington, Va

RUEMMELE, Albert M. (*J* 1938) Sales Engr (for mail) L L Silkensen & Co, Inc, 401-23rd St, and 3628-0½, Galveston, Tex

RUFF, Adolph G. (*M* 1935) Supt of Power, U S Playing Card Co, Park Ave, Norwood, and (for mail) 3824 Woodford Rd, Cincinnati, O

RUFF, DeWitt C. (*M* 1922) Healy-Ruff Co, 765 Hampden Ave, St. Paul, Minn

RUFF, Herbert A. (*A* 1938) Pres & Treas, Herbert A Ruff, Inc, 39 Olcott Place, Station E, Buffalo, N. Y.

RUGART, Karl (*A* 1924) Owner Repr, Warren Webster & Co, and Kewanee Boiler Corp (for mail) 26 S. 20th St., Philadelphia, and 612 Bryn Mawr Ave, Narberth P O, Penn Valley, Pa

RUGGLES, Robert F. (*M* 1936; *A* 1927, *J* 1926) Dist. Mgr, Autovent Fan & Blower Co, 2 Rector St, New York, and (for mail) 15 Gregg Place, Randall Manor, S I, N Y

RUMMEL, Adolph J. (*M* 1937) Air Cond Engr (for mail) San Antonio Public Service Co, 201 N. St. Marys St., and 319 Thorman Place, San Antonio, Tex.

RUNKEL, Charles (*M* 1935) Pres (for mail) Acme Heating & Ventilating Co, Inc, 4224 S Lowe Ave, and 7921 S Hermitage Ave, Chicago, Ill.

RUPLE, Paul E. (*A* 1936) Chief Engr, Manhattan Mfg. Co, 210 S Lexington Ave, and (for mail) 170 Grand St, White Plains, N. Y.

RUSSELL, Edward A. (*M* 1936) Chief Engr, Vapor Car Heating Co., Inc., 1600 S. Kilbourn Ave., and (for mail) 8103 Dorchester Ave, Chicago, Ill.

RUSSELL, J. N. (*Life Member*; *M* 1899) Managing Dir., Rosser & Russell, Ltd (for mail) Romney House Marsham St., Westminster, and Fernacres Fulmer, Buckinghamshire, England.

RUSSELL, Trafton W. (*A* 1938) Sales Engr, L. J. Mueller Furnace Co., and (for mail) 3626 Harriet Ave., Minneapolis, Minn

RUSSELL, Wayne B. (*A* 1936) Engr., Russell Furnace Co., Inc., 601 N. Monroe, and (for mail) 1203 S. Cedar, Spokane, Wash.

RUSSELL, William A. (*M* 1921) (Council, 1934-1938) (for mail) Hoffman Specialty Co, Waterbury, Conn., and 628 West 57th Terrace, Kansas City, Mo.

RYAN, H. J. (*M* 1922) Sales Engr (for mail) The Trane Co, La Crosse, Wis., and 47 Harris Ave, Albany, N Y.

RYAN, James D. (*M* 1935) Supt and Engr, Whitney National Bank, St Charles and Gravier St, and (for mail) 215 N Rendon St, New Orleans, La

RYAN, Joseph B. (*M* 1938) Chief Engr & Mgr (for mail) Engineering Service Co, 113 Railway Exchange Bldg., and 3904 Tracy Ave, Kansas City, Mo.

RYAN, William F. (*J* 1933) Sales Engr, The Salina Supply Co, 302-304 North Santa Fe, and (for mail) 310 West Republic, Salina, Kan

RYBOLT, Arthur L. (*A* 1938) Gen Mgr, The Rybolt Heater Co, Miller St, Ashland, O

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SADLER, C. Boone (*M* 1928) Associate Civil Engr, 11th Naval Dist, and 4828 Orchard Ave, San Diego, Calif

SAHLMANN, Frank L. (*A* 1937) Member Transportation Dept, Electrical Mfg (for mail) General Electric Co, and 3926 Beech Ave, Erie, Pa

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SALINGER, Robert J. (*J* 1937) Engr (for mail) Reginald F Taylor, Consulting Engineer, 909 Bankers Mortgage Bldg, and 1654 Danville, Houston, Tex

SALLANDER, H. A. (*A* 1937) Branch Mgr, Sidles Co, Airtemp Div, and (for mail) 4514 Fontenelle Blvd, Omaha, Nebr

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SANDERS, Charles M., Jr. (*J* 1938) Air Cond Syndicate Repr (for mail) Westinghouse Electric & Mfg. Co, 20 N. Wacker Drive, and 5830 N Kenmore, Chicago, Ill

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SANFORD, Arthur L. (*M* 1915) Mech. Engr, C. H. Johnston, Archts and Engrs, 360 Robert St., and 1129 Portland Ave, St Paul, Minn
SANFORD, S. S.* (*M* 1930) Sales Engr (for mail) The Detroit Edison Co, 2000 Second Ave., and 1503 Seyburn Ave, Detroit, Mich
SAPP, Charles L. (*A* 1936) Sales Mgr, Farquhar Furnace Co, and (for mail) 620 North Walnut St., Wilmington, O
SAUNDERS, L. P. (*M* 1933) Chief Engr, Research Div (for mail) Harrison Radiator Div., General Motors Corp, and 507 Pine St, Lockport, N Y
SAUNIER, William C. (*A* 1938) Sales Engr. (for mail) Straus-Frank Co, 301 South Flores St, and 237 Pershing Ave, San Antonio, Tex
SAURWEIN, George K. (*M* 1938) Supt Engrg Dept (for mail) Harvard University, Lehman Hall, Cambridge, and 247 Slade St., Belmont, Mass.
SAWDON, W. M.* (*M* 1920) Prof of Experimental Engrg. (for mail) Cornell University, College of Engrg, and 1018 E State St, Ithaca, N. Y
SAWHILL, R. V. (*A* 1929) Exec Vice-Pres (for mail) Domestic Engineering Co, 110 East 42nd St., New York, and 115 Townsend Ave, Pelham Manor, N Y
SAWYER, J. Neal (*J* 1933) Asst Mgr Industrial Dept, Gustin-Bacon Mfg Co, 1416 West 12th St, and 33 W 58th St, Kansas City, Mo
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SCANLON, Edward S. (*A* 1934) Utilization Engr, Equitable Gas Co, 427 Liberty Ave, and (for mail) 3310 Regan Ave, Brentwood, Pittsburgh, Pa
SCARLETT, William J. (*M* 1936) Kooler-Keg Div, Novadel Agene Corp, Belleville, and (for mail) 91 Haddon Place, Montclair, N J
SCHAD, Clifford A. (*A* 1938, *J* 1937) Engr, United States Air Conditioning Corp, 2101 N. E Kennedy St, and (for mail) 4425-43rd Ave, S, Minneapolis, Minn
SCHAFER, Harry C. (*M* 1937) Sales Mgr (for mail) Iroquois Gas Corp, 45 Church St, Buffalo, and 197 Union St, Hamburg, N Y
SCHECHTER, Jack E. (*J* 1937) Sales Engr, York Ice Machinery Co, and (for mail) 1720 N Orange Grove Ave, Los Angeles, Calif
SCHECHTER, John P. (*J* 1935) House Htg Engr, Michigan Consolidated Gas Co, 415 Clifford, and (for mail) 1812 Burns Ave, Detroit, Mich
SCHEIDECKER, Daniel B. (*A* 1919) Secy (for mail) Hunter-Clark Ventilating System Co, 2800 Cottage Grove Ave, and 4626 N Kilbourn Ave, Chicago, Ill.
SCHERMER, Richard (*J* 1938; *S* 1936) Sales Engr, American Radiator Co, 40 West 40th St, New York, and (for mail) 40-67 Hampton St, Elmhurst, L I, N Y.
SCHERNBECK, Fred H. (*A* 1930) Salesman (for mail) William Bros Boiler & Mfg Co, Nicollet Island, and 5045 Portland Ave, Minneapolis, Minn
SCHERRER, Kenneth C. (*J* 1936) Engr, Natkin & Co, 114 E. Third, and (for mail) 12 East 12th St., Tulsa, Okla.
SCHERRER, Leon B. (*J* 1936) Sales Engr, Adams Furnace Co, 3346 Watson Rd., and (for mail) 6112 Simpson Terrace, St Louis, Mo
SCHLEMMER, Byron C. (*J* 1938; *S* 1937) Engr (for mail) Johnson Service Co, 300 Bond Bldg, Washington, D C., and 162 Manchester Ave, Wabash, Ind
SCHLICHTER, Charles F. (*M* 1938) Dist Mgr, Surface Combustion Corp, 2375 Dorr St, Toledo, O, and (for mail) 600 W St, N E, Washington, D. C
SCHLICHTING, Walter G. (*M* 1932) Mgr, Air Cond Dept., Clarage Fan Co, and (for mail) 1417 W Lovell St., Kalamazoo, Mich
SCHMID, John U. (*J* 1938) Sales Engr, Louis Allis Co, 427 E. Stewert, and (for mail) Milwaukee Athletic Club, Milwaukee, Wis
SCHMIDT, E. Georg (*M* 1938) Consulting Engr., Berlin-Friedenau, Kirchstrasse 29, Germany.
SCHMIDT, Harry (*M* 1937) Air Cond. Engr., Fedders Mfg Co., Inc., 57 Tonawanda St., and (for mail) 233 Norwalk Ave, Buffalo, N. Y.
SCHMIDT, Horace I. (*J* 1937) Branch Mgr. (for mail) Fedders Mfg Co., Inc., 209 S. Pearl St, and 2615 Laclede St., Dallas, Tex.
SCHMIDT, Karl, Jr. (*J* 1937) House Htg. Engr., Michigan Consolidated Gas Co., 415 Clifford, and (for mail) 14438 Mayfield Ave, Detroit, Mich
SCHMIELER, Joseph B. (*J* 1938) Research Engr, A.S H V.E Research Lab, U S. Bureau of Mines, 4800 Forbes St, and (for mail) 220 Zara St, Pittsburgh, 10, Pa
SCHMUTZ, Jean (*M* 1933) Administrateur-Delegue (for mail) Societe P. R S. M., 8 Passage de l'Atlas, and 18, rue Dufrenoy, Paris, XIX-0, France.
SCHNEIDER, Charles H. (*J* 1937) Sales Engr (for mail) Ilg Electric Ventilating Co, 1031 Commercial Trust Bldg, Philadelphia, Pa., and 222 Second Ave, Haddon Heights, N J.
SCHNELL, Robert H. (*A* 1938) Asst Mech. Engr (for mail) B. E Landes, Mech Engr., 915 Hubbell Bldg, and 1617-33rd St, Des Moines, Ia
SCHOEFFTER, Hans M. (*J* 1939) Sales Engr, Aerofin Corp, 11 West 42nd St., New York, N Y, and (for mail) 365 Westwood Ave, Old Tappan, N J
SCHOENHOFEN, Leo H., Jr. (*J* 1938) Draftsman, Oklahoma Gas & Electric Co, 3rd & Harvey (for mail) 605 N W 18th St, Oklahoma City, Okla
SCHOENIJAHN, Robert P. (*M* 1919) Consulting Engr (for mail) Industrial Trust Bldg, and 719 Nottingham Rd, Wilmington, Del.
SCHOEPFLIN, Paul H. (*M* 1920) Pres (for mail) Niagara Blower Co, 6 E 45th St, New York, and 91 Valley Rd, Larchmont, N Y.
SCHOLL, Howard O. (*J* 1938, *S* 1937) Engr, Colfax, Ill.
SCHREIBER, Herbert W. (*A* 1937) Sales (for mail) Johnson Service Co, 507 East Michigan St, and 3136 N. Eighth St, Milwaukee, Wis
SCHROTH, August H. (*M* 1937) 167 N. Grove St, P O Box 47, East Orange, N J.
SCHUETT, Donald F. (*J* 1938) Sales Engr, Curtis Refrigerating Machine Co, South Central Dist Repr (for mail) 8038 Bartow, Dallas, Tex.
SCHUETZ, Clyde C. (*A* 1936) Research Engr (for mail) United States Gypsum Co, 1253 Diversey Pkwy, and 2728 W. Agatite Ave, Chicago, Ill.
SCHULEIN, Ernst H. (*J* 1937) Consulting Engr, Birch & Krogboe, 31 V Farimagsgade, Copenhagen V., and (for mail) 3 Dalgas Blvd, Copenhagen F, Denmark.
SCHULER, William B. (*A* 1937) Sales, Taco Heaters, Inc., 342 Madison Ave, New York, N Y, and (for mail) 1536 East 69th St., Chicago, Ill
SCHULTZ, Albert W. (*M* 1936) Engr, Grinnell Co, Inc, 240-7th Ave, S, and (for mail) 5204 France Ave, S, Minneapolis, Minn
SCHULTZ, Stewart F. (*A* 1938) Sales Engr, Bruce Wigle Co, 9117 Hamilton Ave, and (for mail) 19312 Yacama, Detroit, Mich
SCHULZ, Edward L. (*J* 1937) Engr, Carrier Corp, S Geddes St, and (for mail) 106 Dewitt Rd, Syracuse, N Y
SCHULZ, Howard I. (*A* 1915) Crane Co, 1223 W Broad St, Richmond, Va.
SCHULZE, Ben H. (*M* 1921) Eastern Sales Mgr (for mail) Kewanee Boiler Corp, 37 W. 39th St, and 67 Park Ave, New York, N Y.
SCHURMAN, John A. (*M* 1936, *J* 1935) Mgr., Central Region Air Cond Div (for mail) York Ice Machinery Corp, 2700 Washington Ave, N W, Cleveland, and 14507 Delaware Ave, Lakewood, O

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SCHWARTZ, Norman E. (*M* 1938) Gen Mgr, Sidles Co, Airtemp Div, 502 S 19th, and (for mail) 4611 Davenport St, Omaha, Nebr
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SCOTT, Allison F. H. (*M* 1937) Asst to Pres (for mail) Anthracite Industries, Inc, 3207 Chrysler Bldg, New York, and 3 Hawthorne Rd, Bronxville, N Y
SCOTT, G. M. (*M* 1915) Pres (for mail) Child & Scott-Donohue, Inc, 153 East 38th St, New York, and 654 King St, Port Chester, N Y
SCRIBNER, Eugene D. (*A* 1933, *J* 1929) Engr, Alfol Insulation Co, Inc, 155 E 44th St, New York, N. Y, and (for mail) 314 First Ave, Westfield, N J
SEAL, Alfred T. (*M* 1938) Air Cond Engr. & Asst Purchasing Agent, Research Corp, Bound Brook, and (for mail) 60 Jennings Lane, North Plainfield, N. J
SEARLE, William J., Jr. (*M* 1937) Air Cond Engr., The Ballinger Co, 105 S. 12th St, Philadelphia, and (for mail) 207 Maple Ave, Narberth, Pa.
SEARS, Charles B. (*J* 1938) College Traveler, John Wiley & Sons, Inc., 440-4th Ave, New York, N Y.
SEEBER, R. R.* (*M* 1934) Head Mech Engrg Dept, Michigan College of Mining & Technology, Houghton, Mich
SEELBACH, Herman (*M* 1931) Pres (for mail) Equipment Sales, Inc, 800 Erie County Bank Bldg, Buffalo, and 31 Central Ave, Hamburg, N Y
SEELBACH, Herman, Jr. (*A* 1937) Sales Engr, Minneapolis-Honeywell Regulator Co, 45 Allen St, and (for mail) 280 Crescent Ave, Buffalo, N Y.
SEELERT, Edward H. (*A* 1935) Secy-Treas, McQuay Inc, 1600 N E Broadway, Minneapolis, Minn
SEELEY, L. E.* (*M* 1930) Asst Prof of Mech Engrg (for mail) Mason Laboratory, Yale University, and 130 Everit St, New Haven, Conn
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SEELIG, Lester (*M* 1925) Head Engrg Dept, Museum of Science & Industry, Jackson Park, and (for mail) 725 Irving Park Rd, Chicago, Ill
SEIDEL, Glenn E. (*J* 1937; *S* 1936) Tulane University, and (for mail) 1437 Audubon St, New Orleans, La
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SELIG, Ernest T., Jr. (*M* 1936) Registered Professional Engr and Industrial Fellow (for mail) Mellon Institute of Industrial Research, 4400 Fifth Ave, and 6609 Woodwell St, Squirrel Hill, Pittsburgh, Pa
SELLMAN, Nils T. (*M* 1922) Asst Vice-Pres, Consolidated Edison Co of New York, Inc, 4 Irving Place, New York, N Y
SELTZER, Paul A. (*J* 1938) Engr (for mail) Bryant Air Conditioning Corp., 830 N Broad St., Philadelphia, and 154 E. Marshall Rd, Lansdowne, Pa
SENIOR, R. L. (*M* 1925) Pres (for mail) R L Senior, Inc, 103 Park Ave, New York, and 10 Cherry Ave, New Rochelle, N Y
SESSLER, Robert E. (*S* 1938) 52 Fountain Rd, Arlington, Mass
SETTELMEYER, James T. (*J* 1937) Engr, Blocker Air Conditioning Corp., 825 Frelinghuysen Ave, Newark, and (for mail) 293 North Oraton Pkwy, East Orange, N J.
SEVERNS, W. H * (*M* 1933) Prof Mech Engrg (for mail) University of Illinois, and 609 Indiana, Urbana, Ill
SEYMOUR, James E. (*A* 1937) Owner, Lee & Seymour Warm Air Htg & Sheet Metal Works, 346 Russell St, and (for mail) 208 Lakewood Blvd, Madison, Wis
SHAER, I. Ernest (*A* 1934) Treas, Capitol Engineering Co, 71 Rogers St, Cambridge, and (for mail) 43 Ormond St, Dorchester, Mass
SHAFFER, Chester E. (*M* 1937) Research Engr, Koppers Co, Kearny, N J
SHANKLIN, Arthur P. (*M* 1929) Divisional Sales Mgr (for mail) Carrier Corp, and 618 Rugby Rd, Syracuse, N Y
SHANKLIN, John A (*M* 1928) Vice-Pres & Treas. (for mail) West Virginia Heating & Plumbing Co, 233 Hale St, and 1507 Quarrier St, Charleston, W Va
SHAPIRO, Charles A. (*J* 1938) Sales Engr, Johnson Service Co, 2853 N 12th St, and (for mail) 2041 N Wanamaker St, Philadelphia, Pa
SHAPIRO, Maurice M. (*J* 1937) Branch Engr, Sidles Co, Airtemp Div, 1228 "P" St, and (for mail) 2145 "N" St, Lincoln, Nebr.
SHARP, Henry C. (*M* 1935) Dist Repr (for mail) Herman Nelson Corp, and 1204 24th Ave, Moline, Ill
SHARP, John R. (*A* 1937) Supervisor, Htg and Air Cond Reprs, Bergen Div, Public Service Electric & Gas Co, 235 Main St, Hackensack, and (for mail) Maple St, Haworth, N J
SHAW, Burton E * (*A* 1936, *J* 1934) Research Chief (for mail) Penn Electric Switch Co, Goshen, and The Maples, Bristol, Ind
SHAW, Charles G. (*A* 1936) Engr and Prop, Shaw Engineering Co, Port Arthur, Tex
SHAW, Norman J. H. (*M* 1927, *J* 1925) Barnes & Jones, Inc, 128 Brookside Ave, Jamaica Plain, and (for mail) 37 Benjamin Rd, Arlington, Mass
SHAW, John A. (*M* 1938) General Elec Engr (for mail) Canadian Pacific Railway Co, Windsor Sta, Montreal, and 448 Lansdowne Ave, Westmount, P Q, Canada
SHAWLIN, Walter C. (*A* 1931) Mgr, Industrial Air Cond, Northwestern Ventilation Co, 2540 West Wells St, Milwaukee, Wis
SHEA, Michael B (*M* 1921) Sales Dept (for mail) American Radiator Co, 8019 Jos Campau, and 4080 Blaine, Detroit, Mich
SHEARS, Matthew W. (*M* 1922) Engr (for mail) C A Dunham Co, Ltd, 1523 Davenport Rd, and 39 Sylvan Ave, Toronto, Ont, Canada
SHEFFIELD, Raymond A (*M* 1937) Prop, Air Conditioning Engineering Co, Cambridge, and (for mail) 84 Governor Winthrop Ave, Somerville, Mass.
SHEFFLER, Morris (*M* 1921) Pres (for mail) Sheffler-Gross Co, 1000 Drexel Bldg, Philadelphia, and 419 Chapel Rd, Melrose Park, Montgomery Co, Pa
SHELDON, Nelson E. (*M* 1927) Dist Mgr (for mail) Carrier Corp, 302 S Geddes St, Syracuse, and 41 Lanark Crescent, Rochester, N Y
SHELDON, William D., Jr. (*A* 1936, *J* 1934) Chief Engr, Sheldon's, Ltd, and (for mail) Cedar St, Galt, Ont, Canada
SHELEY, Earle D (*M* 1937) Pres (for mail) Glanz & Killian Co, 1761 W Forest Ave, Detroit, and Box 243, Birmingham, Mich
SHELNEY, Thomas (*M* 1931) Pres, Pierce Blower Corp, 105 Brayton St, Buffalo, N Y
SHENK, Donald H. (*M* 1934) Assoc Prof Mech. Engrg (for mail) Clemson Agriculture College, Riggs Hall, Clemson, S C
SHEPARD, John deB. (*M* 1937, *J* 1929) Air Cond. Repr (for mail) Consolidated Gas Electric Light & Power Co, Lexington Bldg, Room 406, and Tudor Arms Apts, W University Pkwy, Baltimore, Md.

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SHORB, Will A. (*M* 1909) Dist Mgr, Decatur Pump Co, Decatur, Ill, and (for mail) 47 N Lime St, Apt 6, Lancaster, Pa

SHORE, David (*J* 1938) Research Engr (for mail) A S H V E Research Laboratory, 4800 Forbes St, and 969 Flemington St, Pittsburgh, Pa

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SHULTZ, Earle (*A* 1919) Commercial National Safe Deposit Co, 72 West Adams St, Chicago, Ill

SIDELL, Philip A. (*J* 1938, *S* 1937) Gale Prod. Div (for mail) Outboard Marine & Mfg Co, and 456 N Cherry St, Galesburg, Ill

SIEBS, Claude T. (*A* 1927) Service Systems Engr (for mail) Western Electric Co, Inc, 195 Broadway, New York, N Y, and 185 Kent Place Blvd, Summit, N J

SIEGEL, Daniel E. (*S* 1938) Student, Washington University (Evening School) St Louis, and (for mail) 7716 Wise Ave, Richmond Heights, Mo

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SIGMUND, R. W. (*M* 1932) Dist Mgr (for mail) B F Sturtevant Co, 913 Provident Bank Bldg, and 304 Oak St, Cincinnati, O

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SIMONS, Byron C. (*M* 1938) St Louis Branch Mgr (for mail) Minneapolis-Honeywell Regulator Co, 3033 Locust Blvd, St Louis, and 442 Woodlawn, Webster Groves, Mo

SIMONS, Edward W. (*M* 1938) Engr, Redwood Manufacturers Co, 1600 Hobart Bldg, and (for mail) 2418-30th Ave, San Francisco, Calif

SIMONSON, George M. (*M* 1937) Consulting Engr (for mail) 74 New Montgomery St, San Francisco, and 20 Loreta Ave, Piedmont, Calif.

SIMPSON, Arthur M.* (*A* 1935) Chief Engr., and Sales Mgr (for mail) Van Kannel Revolving Door Co, 101 Park Ave, New York, and 37-34-85th St, Jackson Heights, N Y

SIMPSON, W K. (*M* 1919) Vice-Pres (for mail) Hoffman Specialty Co, and 9 Sands St, Waterbury, Conn

SINGLETON, John H. (*A* 1937) Gen Mgr (for mail) Annas Heat & Cold, Inc, 13 N Perry, and 66 Franklin Blvd, Pontiac, Mich

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SLUSS, Alfred H. (*M* 1935) Prof. Mech and Industrial Engrg, University of Kansas, and (for mail) 827 Mississippi Ave, Lawrence, Kan

SMAK, Julius R. (*A* 1934) Supt. of Service Dept, Crane Co, South Ave, and (for mail) 3135 Park Ave, Bridgeport, Conn

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SMITH, Elmer G.* (*M* 1929) Assoc. Prof of Physics (for mail) Agricultural & Mechanical College of Texas, Department of Physics, College Station, Tex.

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SMITH, Gerald E. (*J* 1938) Sales Engr (for mail) Canadian Sirocco Co, Ltd, 57 Bloor St, W, and 52 Parkway Ave, Toronto, Ont, Canada

SMITH, Jared A. (*A* 1933) (for mail) Jared A Smith & Co, 481 S High St, and 102 N Parkview Ave, Bexley, Columbus, O

SMITH, J. Darrell (*M* 1933) Mech Engrg Dept Philadelphia & Reading Coal & Iron Co, and 317 North 19th St, Pottsville, Pa

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SMITH, Nelson J. (*M* 1938) Air Cond Design Engr, Frigidaire Div, 300 N Taylor St, Dayton, O

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SMITH, Reginald J. (*M* 1936) Mgr, Smith & Elston, 71 Third Ave, and (for mail) 112 S Maple St, Timmins, Ont, Canada

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SMITH, Wilbur F. (*M* 1920) Consulting Engr, W M Anderson Co, 600 Schuylkill Ave, and (for mail) 709 Braeburn Lane, Penn Valley, Narberth P O, Pa

SMITH, William D (*M* 1937, *A* 1935) Pres (for mail) Bryant-Smith, Inc, 2153 Prospect Ave, Cleveland, and 3265 Enderby Rd, Shaker Heights, O

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SMOOT, T H (*M* 1935) Mgr & Chief Engr, Fluid Heat Div, Anchor Post Fence Co, 6500 Eastern Ave, and (for mail) 2512 Talbot Rd, Baltimore, Md

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SNYDER, Joseph S (*A* 1925) Sales Repr, Detroit Lubricator Co, 1807 Elmwood Ave, Buffalo, and (for mail) 9 Knowlton Ave, Kenmore, N Y

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SONNEBORN, Charles (*M* 1930) R. D No 3, New Castle, Pa

SOPER, H. A. (*M* 1916) Vice-Pres (for mail) American Foundry & Furnace Co, Washington at McClun St, and 1122 E Monroe St, Bloomington, Ill.

SOULE, Lawrence C * (*M* 1908) Secy and Consulting Engr, Aerofin Corp, Syracuse, N Y, and (for mail) Essex Fells, N J

SOUTHMAYD, Richard T. (*J* 1936) Salesman (for mail) American Blower Corp, 1302 Swetland Bldg, Cleveland, and 65 Church St, Chagrin Falls, O

SPARK, William (*M* 1938) Mgr, Insulation Dept (for mail) Atlas Asbestos Co, Ltd, 110 McGill St, and 6171 Sherbrooke St, W, Montreal, P Q, Canada

SPARKS, James D (*A* 1937) Northwest Repr, Ilg Electric Ventilating Co, Chicago, Ill, and (for mail) 7331 W Green Lake Way, Seattle, Wash

SPECKMAN, Charles H (*M* 1918) Consulting Htg and Vtg Engr, Room 375, Philadelphia Bourse, Philadelphia, Pa

SPELLER, F. N * (*M* 1908) Advisory Engr (for mail) National Tube Co, P O Box 266, and 6411 Darlington Rd, Pittsburgh, Pa

SPENCE, Morton R (*J* 1934) Asst Purch Agt (for mail) Runelle & Spence Mfg Co, 445 N Fourth St, and 709 E Lexington Blvd, Milwaukee, Wis

SPENCE, Robert A (*J* 1937) Mech Engr, Boston Edison Co, 39 Boylston St, Boston, and (for mail) 37 Davis Rd, Belmont Mass

SPENCE, Robert T. (*A* 1935) 1556 South 60th St, West Allis, Wis

SPENCER, Dean (*A* 1937) Commercial Mgr (for mail) Brown Electric Division, Brown Supply Co, 120 E Grand and 3110 Northwest 23rd, Oklahoma City, Okla

SPENCER, J Boyd (*M* 1935) Pres & Treas (for mail) Spencer Air Conditioning Co, 515 Essex Bldg, and 2215 Newton Ave, S, Minneapolis, Minn

SPENCER, Roland M (*J* 1934) Branch Mgr (for mail) The Powers Regulator Co, 329 M & M Bldg, and 2715 Rosedale, Houston, Tex

SPENCER, Warner E (*A* 1938) Mgr, Buffalo Branch (for mail) National Radiator Corp, P O Box 136 No Tonawanda, and 212 Bidwell Pkwy, Buffalo, N Y

SPIELMANN, Gordon P. (*A* 1931, *J* 1923) Vice-Pres (for mail) Harrison-Spielmann Co, 480 Milwaukee Ave, Chicago, and 730 N Prospect Ave, Park Ridge, Ill

SPIELMANN, Harold J. (*M* 1933) Air Cond Engr, Vilter Mfg Co, 53 W Jackson Blvd, Chicago, and (for mail) 507 Elmore Ave, Park Ridge, Ill

SPITZLEY, Joseph H (*J* 1939) Junior Member (for mail) R L Spitzley Heating Co, 1200 W Fort St, Detroit, and 26 Renaud Rd, Grosse Pointe, Mich

SPITZLEY, Ray L (*M* 1920) Pres & Gen Mgr (for mail) R L Spitzley Heating Co, 1200 W Port St, Detroit, and 26 Renaud Rd, Grosse Pointe Shores, Mich

SPOERR, Frank F (*J* 1937) Carrier Engr, Air Cond Div, Hearnens So Warren & E Front St, Trenton, N J, and (for mail) 140-19 Queens Blvd, Jamaica, N Y

SPOFFORTH, Walter (*M* 1930) Chief of Mech Services, U S Penitentiary, McNeil Island, and (for mail) 615 N Ainsworth Ave, Tacoma, Wash

SPREKELMEYER, J. M. (*M* 1938) Mgr (for mail) General Engineering Corp, 1014 Jennings Ave, and 1912 Benhall Court, Fort Worth, Tex

SPRING, Claude L (*A* 1938) Htg Engr, Des Moines Stove Repair Co, 107 S W 2nd Ave, and (for mail) 3840 Columbia Ave, Des Moines, Ia

SPROULL, Howard E (*M* 1920) Div Sales Mgr (for mail) American Blower Co, 1005-6 American Bldg, and 3588 Raymar Drive, Cincinnati, O

SPURGEON, Joseph H. (*M* 1924) Salesman (for mail) Spurgeon Co, 5203 General Motors Bldg, and 17215 Pennington Drive, Detroit, Mich

SPURNEY, Felix E. (*A* 1938) Bldg Mgr, Federal Reserve Bldg, 20th & Constitution Aves, N W, Washington, D C, and (for mail) 10 Calvert Place, Kensington, Md

STACEY, Alfred E., Jr.* (*M* 1914) Vice-Pres, Buensod-Stacey Air Conditioning, Inc, 60 East 42nd St, New York, N Y, and (for mail) 35 Wotton Rd, Essex Fells, N J

STACK, Arthur E. (*A* 1935) Asst Mgr of Utilization Dept, Washington Gas Light Co of D C, 411 10th St, N W, Washington, D C, and (for mail) 911 Gist Ave, Silver Spring, Md

STACY, Loyd D. (*A* 1936) Sales Engr, Ilg Electric Ventilating Co, 182 N La Salle St, and (for mail) 7434 N Oakley Ave, Chicago, Ill

STACY, Stanley C (*M* 1931) Mech Engr (for mail) Board of Education, 13 S Fitzhugh St, and 531 Wellington Ave, Rochester, N Y

STAFFORD, J. Fuller (*A* 1938) Owner, J Fuller Stafford, Steam Specialties, 519 N Snelling Ave, St Paul, and 2925-33 Ave, S, Minneapolis, Minn

STAFFORD, Thomas D. (*A* 1937) Secy-Mgr, Alexander-Stafford Corp, 313-19 Allen St, N W, and (for mail) 954 Ogden Ave, Grand Rapids, Mich

STAHL, Walter A. (*M* 1938) Operating Engr, Real Estate Div, Marshall Field & Co, 222 Bank Dr, Chicago, and (for mail) 2504 Harrison St, Evanston, Ill

STALB, Joseph G. (*A* 1934) Mgr, Air Cond Div, Reynolds Corp, 19 Rector St, New York, and (for mail) 149 Columbia Heights, Brooklyn, N Y

STAMMER, Edward L. (*M* 1919) Supt Htg and Vtg, Board of Education, Ninth and Locust Sts, and (for mail) 4430 Tennessee Ave, St Louis, Mo

STANDRING, Ronald A. (*J* 1938) Htg Designer, Gurney Foundry Co, Ltd, 100 Principal St, St Laurent (near Montreal) and (for mail) 4838 Lafontaine St, Viauville, Montreal, P. Q, Canada

STANFIELD, Richard E. (*J* 1938) Industrial Engr (for mail) Nebraska Power Co, 723 Electric Bldg, and 5013 Cuming St, Omaha, Nebr

STANGER, R. B (*M* 1920) Prop (for mail) Robinson & Stanger, Empire Bldg, Pittsburgh, and Middle Rd, Glenshaw, Pa

STANGLAND, B F. (*Charter Member*) (2nd Vice-Pres, 1908, Board of Governors, 1905-1906-1909, Board of Mgrs, 1895-1899, Council, 1896-1897) Retired Htg & Vtg Cons & Constr Engr, Howard & Morse, New York, and (for mail) Kendall, N Y

STANLEY, Robert L. (*M* 1938) Engr, Payne Furnace & Supply Co, 338 N Foothill Rd, Beverly Hills, and (for mail) 2518 Dearborn Drive, Hollywood, Calif

STANNARD, J. M* (*Life Member*, *M* 1906) Pres and Treas (for mail) Stannard Power Equipment Co, 53 W Jackson Blvd, Chicago, and 1402 Elinor Place, Evanston, Ill

STANTON, Harold W. (*M* 1937) Commercial Engr (for mail) Iowa-Nebraska Light & Power Co, and 2807 Washington St, Lincoln, Nebr.

STARK, W. E.* (*M* 1926) Regional Mgr, The Bryant Heater Co, 17825 St Clair Ave, Cleveland, and (for mail) 1875 Rosemont Rd, East Cleveland, O

STARR, Lyndon (*A* 1938) Field Repr, Refrigeration & Air Cond Inst, and (for mail 1340 Midland Dr, University City, Mo

STASZESKY, Francis M. (*S* 1938) Student (for mail) Massachusetts Institute of Technology Dormitories, Cambridge, Mass, and 10 Roselawn, Wilmington, Del

STEEL, R. Justin (*A* 1938) Engr (for mail) Wilmington Auto Sales Co, 221 West Tenth St, Wilmington, and 19 Amstel Ave, Newark, Del

STEELE, John B. (*M* 1932) Chief Operating Engr, Winnipeg School Board, Ellen and William Ave, and (for mail) 184 Waterloo St, Riverheights, Winnipeg, Man, Canada

STEELE, Maurice G. (*M* 1929) Tech Advisor (for mail) Revere Copper & Brass, Inc, 1301 Wicomico St, Baltimore, and Hokeland, Havre de Grace, Md.

STEENKAMP, Willem (*S* 1938) Graduate Student (for mail) Senior House, Massachusetts Institute of Technology, Cambridge, Mass, and Box 12, Sheepmoor, Ermelo, Transvaal, Union of South Africa

STEFFNER, Edward F. (*A* 1937, *J* 1934) Htg. and Air Cond Engr, Henry Furnace & Foundry Co, 3471 East 49th St, Cleveland, and (for mail) 1427 East 133rd St, East Cleveland, O

STEGGALL, Howard B. (*A* 1934) Branch Mgr. (for mail) United States Radiator Corp, 941 Behan St, and 1166 Murray Hill Ave, Pittsburgh, Pa

STEHL, Howard V. (*A* 1936) Sales Engr (for mail) Campbell Metal Window Corp, P O. Box 148, and 10 Gwynnlake Drive, Woodlawn, Baltimore, Md

STEINER, Theodore J. (*A* 1938) Engr, Pomona Sheet Metal Works, and (for mail) 14807 Condon Ave, Lawndale, Calif

STEINHORST, T. F. (*M* 1919) Pres, Emil Steinhorst & Sons, Inc, 612-616 South St, and (for mail) 1664 Brinckerhoff Ave, Utica, N Y.

STEINKE, Bernard J. (*S* 1937) Htg and Vtg. Engr, Bernard H Steinke & Son, 1104 East 180th St, New York, N Y, and (for mail) 17 Westervelt Place, West Englewood, N J

STEINMETZ, C. W. A. (*M* 1934) Mgr of Newark Office (for mail) American Blower Corp, 249 High St, Newark, and 50 Oakwood Ave, Bogota, N J

STELLWAGEN, Frank G. (*A* 1937) Salesman, Fitzgibbons Boiler Co, Inc, 101 Park Ave, New York, and (for mail) 8637-77th, Woodhaven, N Y

STE-MARIE, Gaston P. (*M* 1930) Examiner Technician (for mail) Department of Labour, Provincial Government, 97 Notre-Dame St, East, and 4251 Marcil Ave, Apt 26, N D G, Montreal, P Q, Canada

STENCEL, R. Arthur (*M* 1938) Chief Engr, Canadian Ice Machine Co, Ltd, 65 Villiers St, and 45 Willowbank Blvd, Toronto, Ont, Canada

STENGEL, Frank J. (*A* 1935) Secy (for mail) R F Stengel & Son, 76-80 Rosehill Pl, and 321 Myrtle Ave, Irvington, N J

STEPHENSON, L. A. (*M* 1917) Mgr (for mail) The Powers Regulator Co, 409 East 13th St, and 801 West 57th Terrace, Kansas City, Mo

STERLING, James G, Jr. (*S* 1936) 1841 Wilton Rd, Cleveland Heights, O

STERMER, Clarence J. (*M* 1936) Engr, Crane Co, 836 S Michigan Ave, and (for mail) 7839 Clyde Ave, Chicago, Ill

STERNBERG, Edwin (*A* 1932, *J* 1931) Air Cond Engr, Armo Cooling & Ventilating Co, 30 West 15th St, and (for mail) 58 East 92nd St, New York, N Y

STERNE, Cecil M (*A* 1934) Chief Engr, Metropolitan Refining Co, Inc, 23-28-50th Ave, Long Island City, N Y

STERNER, Douglas S (*J* 1938, *A* 1936) Electrical Div, Barber-Colman Co, Rockford, Ill

STETSON, L R (*M* 1913) Engr, McMurrer Co, 303 Congress St, Boston, and (for mail) 35 Bradfield Ave, Roslindale, Mass

STEVENS, Alfred L. (*J* 1938) Refrigerating Engr, Mollenberg-Betz Machine Co, 20 Henry St, Buffalo, and (for mail) 45 Pasadena Place, Williamsville, N Y.

STEVENS, Harry L (*M* 1934, *A* 1927, *J* 1924) Secy-Treas (for mail) M M Stevens Co, 108 W Sherman St, and 7 West 22nd St, Hutchinson, Kan

STEVENS, Kenneth M. (*J* 1936) Sales Engr (for mail) The Powers Regulator Co, 409 East 13th, and 900 East Armour, Kansas City, Mo

STEVENS, William R. (*A* 1934) Partner, L E Stevens Co, 626 Broadway, Cincinnati, O, and (for mail) 30 Chalfonte Court, Fort Thomas, Ky

STEVENSON, Melvin J (*M* 1935) Dir of Sales Air Comfort Corp, 1307 S Michigan Ave, and (for mail) 5801 Dorchester Ave, Apt 2A, Chicago, Ill.

STEVENSON, W. W. (*M* 1928) Steam Htg Engr (for mail) Allegheny Co Steam Htg Co, 435 Sixth Ave, and 1125 Lancaster Ave, Pittsburgh, Pa
STEWART, Charles W (*M* 1919, *A* 1918) Asst Secy. (for mail) Hoffman Specialty Co, Waterbury National Bank Bldg, and 21 Yates Ave, Waterbury, Conn
STEWART, Duncan J.* (*M* 1936, *A* 1930) Mgr, Electrical Div (for mail) Barber-Colman Co, Drawer 99, and R R No 4, Rockford, Ill
STEWART, James P. (*J* 1937) Engr, Air Conditioning, 517 Brooks Bldg, and (for mail) 40 West Northampton St, Wilkes-Barre, Pa
STEWART, Wesley O. (*A* 1938) Branch Mgr (for mail) Johnson Service Co, 153 West Ave 34, Los Angeles, and 726 Central, Glendale, Calif
STIEGLER, Alvin J. (*A* 1938) Owner (for mail) Valley Sheet Metal Works, 315 Main St, and 319 Monroe St, Neenah, Wis
STILES, Gordon S. (*J* 1936) Sales Engr (for mail) Airtemp Div, Sidles Co, 118 Tenth St, and 206-11th St, Des Moines, Ia
STILL, Fred R.* (*M* 1904) (*Presidential Member*) (Pres, 1918, 2nd Vice-Pres, 1917, Council, 1916-1919) Vice-Pres in charge of Export (for mail) American Blower Corp, 50 West 40th St, New York, and 3457-82nd St, Jackson Heights, L I, N Y
STILLER, F. W. (*J* 1933) Estimator (for mail) F C Stiller & Co, 129 South Tenth St, and 138 West 49th St, Minneapolis, Minn
STINARD, R. L. (*J* 1934) Sales Engr, American Radiator Co, 40 West 40th St, New York, N Y, and (for mail) 1377 Boulevard East, West New York, N. J
STITES, Richard, Jr. (*J* 1937) Sales Engr, Buffalo Forge Co, 2051 W Lafayette, and 2170 E Jefferson, Detroit, Mich
STOCK, Charles S. (*M* 1936) Dist Repr, The Herman Nelson Corp, Rm 404-1108-16th St, N. W, Washington, D C, and (for mail) 6752 Fairfax Rd, Bethesda, Md
STOCKWELL, William R. (*M* 1903, *J* 1901) Gen Mgr, Mfg Div, Weil-McLain Co, Michigan City, Ind.
STOKES, Alvin D. (*M* 1936) Engr, York Ice Machinery Corp, 1238 N 44th St, Philadelphia, and (for mail) 4010 Ellendale Rd, Drexel Hill, Pa
STOKES, Arledge (*J* 1936) Air Cond Engr (for mail) Mehring and Hanson Co, Room 1006, Santa Fe Bldg, and 5211 Worth St, Dallas, Tex
STOLTZ, Guy C. (*A* 1938) Chief Engr, Air Conditioning Dept (for mail) Straus-Frank Co, and 315 Burr Rd, San Antonio, Tex
STORCH, Clemens A. (*M* 1930) Sales Engr, Johnson Service Co, 1355 Washington Blvd, Chicago, and (for mail) 331 Cumnor Rd, Kenilworth, Ill
STORMS, Robert M. (*M* 1936) Mech Engr, Consulting, Htg and Vtg (for mail) 816 West 5th St, Los Angeles, and 354 W Wilson, Glendale, Calif
STOTT, F. W. (*M* 1937) Sales Engr (for mail) C A Dunham Co, Ltd, 1139 Bay St, Toronto, and Palmer Ave, Oakville, Ont, Canada
STOTZ, Robert B. (*J* 1938) Sales Engr (for mail) Frigidaire Div, General Motors Sales Corp, 675 Greenwood Ave, and 1702 Harvard Rd, Atlanta, Ga
STRAUCH, Paul C. (*A* 1934) Sales Engr, The Henry Furnace & Foundry Co, S 18th & Merreman Sts, and (for mail) Cambridge Court Apts, 131 Edgewood Ave, Edgewood, Pittsburgh, Pa.
STREVELL, R. P. (*M* 1934) Pres & Treas (for mail) The Wm R Hogg Co, Inc, 900 Fourth Ave, Asbury Park, and Cor Victor Place & State Highway, Neptune, N J
STRICKLAND, Albert W. (*A* 1929) Htg & Vtg Engr, Big Timber, Mont
STROCK, Clifford (*M* 1937; *A* 1929) Associate Editor (for mail) Heating & Ventilating, 148 Lafayette St, New York, and 82-15 Britton Ave, Elmhurst, N Y
STROMGREN, Sven G. (*M* 1938) Consulting Engr, Asea Electric, Ltd, 4 Lyon Range, Calcutta, India
STROUSE, Sherman W. (*A* 1934) Sales Mgr, Cooney Refrigeration Co, and (for mail) 198 Livering Ave, Buffalo, N Y
STROUSE, Sidney B (*M* 1921) Consulting Engr (for mail) 500-529 Guarantee Trust Bldg, and 22 S Illinois Ave, Atlantic City, N J
STRUNIN, Jay (*J* 1933) Engr & Contractor, Strunin Plbg & Htg Co, Inc (for mail) 408 Second Ave, and 51 West 89th St, New York, N Y
STUART, Milton C* (*M* 1935) Prof of Mech Engrg (for mail) Lehigh University, Mech Engrg Dept, and 505 Norway Place, Bethlehem, Pa
STUBBS, William C. (*M* 1934) Associate Naval Archt, U S Government (for mail) Norfolk Navy Yard, and 37 Channing Ave, Portsmouth, Va
STURDY, Oswald C (*M* 1938) Sales Engr (for mail) Foster Wheeler, Ltd, 159 Bay St, and 24 Dorval Rd, Toronto, Ont, Canada
STURM, William (*J* 1937, *S* 1936) Engr, Spencer Cooling & Air Cond Co, 413 S Sixth St, and (for mail) 315-16th Ave, S E, Minneapolis, Minn
SUDDERTH, Leo (*J* 1936) Branch Mgr (for mail) Johnson Service Co, 311 Bona Allen Bldg, and 1115 Los Angeles Ave, N E, Atlanta, Ga
SULLIVAN, Charles J. (*A* 1938) Owner & Mgr, C J Sullivan, 5458 Baltimore Ave, and (for mail) 5470 Baltimore Ave, Philadelphia, Pa
SUMMERS, Clarence G (*A* 1938) Dist Mgr, Ilg Electric Ventilating Co, 2144 Madison Ave, Toledo, O
SUMMERS, Ernest T (*A* 1930) Pres (for mail) Summers-Darling & Co, 121 Smith St, and Ste 22 Newcastle Apts, Winnipeg, Man, Canada
SUNDELL, Samuel S. (*J* 1935, *S* 1933) Engr, Larx Co, Inc, 607 S. Fifth Ave, and (for mail) 3040 Longfellow Ave, Minneapolis, Minn
SUNDERLAND, Richard P. (*A* 1938) Pres (for mail) General Meters & Controls Co, 25 W Wacker Drive, Chicago, and 936 Judson Ave, Evanston, Ill
SUPPLE, Graeme B (*M* 1934) Dist Sales Engr (for mail) American Blower Corp, 625 Architects & Builders Bldg, and 6224 Park Ave, Indianapolis Ind
SUTCLIFFE, A. G. (*M* 1922, *A* 1918) Chief Engr, Ilg Electric Ventilating Co, 2850 N Crawford Ave, and (for mail) 4116 N St Louis Ave, Chicago, Ill
SUTFIN, George V (*A* 1937) Sales Engr (for mail) American Blower Corp, 1005-6 American Bldg, and 3270 Hildreth Ave, Cincinnati, O
SUTHERLAND, David L. (*A* 1934) Pres and Treas, Sutherland Air Conditioning Corp, 15 N Eighth St, and (for mail) 1815 S Colfax Ave, Minneapolis, Minn
SUTHERLAND, Floyd A. (*M* 1938) Design Engr, Electric Products Corp, 5624 Penn Ave, Pittsburgh, and (for mail) 403 Crest Ave, Charleroi, Pa
SUTTER, Edgar E (*A* 1936) Sales Engr, Mueller Brass Co, Port Huron, Mich, and (for mail) 6705 Sixth St, N W, Washington, D C
SWANEY, Carroll R. (*M* 1929, *J* 1921) Co-Partner, (for mail) Gilbert Howe Gleason & Co, 28 St Botolph St, Boston, and 43 Clyde St, Newtonville, Mass
SWANSON, Donald F. (*J* 1938) Test Engr, Seeger Refrigerator Co, 850 Arcade St, St Paul, and (for mail) 4316 Bloom Ave, Minneapolis, Minn
SWANSON, Earl C. (*A* 1935) Vice-Pres, Andersen Corp, Bayport, Minn
SWANSON, Nils W. (*A* 1936) Salesman, McDonnell & Miller, 400 N Michigan Ave, and (for mail) 2746 Morse Ave, Chicago, Ill
SWENSON, J. E. (*A* 1930) Mgr, House Htg Dept (for mail) Minneapolis Gas Light Co, 800 Hennepin Ave, and 4853-14th Ave, S, Minneapolis, Minn

SWINGLE, Wayne T (*A* 1938) Chief Engr, Jaden Mfg Co, Inc, and (for mail) Y M C A, Hastings, Nebr

SWISHER, Stephen G, Jr (*M* 1936, *A* 1934) Branch Mgr (for mail) The Trane Co, 1835 N 3rd St, and 1711 E Dean Rd, Milwaukee, Wis

SYDOW, Louis J (*M* 1936) Htg Engr, Federal Heating Co, 2735 N Union, and (for mail) 9456 Midland Ave, St Louis, Mo

SYMONDS, Edward S. (*M* 1939) Mgr, Abain Engineering, Ltd, 1 Devonshire Square, London, E C 2, and (for mail) 84 The Ridgeway, Chingford, Essex, England

SYSKA, Adolph G. (*M* 1933) Consulting Engr, Syska & Hennessy, 420 Lexington Ave, New York, N Y

SZEKELY, Ernest (*M* 1920) Vice-Pres and Gen Mgr (for mail) Bayley Blower Co, 1817 S 66th St, Milwaukee, and 6026 W Washington Blvd, Wauwatosa, Wis

SZOMBATHY, L R (*A* 1930) Pres (for mail) Ferguson Sheet Metal Works, 34 N Florissant Blvd, Ferguson, and 3125 Hawthorne Blvd, St Louis, Mo

T

TAGGART, Ralph C † (*M* 1912) 14 Lyon Ave, Menands, Albany, N Y

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TALLIANOS, Peter C (*A* 1938) Mgr, The Egyptian Wireless Co, 36 Nebi Daniel St, Alexandria, Egypt

TALLMADGE, Webster (*M* 1921) Pres (for mail) Webster Tallmadge & Co, Inc, 255 North 18th St, East Orange, and 7 Clairemont Place, Montclair, N J

TANGER, Othon C F (*A* 1937) Dir, N V Technische Handelsmaatschappij "Renova" Rembrandtlaan 34, Arnhem, Netherlands

TANKER, George E (*J* 1937, *S* 1936) Mech Engr, Weatherhead Co, 620 Frankfort Ave, Cleveland, and (for mail) 1303 Virginia Ave, Lakewood, O

TAPLEY, Mark S (*M* 1937) Htg, Vtg and Air Cond Engr, 3280 Holdrege St, Lincoln, Nebr

TARR, Harold M (*M* 1931) Htg & Vtg Engr, 21 Montague St, Arlington Heights, Mass

TASKER, Cyril* (*M* 1935) Research Fellow (for mail) Ontario Research Foundation, 43 Queens Park, and 737 Avenue Rd, Toronto, Ont, Canada

TAVERNA, F F (*M* 1928, *A* 1927, *J* 1921) Engr, Kaisler Corp, 129 Amsterdam Ave, New York, N Y, and (for mail) 406-12th St, Union City, N J

TAYLOR, Edward M (*A* 1934) Tech Mgr (for mail) Taylors, Ltd, 643 Colombo St, and 51 Totara Rd, Christchurch, New Zealand

TAYLOR, Fielding (*J* 1938) Sales Engr, Fairbanks, Morse & Co, 178 Atlantic Ave, Boston, and (for mail) 3 St James Ave, Haverhill, Mass

TAYLOR, Harold J (*M* 1937) Owner, Harold J Taylor, Htg & Vtg, 17514 Greenlawn Ave, Detroit, Mich

TAYLOR, Robert B. (*J* 1938) Sales Engr (for mail) Buffalo Forge Co, 702 Tower Petroleum Bldg, and 3000 Yale Blvd, Dallas, Tex

TAYLOR, R. F (*M* 1915) Consulting Engr (for mail) 911 Bankers Mortgage Bldg, and 1734 W Alabama, Houston, Tex

TAYLOR, Thomas E (*J* 1937) Consulting Mech Engr (for mail) 307 Postal Bldg, and 8424 S E 13th Ave, Portland, Ore

TAZE, D L. (*M* 1931) Mgr (for mail) American Blower Corp, 1302 Swetland Bldg, Cleveland, and 19412 Winslow Rd, Shaker Heights, O

TAZE, Edwin H. (*M* 1937) Branch Mgr (for mail) American Blower Corp, 620 Court Square Bldg, Baltimore, and Towson, Md

TEASDALE, Lawrence A. (*M* 1926) Engr, University Service Bureaus (for mail) Yale University, 20 Ashmun St, and 262 West Rock Ave, New Haven, Conn

TECKMYER, Fred C, Jr (*S* 1936) 1515 Woodward Ave, Lakewood, O

TEELING, Geo A (*M* 1930) Consulting Engr (for mail) 1 Columbia Place, Albany, and Box 81, Clarksville, N Y

TEMPLE, W. J. (*M* 1931) Engr, J A Temple Co, 108 Parkway, and (for mail) 1215 Reed St, Kalamazoo, Mich

TEMPLIN, Charles L (*M* 1921) Pres (for mail) Carrier Atlanta Corp, 348 Peachtree St, and (for mail) 781 Sherwood Rd, N E, Atlanta, Ga

TENKONOHY, Rudolph J. (*M* 1923) Vice-Pres, Airtherm Mfg Co, 1474 S Vandeventer Ave, and (for mail) 3650 Shaw Blvd, St Louis, Mo

TENNANT, Raymond J. J (*A* 1929) Engr (for mail) Pittsburgh Business Properties, Inc, 2237 Oliver Bldg, and 1215 Mississippi Ave, Pittsburgh, Pa

TENNEY, Dwight (*M* 1932) Pres & Chief Engr (for mail) Tenney Engineering, Inc, 46 Farrand St, Bloomfield, and 33 Summit Rd, Verona, N J

TERHUNE, Ralph D. (*A* 1936) Repr, American Gas Products Corp, 4th & Channing Sts, N E, Washington, D C, and (for mail) 4516 Highland Ave, Bethesda, Md

TERRILL, Mark (*A* 1938) Sales Engr, The Philip Carey Co, and (for mail) 656 Fuller, S E, Grand Rapids, Mich

TERRY, Matson C. (*M* 1936) Mgr, Production & Engrg (for mail) Standard Air Conditioning, Inc, 2nd & Beechwood, New Rochelle, and Apt 7K, Hawthorne Gardens Apts, Mamaroneck, N Y

ter WEEME, Albert (*A* 1938) Sales Engr, N V Radiatoren, Singel 206-208, Amsterdam, Holland

TEVES, Hendrik L (*A* 1938) General Mgr, N V v h Becht & Dyserinck, Amsterdam-Noord, Corn Douwesweg 1 (for mail) Huize "Vechtvliet", Breukelen, Netherlands

THEOBALD, Art (*A* 1937) Research Engr (for mail) Payne Furnace & Supply Co, Inc, 336 N Foothill Rd, Beverly Hills, and 116½ S Kings Rd, Los Angeles, Calif

THEORELL, Axel T (*M* 1939) Consulting Engr, Theorells Ingeniorsbyra, and (for mail) Lokattsvagen 11, Appelviken, Stockholm, Sweden

THEORELL, Hugo G T * (*Life Member*, *M* 1902) Consulting Engr, Hugo Theorells Ingeniorsbyra, Skoldungagatan 4, Stockholm, Sweden

THINN, C A * (*M* 1921) C A Dunham Co, 450 East Ohio St, Chicago, Ill

THOM, George B (*M* 1937) Asst Prof Mech Engrg, Swarthmore College, Swarthmore, Pa

THOMAN, Estell O. (*A* 1938) Mgr of Air Cond, Detroit Branch, Burge Ice Machine Co, 3208 Gratiot Ave, Detroit, Mich

THOMAS, Arthur E. (*J* 1938) Contracts Mgr, Young, Austin & Young, Ltd, H35/36 Exchange Bldgs, Liverpool, 2, and (for mail) "Arlen" 56, Thingwall Rd, Wavertree, Liverpool, 15, England

THOMAS, Bernard A. (*A* 1927, *J* 1923) **Mgr**, Htg & Engrg Dept Crane Co, 1405 Twiggs St, and (for mail) 405 E Idlewild Ave, Tampa, Fla.

THOMAS, Glegge (*M* 1923) Mgr, Wash Office (for mail) Clarage Fan Co, 723 Albee Bldg, Washington, D C, and 7 W Leland St, Chevy Chase, Md

THOMAS, L. G. Lee (*M* 1934) Vice-Pres (for mail) Economy Pumps, Inc, Weller & Zimmerman Aves, and Anthony Wayne Hotel, Hamilton, O

THOMAS, Melvern F. (*M* 1909) Consulting Engr (for mail) Thomas & Wardell, 24 Bloor St, W, and 74 Rivercrest Rd, Toronto, Ont, Canada

THOMAS, Norman A. (*M* 1928) Pres, Thomas Heating Co, 142 South 14th St, La Crosse, Wis

THOMAS, Ralph C. (*A* 1938) Vice-Pres & Mgr, Bonair Conditioning & Refrigerating Co, Inc, West Norfolk, and (for mail) 819 Westover Ave, Norfolk, Va

THOMAS, Richard H (*Life Member*, *M* 1920) Pres, Economy Pumping Machinery Co, 3431 W 48th Pl, Chicago, Ill

THOMMEN, Adolph A. (*A* 1929) Air Cond Fitting Mfr, John W Thomson Co, 1437 W 103rd St, and (for mail) 3400 W 61st Place, Chicago, Ill

THOMPSON, Edward B. (*A* 1938) Htg Engr, Cincinnati Gas & Elec. Co, 4th & Main Sts, and (for mail) 1198 Coronado Ave, Cincinnati, O

THOMPSON, Frank (*M* 1935) Chief Engr, Vulcan Iron Works, Ltd, Pt Douglas Ave, and (for mail) 543 Newman St, Winnipeg, Man, Canada

THOMPSON, Nelson S.* (*Life Member*, M 1917, *J* 1897) 1615 Hobart St, N W, Washington, D C

THOMSEN, Nis B. (*M* 1938) Consulting Engr (for mail) Baymond Corp, Victory Bldg, 21st Floor, Toronto, Ont, Canada.

THOMSON, Thomas N.* (*Life Member*, *M* 1899) Consulting Engr, Plbg and Htg, 37 Irwin Place, Huntington, L I, N Y

THORNBURG, Harold A. (*M* 1932, *J* 1929) c/o N V Industrieele, Mi j Gebr, Van Swaay, Societeitstraat 16, Soerabaja Java, Dutch East Indies

THORNTON, Thaddeus L. (*M* 1937) Maintenance Engr, Prudential Insurance, 96 Barclay St, Newark, and (for mail) 37 Perry St, Belleville, N J

THORNTON, W. B* (*M* 1931) Engr, Carrier Corp, Merchandise Mart, and 8314 Indiana Ave, Chicago, Ill

THRUSH, Homer A. (*M* 1918) Pres, H A Thrush & Co, 21 East Riverside Drive, Peru, Ind

THULMAN, Robert K. (*M* 1938) Mech Engr, Federal Housing Administration, Vermont and K Sts, N W, Washington, D C, and (for mail) 6505 Ridgewood Ave, Chevy Chase, Md

THUNEY, F. M. (*J* 1936) Application Engr (for mail) Wm E Kingswell, Inc, 3707 Georgia Ave, N W, and 4474 Conduit Rd, N W, Washington, D C

TIDMARSH, Patrick M (*M* 1938) Vice-Pres & Gen Mgr, Tidmarsh Engr Co, P O Box 2425, Tucson, Ariz

TILLER, Louin (*A* 1935, *S* 1933) Air Cond Engr, Oklahoma Gas & Electric Co, 321 N Harvey, and (for mail) 2712 Northwest 15th St, Oklahoma City, Okla

TILTZ, Bernard E (*M* 1930) Pres (for mail) Tiltz Air Conditioning Corp, 230 Park Ave, New York, and 24 Barnum Rd, Larchmont, N Y

TIMMINS, W W (*M* 1937) Dist Mgr (for mail) Canadian Powers Regulator Co, Ltd, 344 University Tower Bldg, Montreal, and 351 Brock Ave, North, Montreal West, P Q, Canada

TIMMIS, Pierce (*M* 1920) Service Equip Engr (for mail) United Engineers & Constructors, Inc, 1401 Arch St, Philadelphia, and 202 Midland Ave, Wayne, Pa

TIMMIS, W. W. (*M* 1933, *A* 1925) Mgr, Air Conditioning Systems and Control Div, American Radiator Co, 40 West 40th St, New York, and Pleasantville, N Y

TJERSLAND, Alf (*M* 1916, *J* 1906) E Sunde & Co, Ltd, Oslo, Norway

TOBIN, George J. (*Life Member*, *M* 1905) Owner and Prop (for mail) Geo J Tobin, 187-191 North Ave, and 510 Grant Ave, Plainfield, N J

TOBIN, John F. (*A* 1934) Salesman (for mail) American Blower Corp, 228 N La Salle St, and 11256 S Artesian Ave, Chicago, Ill

TODD, Meryl L (*J* 1936) Meryl L Todd, Mech Engr (for mail) 901 Waterloo Bldg, and 1119 Vine St, Waterloo, Ia

TODD, Stanton W. (*J* 1935) Sales Repr, American Radiator Co, 8019 Jos Campau St, Detroit, and (for mail) 309 Paris, S E, Grand Rapids, Mich

TOLHURST, George C. (*M* 1936) in Charge of Engr Dept, Gurney Massey Co, Ltd, 36 Principal St, and (for mail) 142 Blvd St Germain, St Laurent (near Montreal) P. Q, Canada

TONRY, Robert C. (*M* 1936) Mgr (for mail) Wiedbusch Plumbing & Heating Co, 511 First St, and 217 Fairmont Ave, Fairmont, W Va

TOONDER, C L. (*M* 1933) Air Cond Sales Engr, Norge Div, Borg-Warner Corp, 670 E Woodbridge, and (for mail) 13391 Marlowe, Detroit, Mich

TORNQUIST, Earl L. (*A* 1934) Supv Distribution Operation (for mail) Public Service Co. of Northern Illinois, 72 West Adams St, Chicago, and 465 Parkside Ave, Elmhurst, Ill

TOROK, Elmer (*M* 1936) Supt of Power (for mail) North American Rayon Corp, and 203 West G St, Elizabethton, Tenn

TORR, Thomas W (*M* 1933) Chief Engr, Rudy Furnace Co, and (for mail) P O Box 73, Dowagiac, Mich

TORRANCE, Henry (*M* 1933) Chairman, Carbondale New York Co, 175 Christopher St, and (for mail) 112 East 17th St, New York, N Y

TOUTON, R. D. (*M* 1933) Tech Director (for mail) Bayuk Cigars, Inc, Ninth and Columbia Ave, Philadelphia, and 19 Lodges Lane, Cynwyd, Pa

TOWER, Elwood S (*M* 1930) Engr (for mail) 213 Investment Bldg, and 5516 Woodmont St, Pittsburgh, Pa

TOWLE, Philip H (*J* 1938) Air Cond Engr (for mail) General Air Conditioning Co, 1313 Jay St, and 514 23rd St, Sacramento, Calif

TOWNE, Charles O (*J* 1938) Air Cond Engr, Apartment 306, Kingston Courts, LaCrosse, Wis

TRACY, William E. (*J* 1937) Dist Mgr (for mail) B F Sturtevant Co, 237 Grand Exchange Bldg, and 5016 Cass St, Omaha, Nebr

TRAMBAUER, Charles W (*J* 1936) Sales Engr, Hoffman Specialty Co, 500 Fifth Ave, New York, N Y, and (for mail) 77 Deoon St, North Arlington, N J

TRANE, Reuben N.* (*M* 1915) Pres (for mail) The Trane Co, and 126 South 15th St, La Crosse, Wis

TRAUGOTT, Mortimer (*A* 1930) Gen Mgr (for mail) Bryant Air Conditioning Corp, 830 N Broad St, Philadelphia, and 721 Meeting House Rd, Elkins Park, Pa

TRAWICK, Jack G. (*M* 1937) Dist Repr (for mail) Minneapolis-Honeywell Regulator Co, 1316 Comer Bldg, and 315 Altamont Apts, Birmingham, Ala

TRAYNOR, Harry S (*J* 1937) Engr, Carrier Corp, S Geddes St, and (for mail) 442 Salt Springs Rd, Syracuse, N Y

TREADWAY, Quentin (*A* 1936, *J* 1932) Dist Sales Mgr (for mail) Clarage Fan Co, 410 Reynolds Arcade, and 826 Winona Blvd, Rochester, N Y

TRELEAVEN, Herbert M (*J* 1938) Junior Engr, Weathermakers (Canada) Ltd, 593 Adelaide St, W, and (for mail) 21 Glenfern Ave, Toronto, Ont, Canada

TRENNER, Kelvin (*A* 1938) Mgr, Engrg Dept, Staats Coal Co, Malvern 724 W Marshall St, Norristown, and (for mail) 4819 N Mascher St, Philadelphia, Pa

TRIGGS, Fred E. (*M* 1938) Sales Engr & Mfrs Agent (for mail) P O Box 1, H P Sta, and 3901-2nd St, Des Moines, Ia

TROSTEL, Otto A. (*M* 1935) Engr (for mail) Kern Engineering Co, Inc, 161 W Wisconsin Ave, and 3155 N 7th St, Milwaukee, Wis.

TROUP, John D. (*M* 1938) Managing Dir (for mail) John D. Troup, Ltd, 90 High Holborn, London, W. C 1, and 48 Plough Lane, Purley, Surrey, England

TRUITT, G. Scott (*J* 1937; *S* 1936) Production Dept, Bastian-Morley Co, Inc, and (for mail) 907 Indiana Ave, LaPorte, Ind

TRUMBO, Silas M. (*A* 1926) Sales (for mail) Buffalo Forge Co, 20 N Wacker Dr, Chicago, and 921 Franklin St, Downers Grove, Ill

TRUMP, Charles C. (*M* 1934) Pres (for mail) James Spear Stove & Heating Co, 1823 Market St, Philadelphia, and 503 Baird Rd, Merion Station, Pa

TRZOS, Otto A. (*J* 1938) Industrial Gas Engr, Consumers Power Co, 26 W Lawrence St, Pontiac, and (for mail) Box 103, Keego Harbor, Mich

TUCKER, Frank N (*M* 1926) Field Engr, Ilg Electric Ventilating Co, 15 Park Row, New York, and (for mail) 239 Whaley St, Freeport, L I, N Y

TUCKER, Leonard A. (*M* 1935) Service Mgr, J J Pocock, Inc, 31st & Jefferson Sts, Philadelphia, and (for mail) 518 Monroe Ave, Ardsley, Pa

TUCKER, Thomas T (*M* 1938, *A* 1936) Chief Engr, Armor Insulating Co, 260 Peachtree St, and (for mail) 3619 Old Ivy Rd, N E, Atlanta, Ga

TUCKERMAN, George E (*M* 1932) Mgr (for mail) Anderson-York Co, 600 Schuylkill Ave, Philadelphia, and 502 Rodman Ave, Jenkintown, Pa

TUPPER, George B (*A* 1930) Sales Mgr, General Regulator Corp, 2608 Arthington St, and (for mail) 5921 Kenmore Ave, Chicago, Ill

TURK, Leonard G (*S* 1938) Student (for mail) Carnegie Institute of Technology, 4903 Forbes St, Pittsburgh, Pa, and 346 Avenue B, Rochester, N Y

TURLAND, Charles H (*M* 1934, *A* 1930) Sales Engr (for mail) R E Johnston Co, Ltd, 1070 Homer St, and 4579 W 1st Ave, Vancouver, B C, Canada

TURNER, George G (*A* 1934) Western Repr (for mail) Heating & Ventilating, 228 N LaSalle St, Chicago, and 827 Hinman Ave, Evanston, Ill

TURNER, Harry S, Jr (*J* 1937, *S* 1936) Asst Engr, Dallas Power & Light Co, 1001 Dallas Power & Light Bldg, and (for mail) 4950 Gaston, Dallas, Tex

TURNER, John (*M* 1930) Engr (for mail) Capitol Engineering Co, Potter and Binney Sts, Cambridge, Mass, and Contoocook, N H

TURNER, Prescott K (*A* 1937, *J* 1935) Engr, 10 Windemere Rd, Worchester, Mass

TURNO, W G W. (*M* 1917, *A* 1912) Secy, H W Porter & Co, Inc, Newark, and (for mail) 71 Lafayette Ave, East Orange, N J

TUSCH, Walter (*M* 1917) Secy, Tenney & Ohmes, Inc, 101 Park Ave, New York, and (for mail) 881 Sterling Place, Brooklyn, N Y

TUTHILL, Arthur F (*S* 1938) Student (for mail) University of Wisconsin, 1709 Regent St, Madison, Wis, and Cutchoque, L I, N Y

TUTTLE, George H (*M* 1937, *A* 1936, *J* 1934) Htg Engr (for mail) The Detroit Edison Co, 2000 Second Ave, and 16714 Kentfield, Detroit, Mich

TUTTLE, J. Frank (*M* 1913) Sales Agent (for mail) Warren Webster & Co, 127 Federal St, Boston, and 9 Lewis Rd, Winchester, Mass

TUVE, G. L.* (*M* 1932) Prof of Heat-Power Engrg (for mail) Case School of Applied Science, and 1294 Cleveland Heights Blvd, Cleveland, O

TUXHORN, David B (*M* 1936) Engr, L P Steuart and Bro, Inc, 138-12th St, N E, and (for mail) 4853 Sedgwick St, N W, Washington, D C

TWIST, Charles F. (*M* 1921) Pres (for mail) Ashwell-Twist Co, 967 Thomas St, and 2310 Tenth Ave, N, Seattle, Wash

TWIZELL, Edwin W (*M* 1937) Partner (for mail) Connolly & Twizell Regd, 1405 Bishop St, and 5176 Westbury Ave, Montreal, P Q, Canada

TYLER, Roy D. (*M* 1928) Mgr (for mail) Modine Mfg Co, 101 Park Ave, Room 1734, New York, and 15 Highbrook Ave, Pelham N Y

TYSON, William H. (*M* 1928) Mgr of Engrg (for mail) Goodyear Tyre & Rubber Co, Ltd, and "Kipewa" Codsall Rd, Nr Wolverhampton, England.

U

UHL, Edwin J (*M* 1925) Partner (for mail) Uhl Co, 132 S Tenth St, and 4830 Pleasant Ave, S, Minneapolis, Minn

UHL, Willard F. (*M* 1918) Partner (for mail) Uhl Co, 132 S Tenth St, and 4716 Lyndale Ave, S, Minneapolis, Minn

UHLHORN, W. J (*M* 1920) 733 S Highland Ave, Oak Park, Ill

ULLMAN, Herbert G (*A* 1928) Heating Engr, 107 White Rd Scarsdale, N Y

ULLRICH, Anton B., Jr. (*J* 1937) Sales Engr, Gilbert Engrg Co, 1314 Liberty Bank Bldg, and (for mail) 1330 Hollywood, Dallas, Tex

UPSON, Walter L. (*M* 1938) Dir of Research (for mail) Torrington Manufacturing Co, Torrington, and Litchfield, Conn

URDAHL, Thomas H (*M* 1930) Consulting Engr (for mail) 726 Jackson Place, N W, and 1505 44th St, N W, Washington, D C

V

VALE, Henry A L. (*M* 1929) Managing Director (for mail) Vale & Co, Ltd, 141-43 Armagh St, and 203 Ilam Rd, Fendalton, Christchurch, New Zealand

VAN ALSBURG, J H* (*M* 1931) Sales Engr., Hart & Cooley Mfg Co, 61 W Kinzie St, Chicago, Ill

VANCE, Louis G. (*M* 1919) Partner, Vance-McCrea Sales Co, 2700 Sisson St, and (for mail) 4402 Maine Ave, Forest Park, Baltimore, Md

VANDERHOOF, A. L. (*A* 1933) Dist Repr (for mail) Warren Webster & Co, 2341 Carnegie Ave, Cleveland, and 2762 Landon Rd, Shaker Heights, O

VANDERLIP, P. J (*A* 1935) Consulting Engr (for mail) Howland Engrg Co, 206 S Grand Ave, and 911 W Ottawa St, Lansing, Mich

VAN NOUHUYS, Herbert C. (*J* 1937) Engr, Air Cond Div, Nash-Kelvinator Corp, 14250 Plymouth Rd, and (for mail) 2007 Seward Ave, Detroit, Mich

VAN NUYS, Jay C (*J* 1938, *S* 1936) Junior Partner, P C Van Nuys, Archt, 1 W Main St, and (for mail) 56 W Cliff St, Somerville, N J

VARNER, John L. (*A* 1935) Air Cond and Commercial Engr, Jacksonville Refrigeration, Inc, 35 W Monroe St, and (for mail) 26 W 6th St, Jacksonville, Fla

VARNEY, Frank H, Jr. (*J* 1938) Product Engr, Pacific Dist, Air Cond Dept (for mail) General Electric Co, 116 New Montgomery St, and 10 Russian Hill Place, San Francisco, Calif

VAUGHAN, John G, Jr (*J* 1935) Norair Engineering Corp, 1114-18th St, N W, Washington, D C, and (for mail) 8405-16th St, Silver Spring, Md

VAUGHAN, Lillian Lee (*M* 1938) Prof of Mech Engrg (for mail) North Carolina State College, State College Station, and 11 Enterprise St, Raleigh, N C

VAUGHN, Frank R. (*M* 1937, *A* 1936) Vice-Pres (for mail) Green Foundry & Furnace Wks, and 532 Polk Blvd, Des Moines, Ia

VEALE, Tinkham (*J* 1938) Sales Engr, Avery Engineering Co, 2341 Carnegie Ave, Cleveland, and (for mail) 18519 Kinsman Rd, Shaker Heights, O

VELTMAN, B M (*M* 1936) Dept Mgr, Engrg Dept, Sears Roebuck & Co, and (for mail) 5531 Seward Park Ave, Seattle, Wash

VERNON, J. Rexford (*M* 1928, *A* 1926) Advertising Mgr (for mail) Johnson Service Co, 1355 Washington Blvd, Chicago, and 733 Brummel St, Evanston, Ill

VERVOORT, Edward L (*J* 1937, *S* 1936) Sales Engr, Brooklyn Union Gas Co, 180 Remsen St, Brooklyn, and (for mail) 31 Yale Place, Rockville Centre, N Y

VETLESEN, G Unger (*M* 1930) 1 Beekman Place, New York, N Y

VIDALE, Richard (*M* 1935) Air Cond Engr (for mail) Flesch & Schmidt, Inc, 60 Brown St, and 572 Flower City Park, Rochester, N Y.

VINCENT, Paul J (*M* 1931) Engr, Paul J Vincent Co, 2133 Maryland Ave, Baltimore, Md

VINSON, Neal L. (*J* 1936, *S* 1932) Engr and Estimator, L W Vinson & Son, Bisbee and Douglas, and (for mail) Box 3007, Lowell, Ariz

VISSAC, Gustave A (*M* 1937) Consulting Mining Engr, Coal Preparation, 1325 Frontenac Ave, Calgary, Alta, Canada

VIVARTTAS, E. Arnold (*Life Member*, *M* 1910) Engr, 222 Utter Ave, West Brighton, S I, N Y

VOISINET, Walter E (*M* 1930) Repr (for mail) John J Nesbitt, Inc, 250 Delaware Ave, Buffalo, and 151 Warren Ave, Kenmore, N Y

VOLBERDING, Leroy A (*A* 1936) Air Cond Engr (for mail) Norge Div of Borg-Warner, 670 E Woodbridge, Detroit, and 471 Oakland Ave, Birmingham, Mich

VOLK, Joseph H (*M* 1923) Pres and Treas (for mail) Thos E Hoye Heating Co, 1906 W St Paul Ave, and 2965 South 43rd St, Milwaukee, Wis

VOLKHARDT, Aquila N (*M* 1938) Dist Mgr, Alfred L Hart, Inc (Distributor Gen Electric Air Conditioning) 1243 Castleton Ave, West New Brighton, and (for mail) 104 Townsend Ave, Stapleton, S I, N Y

VOLLMANN, Carl W. (*M* 1938) Pres (for mail) Linde Canadian Refrigeration Co, Ltd, 355 St Petre St, Montreal, and 517 Roslyn Ave, Westmount, P Q, Canada

vonCHRISTIERSON, Carl A. (*J* 1937) Field Engr, Carbondale Dept, J H Vivian & Co, P O Box 301, and (for mail) P O Box 3249 Johannesburg, South Africa

VOORHEES, G. A. (*M* 1922) Mgr (for mail) Furblo Co, and P O Box 63, Hermansville, Mich

VOSS, Walter W. (*A* 1938) Instruction Engr, Utilities Engineering Institute, 404 N Wells St, and 1347 N Dearborn St, Chicago, Ill

VROOME, Albert E (*M* 1932) Air Cond Engr, Phoenix Engineering Corp, 2 Rector St, New York, N Y, and (for mail) 6218 Amboy Rd, Prince Bay, S I, N Y

W

WACHS, Louis J (*A* 1936, *J* 1930) Salesman, Carrier Corp, Chrysler Bldg, and (for mail) 1820 Cortelyou Rd, Brooklyn, N Y

WADDINGTON, B C. (*M* 1922) Chief Engr, Major Appliance Co, and (for mail) 4523 Wirt St, Omaha, Nebr

WADE, Richard H. (*M* 1937) Engr, Spencer Heater Co, 101 Park Ave, New York, and (for mail) 130-73-230th St, Laurelton, L I, N Y

WADSWORTH, Raymond H (*J* 1937) Sales Engr (for mail) Clarage Fan Co, 500 Fifth Ave, New York, N Y, and 112 Summit St, East Orange, N J

WAECHTER, Herman P (*A* 1930, *J* 1927) Air Cond Engr, W T Grant Co, 1441 Broadway, New York, and (for mail) 89 Sherman Ave, Tompkinsville, S I, N Y

WAGGONER, Jack H (*M* 1937) Test Engr, Owens-Corning Fiberglas Corp, Newark, O

WAGNER, Earle K (*M* 1938) Sales Engr (for mail) The Powers Regulator Co, 2240 N Broad St, Philadelphia, and 312 Myrtle Ave, Cheltenham, Pa

WAGNER, Edward A (*M* 1937, *A* 1936) Pres, Wagner Engineering Corp, 22 Dunham St, and (for mail) 28 Waverly St, Pittsfield, Mass

WAGNER, Frederick H, Jr (*M* 1934) Pres (for mail) Jenkins Mfg Co, 2 West 45th St, New York, and 1126 Post Rd, Scarsdale, N Y

WAGNER, John G. (*A* 1937) Employment Interviewer (for mail) N J Employment Service, 283 Market St, Paterson, and 254 Williams Ave, Hasbrouck Heights, N J

WAHRENBROCK, Orin K. (*J* 1936) Engr, Automatic Appliance Corp, 36 Richmond Hill Ave, Stamford, and (for mail) 366 West Ave, Box 117, Glenbrook, Conn

WAID, Glen H (*A* 1930) Dist Sales Mgr, Scott Valve Mfg Co, 3963 McKinley Ave, and (for mail) 2928 Northwestern Ave, Detroit, Mich

WALDON, Charles D. (*A* 1932) Consulting Engr, Spencer Foundry Co, Penetang, and (for mail) 32 Ferndale Ave, Toronto, Ont, Canada

WALFORD, Leslie C A (*M* 1938) Chief Designer, G Lorne Wiggs, Consulting Engineer, 727 University Tower, and (for mail) 4264 Royal Ave, Montreal, P Q, Canada

WALKER, Edmund R. (*M* 1934) Sales Mgr (for mail) Fedders Mfg Co, Inc, 57 Tonawanda St, Buffalo, and 365 McKinley Ave, Kenmore, N Y

WALKER, James F (*J* 1937, *S* 1936) 214 Rockwood Ave, Dayton, and (for mail) 2139 Abington Rd, Cleveland, O

WALKER, J. Herbert* (*M* 1916) (Council, 1938) Engr Asst to the Gen Mgr (for mail) The Detroit Edison Co 2000 Second Ave, Detroit, and 432 Arlington Rd, Birmingham, Mich

WALKER, Kirby (*M* 1935) Sales Engr, American Radiator Co, 40 West 40th St, New York, N Y

WALLACE, David R (*A* 1937) Htg Engr (for mail) Young & Bortic Coal Co, Ridgewood, and 94 Harding Rd, Glen Rock, N J.

WALLACE, George J. (*M* 1923) Principal Engr and Contractor, 96-19-35th Ave, Corona, and (for mail) 27-36 Ericsson St, East Elmhurst, N Y

WALLACE, George N. (*M* 1937) (for mail) George N Wallace Co, 271 Madison Ave, New York, and 77 Valley Rd, New Rochelle, N Y

WALLACE, Harry P, Jr (*A* 1936) Branch Mgr, Sales Promotion (for mail) Crane Co, 400 Third Ave, N, and 4909-34th Ave, S, Minneapolis, Minn

WALLACE, William M, II (*M* 1929) Resident Partner (for mail) Syska & Hennessy, Consulting Engrs, 111 N. Corcoran St, and 1011 Monmouth Ave, Durham, N C

WALSH, Edward R, Jr (*M* 1936, *A* 1935) Head of Automatic Htg Div, York Ice Machinery Corp, and (for mail) 32-34 Elm Terrace Apts, York, Pa

WALSH, James A. (*A* 1932, *J* 1929) Sales Mgr (for mail) Air Conditioning Co, Main at Richmond St, and 803 Hawthorne St, Houston, Tex

WALTERS, Arthur L. (*M* 1926, *A* 1925, *J* 1924) Chief Engr (for mail) Green Foundry & Furnace Works, 322 S W 3rd St, and 900-29th St, Des Moines, Ia

WALTERS, William T. (*M* 1917) Engr, Illinois Engineering Co, Cor 21st St and Racine Ave, and (for mail) 12747 Wallace Ave, Chicago, Ill

WALTERTHUM, John J. (*A* 1922) Htg-Vtg Contractor, 212 E 58th St, New York, N Y, and (for mail) 42-a Van Reipen Ave, Jersey City, N J

WALTHER, Frederick G (*J* 1938) Draftsman and Engr, York Ice Machinery Corp, 2nd Ave and 42nd St, Brooklyn, and (for mail) 219 Bronx River Rd, Yonkers, N Y

WALTON, Charles W., Jr. (*M* 1934) Mech Engr (for mail) Rockefeller Center, Inc, 50 Rockefeller Plaza, New York, N Y, and 120 Monte Vista Ave, Ridgewood, N J

WALZ, George R. (*J* 1937) Sales Engr (for mail) Minneapolis-Honeywell Regulator Co, 1101 Vermont Ave, N W, Washington, D C, and 1808 Queens Lane, Apt 209, Arlington, Va

WARD, Edward B (*M* 1937) Pres (for mail) Edward B Ward & Co, 270 Tremont St, and 235 Lansdale Ave, San Francisco, Calif

WARD, Frank J. (*M* 1935) Owner (for mail) Frank J Ward Co, 237 W Court St, Cincinnati, O, and Cold Spring, Ky

WARD, Harry H. (*A* 1937) Dist Engr, Delco-Frigidaire Cond Div (for mail) General Motors Sales Corp, 230 N E 14th St, and 724 N W 12th St, Miami, Fla

WARD, Jerry J. (*A* 1921) Pres (for mail) Wenzler & Ward, Inc, 1703 Textile Tower, and 1107-31st Ave, Seattle, Wash

WARD, Oscar G. (*M* 1919) Dist Mgr, Johnson Service Co, 1355 Washington Blvd, Chicago, Ill

WARDELL, Arthur (*M* 1935) Asst Prof of Engrg Drawing, University of Toronto, and (for mail) 124 Melrose Ave, Toronto, Ont, Canada

WARE, John H., III (*M* 1937) Vice-Pres, Citizens Gas & Fuel Co, Pres, Oxford Co, Vice-Pres, Gas Oil Products, Inc, 45 S Third St, Oxford, Pa

WARING, James M. S. (*M* 1932) Consulting Engr, 277 Park Ave, New York, N Y

WARREN, Francis C. (*M* 1934) Branch Mgr, (for mail) American Blower Corp, 200 Division Ave, N, and 329 Gladstone Ave, S E, Grand Rapids, Mich

WARREN, John S., Jr. (*J* 1937) Sales Engr (for mail) York Ice Machinery Corp, 115-121 South 11th St, and 2017 Maury Ave, St Louis, Mo

WARREN, Robert M., Jr (*J* 1938) Air Cond Engr (for mail) Sam E Beck, Inc, 400 Brookstown Ave, and 1126 Walker Ave, Winston-Salem, N C

WASHBURN, Marcus J. (*A* 1934) William Powell Company, Cleveland, O

WASHINGTON, George (*M* 1934) Engr, Hoffman Specialty Co, Waterbury, Conn, and (for mail) 4327 Johnson Ave, Western Springs, Ill

WASHINGTON, Laurence W. (*M* 1929) (for mail) The Powers Regulator Co, 702 American Bldg, and 1627 Northwood Drive Cincinnati, O

WASSER, Munny (*M* 1938) Elec Mech Engr, Igeme S A R, Str Aureliu 25, and (for mail) Bd Carol 62, Bucharest, Roumania

WASSON, Robert A (*M* 1938) Eastern Dist Mgr (for mail) Clarage Fan Co, 500 Fifth Ave, New York, and 15 Willow St, Brooklyn, N Y

WATERMAN, John H. (*M* 1931) Engr, Chas T Main, Inc, 201 Devonshire St, Boston, Mass

WATERS, Frank A. (*A* 1936) Htg-Vtg Engr, Westinghouse Elec Supply Co, 150 Varick St, New York, and (for mail) Bedford Hills, N Y

WATERS, George G. (*M* 1931, *A* 1926) Dist Mgr (for mail) American Blower Corp, 1433 Oliver Bldg, and 110 Longuevue Drive, Pittsburgh (16) Pa

WATKINS, George B. (*A* 1936) Dir of Research (for mail) Libbey-Owens-Ford Glass Co, 1701 E Broadway, and 3004 Berdan Ave, Toledo, O

WATSON, H. Dalton (*A* 1935) Branch Mgr (for mail) Linde Canadian Refrigeration Co, 124 King St, and 830 Mulvey Ave, Winnipeg, Man, Canada

WATSON, M Barry (*M* 1928) Consulting Engr (for mail) 184 College St, and 121 Welland Ave, Toronto, Ont, Canada

WATT, Robert D. (*J* 1937) Engr, H W Beecher Consulting Engr, 502 Securities Bldg, and (for mail) 1306 Madison St, Seattle, Wash

WATTS, Albert E. (*A* 1937) Mgr, A E Watts, 637 Craig St, W, and (for mail) 2347 Beaconsfield Ave, Montreal, P Q, Canada

WAUDBY, Walter (*M* 1938) Engr (for mail) American Radiator Co, 149 Blvd Haussmann, Paris, and 26 Rue de la Tourelle, Boulogne sur Seine, France

WAUNG, Tsing Fi (*M* 1935, *J* 1933) Htg Engr, (for mail) Andersen Meyer & Co, Ltd, Yuen Ming Yuen Rd, and Apt A-5, 1562 Ave Joffre, Shanghai, China

WAYLAND, Clarke E. (*A* 1937) Vice-Pres (for mail) Western Asbestos Co, 675 Townsend St, and 42 Allston Way, San Francisco, Calif

WEATHERBY, Edward P., Jr. (*J* 1936, *S* 1935) Product Engr, Air Conditioning Dept, General Electric Co, 4966 Woodland Ave, Cleveland, and (for mail) 1243 Warren Rd, Lakewood, O

WEBB, Ernest C. (*M* 1935) Engrg Service Mgr (for mail) Iron Fireman Mfg Co, 3170 West 106th St, Cleveland, and 1202 Woodside Drive, Rocky River, O

WEBB, John S. (*M* 1920) Sales Engr, W D Cashin Co, 69 A St, Boston, and (for mail) 345 Brookline St, Needham, Mass

WEBB, John W. (*M* 1926) Managing Dir (for mail) Webb Dust Removing & Drying Co, Ltd, Vinery Works, Town Lane, Denton Nr. Manchester, and "Ebor", Brinnington, Stockport, England,

WEBER, Erwin L. (*M* 1921) Consulting Engr, 534 Medical Arts Bldg, Seattle, Wash

WEBER, Eugene F. (*J* 1937) Sales Engr, York Ice Machinery Corp, 117 S 11th St, and (for mail) 4515 Maryland Ave, St Louis, Mo

WEBSTER, E. Kessler (*M* 1915) Warren Webster & Co, 17th and Federal Sts, Camden, N J

WEBSTER, Warren, Jr (*M* 1932, *J* 1927) Vice-Pres and Treas (for mail) Warren Webster & Co, 17th and Federal Sts, Camden, and 200 Colonial Ridge Drive, Haddonfield, N J

WEBSTER, William H, Jr. (*A* 1935) Vice-Pres (for mail) Hurst Heating Engineers, Inc, 400 York St, and 200 N Shore Rd (Academy Terrace) Norfolk, Va

WECHSBERG, Otto (*M* 1932) Pres-Gen Mgr, Coppus Engineering Corp, 344 Park Ave, and (for mail) 1006 Main St, Worcester, Mass

WEDDELL, George O (*M* 1936) Branch Mgr, York Ice Machinery Corp, 2400 Carson St, and (for mail) 3114 Wainbell Ave, Dormont, Pittsburgh, Pa

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WEID, Harry L (*S* 1938) Test Engr, Hynes Electric Heating Co, 240 Cherry St, and (for mail) 2967 N Mutter St, Philadelphia, Pa.

WEIDELE, Erwin J (*A* 1937) Htg and Vtg Inspector, City of Los Angeles, Room M8, City Hall, and (for mail) 8519 Colecrest Drive, Los Angeles, Calif

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WEIL, Martin (*A* 1925) Vice-Pres (for mail) Weil-McLain Co, 641 W Lake St, and 4259 Hazel St, Chicago, Ill

WEIL, M. I. (*A* 1928) Pres (for mail) Chicago Pump Co, 2336 Wolfram St, and 1409 W. Elmdale Ave, Chicago, Ill

WEIMER, Fred G. (*A* 1919) Mgr, Milwaukee Office, Kewanee Boiler Corp, 312 E Wisconsin Ave, Room 412, and (for mail) 3958 N Stowell Ave, Milwaukee, Wis

WEINERT, Fred C. (*A* 1937) Asst Sales Mgr, Sales Engr (for mail) Chamberlin Metal Weather Strip Co, Inc, 1254 Labrosse St, Detroit, and Route No 2, Plymouth, Mich

WEINSHANK, Theodore* (*Life Member, M* 1906) (Board of Governors, 1913) Consulting Engr, 3307 Belden Ave, Chicago, Ill

WEISS, Arthur P. (*M* 1928) Burnham Boiler Corp, Irvington, and (for mail) 134 Farrington Ave, North Tarrytown, N. Y

WEISS, Carl A. (*M* 1936, *A* 1924) Gen Mgr (for mail) Kornbrodt Kornice Co., 1811-15 Troost Ave, and 29 East 68th St, Kansas City, Mo

WEISSBLATT, Norman (*S* 1938) Student, N Y Technical Inst., 108 Fifth Ave, New York, and (for mail) 1435 53rd St, Brooklyn, N Y

WEITZEL, Cameron B. (*M* 1936) Owner and Operator, 122 E High St, Manheim, Pa

WEITZEL, Paul H. (*J* 1936, *S* 1934) Jr. Engr, Cameron B Weitzel, 122 East High St, Manheim, Pa

WELCH, Louis A, Jr. (*A* 1929) Owner (for mail) Welch Bros, 443-2nd St, and 2001 Campbell Ave, Schenectady, N. Y

WELDY, Lloyd O. (*M* 1930) Branch Mgr (for mail) The Powers Regulator Co, 2341 Carnegie Ave, Cleveland, and 19623 Laurel Ave., Rocky River, O.

WELLS, Earl P. (*M* 1938) Air Cond Engr, Gay Engineering Corp, 2730 E 11th St, Los Angeles, and (for mail) 206 W Shorb St, Alhambra, Calif

WELSH, Harvey A. (*A* 1936) Engr, A P. Woodson Co, 1313 H St, N. W, Washington, D. C, and (for mail) 4118 Lee Highway, Arlington, Va

WELTER, Michael A. (*A* 1925) Engr. and Htg Contractor (for mail) Welter Furnace Mfg. Co, 2023 S Lyndale St, and 4200 S Aldrich Ave, Minneapolis, Minn

WENDT, Edgar F. (*M* 1918) Pres (for mail) Buffalo Forge Co, 490 Broadway, and 120 Lincoln Parkway, Buffalo, N Y

WENDT, Edwin H. (*J* 1936) Engr (for mail) Wendt & Crone Co, 2124 N Southport Ave, and 3809 N Troy St, Chicago, Ill

WERNER, John G. (*M* 1937) Branch Mgr (for mail) L J Mueller Furnace Co, Delaware Ave, and Morris St, Philadelphia, and 215 N Easton Rd, Glenside, Pa

WERNER, Richard K. (*M* 1936) Consulting Engr (for mail) 316 W T Waggoner Bldg, and 3671 Monticello Drive, Fort Worth, Tex

WESLEY, Ray O. (*A* 1937) Sales Engr (for mail) U S Radiator Corp, 334 Boren Ave, N, Seattle, and Yarrow Point, Bellevue, Wash

WEST, Perry* (*M* 1911) Prof of Steam & Power Engrg, Head of Dept Mech Engrg (for mail) College of Engineering, University of Kentucky, and 185 E Maxwell St, Lexington, Ky

WESTOVER, Wendell (*M* 1936) Pres (for mail) Westover-Wolfe, Inc, 170 Washington Ave, and 221 S Main Ave, Albany, N Y

WESTPHAL, Norman E. (*S* 1937) Engr, C A Dunham Co, and (for mail) Long Beach, Michigan City, Ind

WETHERED, Woodworth (*M* 1937) Engr, Johnson & Wethered, Hotel Sir Francis Drake, San Francisco, Calif

WETZELL, Horace E. (*M* 1934) Chief Engr (for mail) The Smith & Oby Co, 6107 Carnegie Ave, Cleveland, and 21144 Aberdeen Rd, Rocky River, O

WHEELER, Joe, Jr. (*M* 1937) Salesman (for mail) Johnson Service Co, 28 E 29th St, New York, and 261 Dogwood Lane, Manhasset, L I, N Y

WHELAN, William J. (*M* 1923) Purchasing and Estimating (for mail) Harrigan & Reid Co, 1365 Bagley, and 3790 Seminole Ave, Detroit, Mich

WHELLER, Harry S. (*M* 1916) Vice-Pres, L J Wing Mfg Co, 154 W 14th St, New York, N Y and (for mail) 725 Union Ave, Elizabeth, N J

WHITE, Elmer D (*J* 1937) Engr (for mail) Ranco, Inc, 601 W Fifth Ave, and 211 W Weisheimer Rd, Columbus, O

WHITE, Elwood S. (*M* 1921) Pres (for mail) U S Radiator Corp, 1056 National Bank Bldg, Detroit, Mich, and Meadowbank Rd, Old Greenwich, Conn

WHITE, Eugene B. (*M* 1934) Arch & Engr. (for mail) Architectural & Engineering Bureau, 19 S LaSalle Street, Chicago, and 309 N Taylor Ave, Oak Park, Ill

WHITE, Everett A. (*M* 1921) Engr Dept, Crane Co, 30 South 16th St, and (for mail) 5244 Nottingham Ave, St Louis, Mo

WHITE, Everett G. (*A* 1938) Asst Custodian Engr, Bronx Central Post Office, Bronx, and (for mail) 425 Rochelle Terrace, Pelham Manor, N Y

WHITE, Harry S. (*A* 1936) Mgr (for mail) Acme Sheet Metal Co, 2201 Broadway, and 20 West Dartmouth Rd, Kansas City, Mo

WHITE, John C. (*M* 1932) State Power Plant Engr (for mail) Wisconsin Bureau of Engineering, Power Plant Div, 624 E Main St, and 622 E Main St, Madison, Wis

WHITE, Robert C. (*A* 1938) Sales (for mail) Carrier Corp, Merchandise Mart, and 2016 W. Berwyn, Chicago, Ill.

WHITE, Taylor G., Jr. (*A* 1937) Sales Engr, U S Radiator Corp, and (for mail) 615 State St, Louisville, Ky

WHITE, Thomas J. (*J* 1938) Sales Engr (for mail) American Blower Corp, 625 Market St, and 1850 Sacramento St, San Francisco, Calif

WHITE, William R. (*M* 1938; *A* 1936) Engr, Air Cond Dept (for mail) Nebraska Power Co, 723 Electric Bldg, and 4339 Larimore Ave., Omaha, Nebr.

WHITELAW, H. Leigh (*M* 1916) Vice-Pres (for mail) American Gas Products Corp, 40 West 40th St, New York, N Y, and Rings End Rd, Noroton, Conn

WHITELEY, Stockett M. (*M* 1933) Consulting Engr. (for mail) Baltimore Life Bldg., and 3931 Canterbury Rd., Baltimore, Md.

WHITESELL, Roy H. (*A* 1938) Sales Repr, Bryant Heater Air Cond Corp, Chicago, Ill, and (for mail) 4617 Zenith Ave, S, Minneapolis, Minn

WHITMER, Robert P. (*M* 1935) Secy (for mail) American Foundry & Furnace Co, McClun & Washington Sts, 1402 E Washington St, Bloomington, Ill

WHITNEY, C. W (*M* 1935) Pres, ABC Oil Burner & Engineering Co, 2012-14 Chestnut St, Philadelphia, and (for mail) Apt F-1, Sevilla Court, Bala-Cynwyd, Pa

WHITT, Sidney A (*A* 1938, *J* 1937) Sales Application Engr (for mail) Nash-Kelvinator Corp, 14250 Plymouth Rd, and 12043 Cloverlawn, Detroit, Mich

WHITTAKER, Wayne K. (*A* 1935) Building Maintenance Mechanic, Irving Trust Co Bldg, 1 Wall St, New York, and (for mail) 119-23-226th St, St Albans, N Y

WHITTEN, Horace E. (*M* 1924) Pres and Treas H E Whitten Co, 9 Federal Court, Boston, and (for mail) 56 Highland Rd, Somerville, Mass

WHITTINGTON, James A. (*M* 1936) Utilization Testing Engr (for mail) Peoples Gas Light & Coke Co, 3921 S Wabash Ave, Chicago, and 622 Sheridan Square, Evanston, Ill

WIDDOWFIELD, Arthur S (*J* 1937) Sales Engr, The Mercoid Corp, 4201 Belmont Ave, and 6109 N Campbell Ave, Chicago, Ill

WIEGNER, Henry B. (*M* 1919) Branch Mgr, Johnson Service Co, 20 Winchester St, Boston, and (for mail) 143 Standish Rd, Watertown, Mass

WIGGS, G. Lorne (*M* 1936, *A* 1932, *J* 1924) (Council, 1938) Consulting Engr (for mail) 727 University Tower, and 4797 Grosvenor Ave, Montreal, P Q, Canada

WIGLE, Bruce (*A* 1926) Pres, Bruce Wigle Plumbing and Heating Co, 9117 Hamilton Ave, Detroit, Mich

WILDE, Ray S. M. (*M* 1916) Consulting Mech Engr, 3500 Union Guardian Bldg, and (for mail) 194 Connecticut Ave, Highland Park, Mich.

WILDER, Edward L. (*M* 1915) Engr, Industrial Dept (for mail) Rochester Gas & Electric Corp, 89 East Ave, and 2 Buffard Drive, Rochester, N Y

WILDER, Herbert P. (*M* 1938) Engr, Patterson-Kelley Co, 101 Park Ave, New York, and Ardsley Rd, Scarsdale, N Y

WILEY, Donald C (*J* 1936) Engr (for mail) John J Nesbitt, Inc, State Road & Rhawn St, and 3338 St Vincent St, Philadelphia, Pa

WILHELM, Joseph E. (*J* 1936, *S* 1934) Office Engr and Purch Agent, Avery Engineering Co, 2341 Carnegie Ave, and (for mail) 1804 East 100th St, Cleveland, O

WILKES, Gordon B. (*M* 1937) Prof of Heat Engrg (for mail) Massachusetts Institute of Technology, Cambridge, and 51 Everett St., Newton Centre, Mass

WILKINSON, Arthur (*A* 1936) Mgr (for mail) Wilkinson Engineering Agencies, 1253 McGill College Ave, Montreal, and 469 Argyle Ave, Westmount, P Q, Canada

WILKINSON, F. J. (*M* 1933) Mgr, Cent Engrg. Service, Montgomery Ward & Co, Chicago Ave & Larabee St, Chicago, and (for mail) 18257 Martin Ave, Homewood, Ill

WILLARD, Arthur C.* (*M* 1914) (*Presidential Member*) (Pres, 1928, 1st Vice-Pres, 1927, 2nd Vice-Pres, 1926; Council, 1925-1929) (for mail) President, University of Illinois, and 711 Florida Ave, Urbana, Ill

WILLER, Murray D. (*J* 1937; *S* 1936) Air Cond Engr, F W Chambers & Co., Ltd, 96 Bloor St, W, and (for mail) 1243 St Clair Ave, W, Toronto, Ont, Canada

WILLEY, Earl C. (*M* 1934) Asst Prof of Mech Engrg, Oregon State College, and (for mail) 1652 "A" St, Corvallis, Ore

WILLIAMS, Allan E. (*M* 1938) Branch Mgr (for mail) Buffalo Forge Co, 428 Dwight Bldg., and 3535 Wabash Ave., Kansas City, Mo.

WILLIAMS, Allen W. (*Life Member*, *A* 1915) Managing Dir, National Warm Air Heating & Air Conditioning Association, 50 West Broad St, and 51 Meadow Park Ave, Columbus, O
WILLIAMS, Chester D (*M* 1938) Mgr, General Air Conditioning & Heating Co (for mail) 4001 Piedmont Ave, Oakland, and 2709 College Ave, Berkeley, Calif
WILLIAMS, Donald D. (*J* 1937) Gas Htg Engr (for mail) Iowa-Nebraska Light & Power Co, 14th and O Sts, and 2236 A St, Lincoln, Nebr
WILLIAMS, Douglas C. (*A* 1938) Engr (for mail) General Motors Sales Corp, 2031 Calumet Ave, Chicago, and 595 Illinois Rd, Lake Forest, Ill
WILLIAMS, Frank H. (*J* 1934) Tech Engr, Delco-Frigidaire Div, General Motors Sales Corp, Taylor St, Dayton, O, and (for mail) 68 Amherst Rd, Pleasant Ridge, Mich
WILLIAMS, Gordon S. (*J* 1937, *S* 1936) Branch Mgr & Engr (for mail) The Trane Co, 207 Orange St, and 35 Dewitt St, New Haven, Conn
WILLIAMS, J Walter (*M* 1915) Pres (for mail) Forest City Plumbing Co, 332 E State St, and 923 E State St, Ithaca, N Y
WILLIS, Leonard L. (*J* 1936, *S* 1935) Engr, Conrad Refrigeration Co, 17 E Hennepin Ave, and (for mail) 5036 Lyndale S, Minneapolis, Minn
WILLNER, Ira (*M* 1937) Pres (for mail) Willner Heating Co, Inc, 415 Lexington Ave, New York, and 3875 Waldo Ave, Riverdale, N Y
WILLS, Fred W (*J* 1938) Western Sales Mgr (for mail) Tuttle & Bailey, Inc, 61 W Kinzie St, and 2511 Leland Ave, Chicago, Ill
WILMOT, Charles S (*M* 1919) (for mail) Building Insulation Co, Lancaster Ave at Jefferson St, Philadelphia, and 406 Essex Ave, Narberth, Pa
WILSON, Alexander (*M* 1936) Consulting Engr (for mail) 1537 St Matthew St, Montreal, P Q, Canada
WILSON, Andrew (*M* 1935) Survey and Estimating Engr, Paragon Oil Burner Corp, 75 Bridgewater St, and (for mail) 5523 Seventh Ave, Brooklyn, N Y
WILSON, Eric D. (*M* 1936) General Mgr in India, (for mail) Carrier Corp, c/o Volkart Bros, Ballard Estate, Bombay, India, and Gorof House, Ystradgynlais, Swansea, England
WILSON, Frederick J. (*A* 1938) Pres (for mail) Wilson Air Conditioning Corp, 1726 Sansom St, Philadelphia, and 527 Baird Rd, Merion, Pa
WILSON, George T. (*M* 1925) Sales Engr, Gurney Foundry Co, Ltd, 4 Junction Rd, Toronto, and (for mail) Tyre Ave, Islington, Ont, Canada
WILSON, Raymond W. (*M* 1934) Member of Firm (for mail) Wilson-Brinker Co, 412 Pythian Bldg, and 429 Creston Ave, Kalamazoo, Mich
WILSON, Robert A. (*M* 1936) Sales Engr, Minneapolis-Honeywell Regulator Co, 4501 Prospect Ave, Cleveland, and (for mail) 1520 Grace Ave, Lakewood, O
WILSON, Victor H. (*A* 1938) Engr, John Bouchard & Sons Co, 1024 Harrison St, and (for mail) 2613 Acklen Ave, Nashville, Tenn
WILSON, W. H. (*A* 1932) Chief Power Plant Engr, Pullman-Standard Car Mfg Co, 11001 Cottage Grove Ave, and (for mail) 22 West 110th Place, Chicago, Ill
WILTBERGER, Constant F. (*M* 1935) Partner, Pennell & Wiltberger, Consulting Engrs, Land Title Bldg, and (for mail) 2650 North 9th St, Philadelphia, Pa
WINANS, Glen D. (*M* 1929) Engr of Steam Distribution (for mail) The Detroit Edison Co, 2000 Second Ave, and 16183 Wisconsin, Detroit, Mich
WINKLER, Ralph A. (*J* 1937) Sales Engr (for mail) Alfred C Goethel Co, 2337 North 31st St, and 2766A North 41st St, Milwaukee, Wis
WINSLOW, C.-E. A.* (*M* 1932) Prof of Public Health (for mail) Yale School of Medicine, 310 Cedar St, and 314 Prospect St., New Haven, Conn.
WINSTEL, Frank E. (*A* 1938) Delco Sales Mgr., The Bimel Co, 305 Walnut St, and (for mail) 1126 Regent Ave, Cincinnati, O
WINTERBOTTOM, Ralph F. (*M* 1923) Engr., Winterbottom Supply Co, and (for mail) 400 Campbell Ave, Waterloo, Ia
WINTERER, Frank C. (*M* 1920) Sales Mgr (for mail) Cochran-Sargent Co, 300 Broadway, and 836 Juno St, St Paul, Minn
WINTHER, Anker (*M* 1937, *A* 1936, *J* 1932) Air Conditioning Engr (for mail) York Ice Machinery Corp, 659 E 6th St, and 3526 Pape Ave., Cincinnati, O
WISSING, Clement B. (*A* 1936) Secy and Sales Mgr, Ebner Ice & Cold Storage Co, Locust & Chestnut, Vincinnes, Ind
WITHERIDGE, David E. (*J* 1936) Sales Engr., W. A. Witheridge Co, 746 S Fourth Ave., Saginaw, Mich
WITMER, Charles N. (*A* 1937, *J* 1930) Sales Engr (for mail) Straus-Frank Co, 1618 Fannin St, and 2301 Southgate Blvd, Houston, Tex.
WITMER, Howard S. (*A* 1937) Sales Engr, Dail Steel Products Co, Hosmer & Main, Lansing, and (for mail) R F D 1, East Lansing, Mich
WOESE, Carl F. (*M* 1934) Consulting Engr (for mail) Robson & Woese, Inc, 1001 Burnet Ave, and 256 Robineau Rd, Syracuse, N Y
WOLF, John C. (*M* 1923) Htg and Vtg Engr. (for mail) B F Sturtevant Co, and 76 Beacon St, Hyde Park, Mass
WOLF, Philip (*M* 1935) Proprietor, City Contracting Co, 304 East 62nd St, New York, N Y
WOLFF, Peter P. (*M* 1935) Engr, Bell & Gossett Co, 3000 Wallace St, and (for mail) 1752 E. 73rd St, Chicago, Ill
WOLIN, Milton W. (*J* 1938, *S* 1937) Engr, Typhoon Air Cond Co, Inc, 1393 Lexington Ave, New York, N Y, and (for mail) R F D. No 2, Box 73-D, New Brunswick, N J
WOLL, Willard M. (*M* 1938) Engr (for mail) Commonwealth Edison Co, Room 1000 Edison Bldg, and 9320 S Throop St, Chicago, Ill
WOLLENBERGER, Louis (*M* 1938) Industrial Gas Engr (for mail) Coast Counties Gas & Electric Co, 22 Pacific Ave, and 424 King St, Santa Cruz, Calif
WONG, W S. B. (*M* 1938) Dir (for mail) American Engineering Corp (China) 989 Bubbling Well Rd, and 669 Hart Rd, Shanghai, China
WONSON, Arthur S, Jr. (*S* 1938) Student, Beverly Trade School, Beverly, and (for mail) Walnut Park Ave, Essex, Mass
WOOD, Alfred W. (*J* 1938) Sales Engr, Clare Bros & Corp, Ltd, and (for mail) 631 William St, Preston, Ont, Canada
WOOD, Charles F. (*M* 1937) Chief Engr or Mgr. Engrg Dept, Delco-Frigidaire Div, Taylor St. Plant, and (for mail) 359 Aberdeen Ave, Dayton, O
WOOD, Roderick A. (*J* 1937) Editorial Asst, Fowler-Becker Publishing Co, 420 Madison Ave, New York, and (for mail) 482 Bard Ave, West New Brighton, Staten Island, N Y
WOODBURY, Clyde D. (*M* 1938) Mech Engr, Leland & Haley, Consulting Engrs, 58 Sutter St, and (for mail) 1321 37th Ave, San Francisco, Calif
WOODHOUSE, Graham D. (*A* 1938) General Supt, Dowagiac Steel Furnace Co, and (for mail) 412 W High St, Dowagiac, Mich
WOODMAN, Lawrence E. (*M* 1934) Pres, Woodman Engineering Corp, Air Cond Engrs, 203 E Capitol, Jefferson City, Mo
WOODS, Bladwin M (*M* 1937) Prof of Mech. Engrg (for mail) University of California, and 249 The Uplands, Berkeley, Calif
WOODS, Edward H. (*M* 1934) Engr, Higgins Zabriskie, 314 E State St, Ithaca, N Y
WOODWARD, Rothwell (*M* 1938) Supvr of Tech Data, Delco-Frigidaire Conditioning Div, 300 Taylor St, and (for mail) 910 Cumberland Ave, Dayton, O
WOODWORTH, Corlis Z. (*M* 1938) Consultant & Resident, Kribs & Landauer, Inc., 303 Tradesmens Bank, and (for mail) 2736 N. W. 17th St., Oklahoma City, Okla.

WOOLCOCK, Edwin (*A* 1938) Mgr (for mail) C J Woolcock Plumbing & Heating, 2217-15th St, and Red Coach Inn, Niagara Falls, N. Y

WOOLLARD, Mason S. (*M* 1934) Htg Engr, H. H. Angus, Consulting Engr, 1221 Bay St, and (for mail) 31 Hillcrest Park Ave, Toronto, 5, Ont, Canada

WOOLLEY, J. Herbert (*A* 1936) Vice-Pres (for mail) Woolley Coal Co, Inc, 12 Burnett Ave, Maplewood, and Oakview Terrace, Short Hills, N J

WOOLSTON, A. H. (*M* 1919) Chief Engr (for mail) Woolston-Woods Co, 2132 Cherry St, and 4815 N 12th St, Philadelphia, Pa

WOOTAN, Charles (*A* 1937) 11118 Clifton Blvd, Cleveland, O

WORLD, Harry P (*M* 1936) Engr (for mail) Mackenzie Waters, Archt, 96 Bloor St, West, and 30 Rosewell Ave, Toronto, Ont, Canada

WORMLEY, Robert F (*A* 1938) Branch Mgr (for mail) Grinnell Co of Canada, Ltd, 700 Beaumont St, and 6092 Terrebonne Ave, Montreal, P Q, Canada

WORSHAM, Herman (*M* 1925, *J* 1918) National Business Dept (for mail) Delco-Frigidaire Cond Div., 300 Taylor St, and 519 W Norman Ave, Dayton, O

WORTHING, Stanley L. (*M* 1936) Consulting Engr (for mail) 433 Kelsey Bldg, Grand Rapids, and Spring Lake, Mich

WORTHINGTON, Thomas H. (*M* 1937) Mgr (for mail) Dominion Radiator & Boiler Co, Ltd, 405 Beaubien St, W, and 5145 Cote St Luc Rd, Montreal, P Q, Canada

WORTON, William (*M* 1937) Branch Mgr (for mail) C A Dunham Co, Ltd, 504 Scott Bldg, and 292 Landowne Ave, Winnipeg, Man, Canada

WRIGHT, Clarence E (*J* 1935, *S* 1933) Mgr, Htg & Vtg Dept, Fairmont Wall Plaster Co, Tenth St, and (for mail) 908 Gaston Ave, Fairmont, W Va

WRIGHT, Daniel K., Jr. (*J* 1938) Instructor in Mech Engrg, Case School of Applied Science, Cleveland, and (for mail) 1676 Lee Rd, Cleveland Heights, O

WRIGHT, Harris H. (*M* 1917) Mfrs Repr (for mail) 320 E 10th St, and 808 Greenway Terrace Kansas City, Mo

WRIGHT, K. A. (*M* 1921) Branch Mgr (for mail) Johnson Service Co, 1113 Race St, Cincinnati, O, and 113 Orchard Rd, Fort Mitchell, Ky

WRIGHT, M. Birney (*A* 1932, *J* 1929) Mech Engr (for mail) E I duPont de Nemours & Co, P O Box 1537, and Cedar Ave, South Hills, Charleston, W Va

WRIGHTSON, Wilbor T. (*M* 1937) Eastern Mgr (for mail) Garden City Fan Co, 55 West 42nd St, New York, and 22 Sagamore Rd, Bronxville, N Y.

WUNDERLICH, Milton S * (*M* 1925) Chairman Research and Test Committee, Insulite Co, 1100 Builders Exchange, Minneapolis, and (for mail) 545 Mt Curve Blvd, St Paul, Minn

WYATT, DeWitt H. (*M* 1936) Consulting Engr, Cooling & Heating Systems, 226 Northridge Rd, Columbus, O

WYLD, Reginald G. (*M* 1937) Vice-Pres, Charge of Engineering, Chrysler Corp, Airtemp Div, 1119 Leo St, Dayton, O

WYLIE, Howard M. (*M* 1925, *J* 1917) Vice-Pres in Charge of Sales (for mail) The Nash Engineering Co, and 51 Elmwood Ave, South Norwalk, Conn

Y

YAGER, John J. (*M* 1921) 425 Woodbridge Ave, Buffalo, N Y

YAGLOU, C. P.* (*M* 1923) Assoc Prof, Industrial Hygiene (for mail) Harvard School of Public Health, 55 Shattuck St, Boston, and 10 Vernon Rd, Belmont, Mass

YARBOROUGH, Thomas R (*M* 1938) Dist Sales Engr, Frigidaire Div of General Motors Sales Corp, and (for mail) 788 Greenwood Ave, Apt 4, Atlanta, Ga

YATES, James E (*M* 1934) Mgr (for mail) Yates, Neale & Co, 231-10th St, and 431-16th St, Brandon, Man, Canada

YATES, James E, Jr. (*J* 1936) Sales Engr (for mail) Yates, Neale & Co, 231-10th St, and 431-16th St, Brandon, Man, Canada

YATES, Walter (*Life Member*, *M* 1902) Governing Dir (for mail) Matthews & Yates, Ltd, Cyclone Works, and Parksend, Swinton, Manchester, England

YEAZEL, Gordon A (*J* 1938) Air Cond Engr, Sampson Electric Co, 3201 S Michigan Ave, and (for mail) 8130 Eberhart, Chicago, Ill

YERKES, William L. (*A* 1937) Engr (for mail) Carrier Corp, 12 S 12th St, Philadelphia, and 295 W Essex Ave, Lansdowne, Pa

YOUNG, Emil O. (*A* 1935) Pres (for mail) Young Regulator Co, 4500 Euclid Ave, and 2040 East 83rd St, Cleveland, O

YOUNG, Forest H, Jr (*A* 1936) Mgr, Secy-Treas, Young Heat Engineering Co, 116 N 26th St, Billings, Mont

YOUNG, Harold J (*M* 1937) Sales Engr, Young Radiator Co, Occidental Hotel Bldg, and (for mail) 1364 Lakeshore Drive, Muskegon, Mich

YOUNG, J. T., Jr (*A* 1936) Mgr, Htg Dept (for mail) Crane Co, Box 1110, and 1364 Kensington Ave, Salt Lake City, Utah

YUSKA, Leonard J. (*S* 1938) Student, State University of Iowa and (for mail) 41-C Quadrangle, Iowa City, Ia

Z

ZACK, Hans J (*M* 1928) Pres, Zack Co, 2311 Van Buren St, Chicago, Ill

ZANGRILLI, Albert J. (*J* 1937, *S* 1935) Htg Engr, Chandler Boyd Co, 51 Terminal Way, and (for mail) 537 Turrett St, Pittsburgh, Pa

ZIBOLD, Carl E (*M* 1929) Mech Engr, Htg and Vtg, 13 Chadwick Rd, Westminster Ridge, White Plains, N Y

ZIEBER, W. E. (*M* 1935) Asst Chief Engr (for mail) York Ice Machinery Corp, Roosevelt Ave, and 112 S Penn St, York, Pa

ZIESSE, Karl L. (*A* 1931) Secy-Treas (for mail) Phoenix Sprinkler & Heating Co, 115 Campau Ave, N W, and 315 Hampton Ave, S E, Grand Rapids, Mich

ZIMMERMAN, Alexander H (*M* 1939, *A* 1930) Ventilation Engr, Chicago Board of Health, 707 City Hall, and (for mail) 5449 N St Louis Ave, Chicago, Ill

ZINK, David D. (*M* 1931) Consulting Engr (for mail) 300 W 47th St, Kansas City, and Hickman Mills, Mo

ZITZMAN, Francis T (*S* 1938) Asst Master Mechanic, South Mills, Jones & Laughlin Steel Corp, Aliquippa, and (for mail) 411 Pine St, Beaver Falls, Pa

ZOKELT, C. G. (*M* 1921) Consulting Engr, 3810-24th Ave, S, Seattle, Wash

ZOLITSCH, Harry G. (*A* 1938) Partner (for mail) George J Zolitsch & Son, 123 Grape St, and 392 Sanders Rd, Buffalo, N Y

ZUBER, Otto (*A* 1938) Chief Engr (for mail) Amana Society, Refrig Dept, Amana, and South Amana, Ia

ZUHLKE, W R (*M* 1928) Executive Engr, American Radiator Co, 40 West 40th St, New York, and (for mail) 51 Midland Ave, Yonkers, N Y

ZUMWALT, Ross (*J* 1938) Partner (for mail) Zumwalt & Vinther, 507 Thomas Bldg, and 4115 Travis St, Dallas, Tex

ZUROW, William (*J* 1937) Sales Engr (for mail) St Joseph Railway, Light, Heat & Power Co, 520 Francis St, and 728 South 10th St, St Joseph, Mo

ZWALLY, August L. (*A* 1937) Chief Air Cond Engr, Interstate Electric Co, and (for mail) 908 Elmwood, Shreveport, La

SUMMARY OF MEMBERSHIP

Honorary Members	2	**Associate Members**	800
Presidential Members	21	**Junior Members**	486
Life Members	55	**Student Members**	59
Members	1644	**Total**	3067

LIST OF MEMBERS

(Geographically Arranged)

UNITED STATES *and* ISLAND TERRITORIES

ALABAMA

Birmingham
Cone, W F
Fried, H A
Gause, H C
Hardy, F I
Lichty, C P
Murphree, R L
Trawick, J G

Mobile
Kistler, M I

ARIZONA

Lowell —
Vinson, N L

Phoenix —
Genre, E J
Hummel, G W

Tucson —
Tidmarsh, P M

ARKANSAS

Little Rock—
Cumnock, H
Kellogg, W I
McCoy, C F

Siloam Springs—
Jones, C R

CALIFORNIA

Albany —
Kaup, E O

Alhambra —
Wells, E P

Altadena —
Berringer, S H

Bakersfield—
Baker, H S

Berkeley—
Bentley, C E
Brokaw, G K
Cherry, V H
Harrison, G G
Hutchinson, F W
Peterson, C L
Raber, B F
Woods, B M

Beverly Hills—
Ekings, R M , Jr

Burlingame —
Hill, J A

Culver City —
Owen, J D

Fresno —
Newman, H E

Fullerton
Miles, C N

Glendale
Eggleston, H. L
Orear, A G

Hollywood
Stanley, R L

Lawndale—
Steiner, T J

Los Angeles —
Anderson, C S
Beck, D
Blumenthal, M I
Bullock, H H
Burr, K
Cawby, E L
Cline, E A
Cranston, W E , Jr
Douglas, H H
Downes, A H
Ellingwood, E I
English, H
Fabling, W D
Gabbard, F W
Hendrickson, H M
Hess, A J
Hill, F M
Hogue, W M
Hungerford, I
Kendall, E H
Kennedy, M
Kilpatrick, W S
Lauer, H B
Lehmann, M
Lewis, J C
McKenzie, M C , Jr
Miller, G
Moriarty, J M
Nelson, E L
Ness, W H C
Ott, O W
Park, J F
Phillips, R F
Phillips, R H
Schechter, J E
Scofield, P C
Stewart, W O
Storms, R M
Theobald, A
Weidele, E J

Monterey Park—
Griffith, J B

Oakland—
Cummings, G J
Foote, A G
Masters, R B
Mears, L A
Passur, N A
Williams, C D

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Gayner, J

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Kindorf, O
Porter, N F.
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Sadler, C B

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Cockins, W W
Cooley, E C
Corrao, J
Feyge, H
Folsom, R A
Gee, W W , Jr
Haley, H S
Hickman, H V
Higdon, H S
Holland, R B
Hook, F W
Hudson, R A
Kindorf, H L
Kolb, F W
Kooistra, J F
Krueger, J I
Leland, W E
Marshall, T A
Molfino, P
O Connor, G P
Parker, R A
Peterson, N H
Reed, V C
Shepperd, P D
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Wayland, C E
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White, T J
Woodbury, C D

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Leilich, R K

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Hart, S
Hart, T. S
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Williams, G S.
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Miller, F A
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Seelig, L
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Shultz, E
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Spielmann, G P
Stacy, L D
Stannard, J M
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Sutcliffe, A G
Swanson, N W.
Thinn, C A
Thomas, R H
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Thornton, W B.
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Turner, G. G
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Walters, W. T.
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Weil, M
Weil, M I.
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White, E B
White, R C
Whittington, J. A.
Widdowfield, A S
Williams, D C
Wills, F W
Wilson, W H
Wolff, P P
Woll, W M
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Zimmerman, A H.

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Boyar, S L

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Scholl, H O

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Otis, G E
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Hauer, F
Meyer, F L

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Pruden, B
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Edwards, D. F

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Compton, W E.
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Killian, T J

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Redrup, W D.
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Fillo, F B
Garber, W E, Jr.
Hagedon, C H.
Hayes, J G
Hildreth, E S.
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Parker, P E
Poehner, R E.
Supple, G B.

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Lawrenceburg—
Bechtol, J J

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Westphal, N E.

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Price, C F

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Thrush, H A.

Vincennes—
Wissing, C B.

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Shivers, P F

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Forbes, H B, Jr.
Miller, W T.

IOWA

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Nelson, G O.

Amana—
Foerstner, G C.
Zuber, O.

Ames—
Norman, R A

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Friedline, J M.

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Borg, E H
Campbell, B
Danielson, E H.
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Delavan, N B.
Elbert, B F
Hedeen, L E
Helstrom, C W.
Johnson, T R.
Kimble, C W.
Landes, B E.
LaRue, P.
Marshall, R D.
McReynolds, C. V.
Olchoff, M.
Schnell, R H.
Smith, R A
Spring, C L.
Stiles, G. S
Triggs, F. E.
Vaughn, F. R.
Walters, A. L.

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Brashaw, C J

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Yuska, L J

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Shirley, W. B

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Raven, A H

Waterloo—
Knox, J C
Mitchell, J A
Todd, M L
Winterbottom, R F

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Morrison, W L

Hutchinson—
Mann, A R
Stevens, H L

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Sluss, A H

Neodesha—
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Overland Park—
Robb, J E

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Danielson, L C

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Bachofer, H A, Jr
Ryan, W F

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Droppers, C J

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Danielson, W A

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Stevens, W R

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May, J W
O'Bannon, L S
West, P.

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Grabensteder, L
Graham, J M.
Groot, H. W.
Hellstrom, J
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Cropper, R. O.

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Salzer, A R, Jr
Seidel, G E

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Fitzgerald, W T
Zwally, A L

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Prince, R F

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Fowles, H H

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Fels, A B
Merrill, C J

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McCormack, D
Nest, R E
Posey, J
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Sklarevski, R.
Smoot, T H
Steele, M G
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Vincent, P J
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Goodwin, E W
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Terhune, R D

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Jennings, R A
Jennings, W G
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Spitzley, R L
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Hendriksen, I

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Graff, W F.
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Osberger, T L.
Stafford, T D.
Terrill, M
Todd, S W
Warren, F C
Ziesse, K L

Grosse Pointe—
Buckeridge, V, L

Grosse Pointe Park—
Feely, F J

Hermansville—
Voorhees, G A

Highland Park—
Champlin, R. C
Harrower, W C
Wilde, R S M

Holland—
Leigh, R L

Houghton—
Seeber, R R

Iron Mountain—
Eisele, L G

Jackson—
Gerhard, D H

Kalamazoo—
Brinker, H A
Downs, S H
Hotop, H C
McConner, C R
Metzger, H J
Schlichting, W G
Temple, W J
Wilson, R W

Keego Harbor —
Trzos, O A

Lansing—
Distel, R E
Hill, V H
McLouth, B F
Parsons, R A
Vanderlip, P J

Mt. Clemens—
Bailey, E P

Muskegon—
Young, H J

Muskegon Heights —
Reid, H F

Pleasant Ridge—
Williams, F H

Pontiac—
Singleton, J H

Royal Oak—
Burch, L A
Helmrich, G B
Keyser, H M

Saginaw—
Witheridge, D E

Spring Lake—
Worthing, S L

MINNESOTA

Bayport—
Swanson, E C

Duluth—
Foster, C

Mankato—
Forderbruggen, K J

Minneapolis—
Albrecht, H P.
Algren, A B
Armstrong, R W.
Bell, E F
Bensen, C L
Benson, M L
Betts, H M
Bjerken, M H
Bredesen, B P
Burns, E J
Burritt, C G
Campbell, R L.
Caple, I
Carlson, C O
Chalmers, C H
Comb, F R , Jr.
Cooper, T E
Copperud, E R
Cumming, F J
Dahlstrom, G A.
Davidson, J C
Dovolis, N J
Doxey, H E
Edelman, B P.
Fedders, M P.
Forfar, D M
Francis, P E
Gable, H R
Gausewitz, W. H.
Gerrish, H. E
Gordon, E B., Jr.
Gross, L C
Haley, R T.
Hall, J R
Hanson, L P
Harris, J B
Hawkinson, C F.
Helstrom, H G
Herbacek, E E
Hitchcock, P. C.
Huch, A J
Jordan, R. C.
King, R L
Kingsland, G D.
Knapp, D S
Knowles, E L.
Kuehn, W C
Lange, F F
Legler, F W
Lilja, O L
Lesch, R T.
Lund, C E
Marshall, S C
McDonald, T
Miller, L B
Mills, H C
Morgan, G C
Morton, H S
Newton, A B
Ogard, N L
Orr, G M
Petersen, C P
Proebstle, L
Roberts, H P
Roberts, J R
Rowley, F B
Russell, T W.
Schad, C A
Schernbeck, F H
Schultz, A W.
Seelert, E H
Shipley, S C
Spencer, J B
Stafford, J F.
Stiller, F W
Sturm, W
Sundell, S S
Sutherland, D L
Swanson, D F.
Swenson, J. E.
Uhl, E J
Uhl, W. F
Wallace, H. P., Jr.
Welter, M A
Whitesell, R H.
Willis, L L

Owatonna—
Anderson, G A M

Rochester—
Adams, N D
Plummer, R S

St. Paul—
Anderson, D B
Backstrom, R E
Bauer, A E
Bean, G S
Cook, G E
Cuthbertson, M W.
Diamond, D D
Estes, E C
Fitts, C D
Gausman, C E
Hickey, D W.
Jones, E F
Lynn, R G
McNamara, W.
Mitchell, J G
Oberg, H C
Persson, N B
Ruff, D C
Sanford, A L
Winterer, F C
Wunderlich, M S.

Wayzata—
Heberling, C W.

Winona—
Hamerski, F D

MISSISSIPPI

Jackson—
Light, J C

MISSOURI

Clayton—
Du Bois, L J

Drexel—
Baender, F G

Ferguson—
Szombathy, L R

Independence—
Cook, B F

Jefferson City—
Woodman, L E

Joplin—
McMullen, E W.

Kansas City—
Allen, D M
Arthur, J M , Jr.
Ball, W.
Barnes, A. R
Barnes, H P.
Betz, H D.
Caleb, D.
Cameron, W. R.
Campbell, E. K.
Case, D V.
Cassell, W L.
Chase, L. R.
Clegg, C.
Dawson, T L
Dean, F J , Jr.
Dean, M H
Disney, M A
Dodds, F F
Downes, N W.
Ellis, L W
Farber, L M.
Fehlig, J B
Flarsheim, C A.
Foley, D F
Forslund, O A
Garnett, R E
Gillham, W E
Gould, H E
Harbordt, O E.
Holuba, H J
Kitchen, J H
Lautz, F A
Mahon, C A
Maillard, A L
Marchio, E , Jr
Marston, A D
Matthews, J E
Middleton, H A
Millis, L W.
Moore, B J , Jr
Natkin, B
Nottberg, G
Nottberg, H
Nottberg, H , Jr
O'Dower, H J
Painter, D H
Pellmounter, T
Pellmounter, T V
Pettit, E N , Jr
Pexton, F S
Rivard, M M
Ryan, J B
Sawyer, J N
Sheppard, F A.
Stephenson, L A
Stevens, K M
Weiss, C A
White, H S
Williams, A E
Wright, H H.
Zink, D. D.

Kirkwood—
Hartwein, C E

Liberty—
Kurek, T C

Maplewood—
Curry, R F
Siegel, W A

Mexico—
Badaracco, J A.

Normandy—
Dulle, W L.

Overland—
Grossenbacher, H E

Richmond Heights—
Nelson, C L.
Siegel, D E

Springfield—
James, R E
Karchmer, J. H.

St. Joseph—
Harton, A. J.
Zurow, W.

St. Louis—
Barry, J G , Jr.
Bayse, H. V.
Boester, C F , Jr
Bradley, E P
Carlson, E E
Carter, J H
Clarkson, J R
Cooper, J W
Corrigan, J A
Davis, C R
Dreher, L F
Driemeyer, R C
Evans, B L
Fagin, D J
Foster, J M
Gilmore, L A
Grossmann, H A
Haller, A L
Hamig, L L
Hester, T J
Hoffberger, J P
Hugoniot, V E
Kuntz, E C
Landes, J M
Lang, J C
Langenberg, E B
Laskaris, N G
Laufketter, F C
Malone, J S
Matousek, A G
McIntire, J L
McLarney, H W
McMahon, T W
Miller, J E
Moon, L W
Norris, W P
Oonk, W J
Rodenheiser, G B.
Rosebrough, J S
Rosebrough, R M
Scherrer, L B.
Simons, B C
Sodemann, P
Sodemann, W C B.
Stammer, E L
Sydow, L J
Tenkonohy, R J.
Warren, J S , Jr
Weber, E F
White, E A

University City—
Falvey, J D
Starr, L

Webb City—
Hallar, E V

Webster Groves—
Harbaugh, J W
Myers, G W F
Ronsick, E H

MONTANA

Big Timber—
Strickland, A. W

Billings—
Cohagen, C C
Young, F H , Jr

Great Falls—
Ginn, T M

NEBRASKA

Clarks—
Manning, W M

Columbus—
Ragatz, T. E.

Hastings—
Swingle, W T

Lincoln—
Carlson, C V.
Green, E W
Hellmers, C C, Jr.
Hennessy, W J
King, L D
Lehman, M G
Ross, O J
Shapiro, M M.
Stanton, H W
Tapley, M S
Williams, D D.

Omaha—
Banner, F L D.
Frederick, K C
Goll, W A
Herbert, R M
Kleinkauf, H.
Larkin, P
Leach, L S
Lindberg, A F.
Lvean, L K
Malcolm, B L
Mathis, J
Millard, E L
Moffitt, L C
Olson, M J
Peiser, M B
Reifschneider, J.
Rigby, R A
Rist, L M
Sallander, H A
Schwartz, N E
Solzman, I I
Stanfield, R E
Tracy, W E
Waddington, B C.
White, W R

Scottsbluff—
Davis, O E
Matthews, W M.
Prawl, F E

NEW HAMPSHIRE

Elkins—
Baker, R H.

NEW JERSEY

Arlington—
Bock, B A

Asbury Park—
Strevell, R P

Atlantic City—
Strouse, S. B.

Bayonne—
Schwartz, J.

Belleville—
Thornton, T L.

Bloomfield—
Faust, F H
Harrington, E
McLenegan, D W.
Tenney, D

Bogota—
Griess, P. G.

Camden—
Brown, W M
Coward, C W
Kappel, G W A.
Lanning, E K
Plum, L H
Webster, E K
Webster, W, Jr

Cliffside Park—
Butler, P D

Clifton—
Hilder, F L

Collingswood—
Bolsinger, R C
Mohrfeld, H H.

Dover—
Hedden, W M

East Orange—
Ferguson, R R
Gombers, H B
Maddux, O L
Raymer, W F, Jr.
Reilly, J H
Schroth, A H
Settelmeyer, J T.
Tallmadge, W
Turno, W G W

Elizabeth—
Anderson, J W
Bentz, H
Cornwall, G I.
Faulkner, G
Wheller, H S

Essex Fells—
Soule, L C
Stacey, A E, Jr

Freehold—
Buck, D. T

Glen Ridge—
Jackes, H D

Hasbrouck Heights—
Goodwin, S L

Haworth—
Sharp, J. R.

Irvington—
Feldermann, W
Reinke, A G
Stengel, F J

Jersey City—
Jones, H L
Kunzog, T W
O'Rourke, H. D, Jr
Ritchie, W
Walterthum, J J

Kearny—
Shaffer, C E

Leonia—
Close, R

Lyndhurst—
Ehrlich, M W

Maplewood—
Kepler, D A
Woolley, J H

Merchantville—
Binder, C G
Rohlin, K W

Montclair—
Broome, J H
Scarlett, W J

Morristown—
Roy, A C

Newark—
Albright, C B
Bryant, P J
Carey, P C
Haitmanek, L M
Kruse, W C, Jr
Leinroth, J P
Morehouse, H P.
Ray, L B
Steinmetz, C. W A.

New Brunswick—
Auchmoody, F W.
Wolin, M W

North Arlington—
Bermel, A H
Trambauer, C W.

North Bergen—
Constance, J D

North Plainfield—
Seal, A T

Nutley—
Atkins, T J
Morris, C R

Old Tappan—
Schoeffter, H M

Orange—
Crawford, J. H, Jr

Paterson—
Bannon, L E
Cox, H F
Frank, O E.
Pryor, F L
Wagner, J G

Perth Amboy—
Simkin, M

Plainfield—
Pond, W H.
Tobin, G J

Red Bank—
Mytinger, K L

Ridgefield Park—
Davis, A C

Ridgewood—
Fitts, J. C
Miller, A T
Wallace, D. R.

Roselle—
Snavely, E R

Roselle Park—
Kampish, N. S.

Somerville—
Van Nuys, J C.

South Orange—
Browne, A L

Summit—
Oaks, O O

Teaneck—
Heebner, W M

Westfield—
Scribner, E D

West Englewood—
Gumaer, P W
Steinke, B. J.

Trenton—
Plaag, A F

Union—
Edwards, L V
Lyman, S. E

Union City—
Taverna, F F.

West New York—
Stinard, R L

West Orange—
Adlam, T N

NEW YORK

Albany—
Bond, H A
Dick, A V
Johnson, H S
Lewis, H F
Murray, T F
Rabe, A E
Taggart, R C
Teeling, G A
Westover, W

Bedford Hills
Waters, F A

Bronxville—
Bishop, C R
Groves, S A

Brooklyn—
Addington, H B
Balsam, C P
Bernhard, G
Blackshaw, J L
Campbell, R E
Charles, T J
Daitsh, A
De Somma, A E.
Dutcher, H S.
Dwyer, T F
Fidelius, W R.
Franck, P.
Gates, R A
Goldberg, M
Gornston, M H.
Goulding, W
Hamje, M C
Harsch, R J
Herty, F B
Hollister, N A.
Jalonack, I G.
Janet, H L
Josephson, S
Kadel, G B
Leventhal, B
Lo Piccolo, A A.
Moss, E

Phillips, F W
Potter, J R
Ritchie, E J
Rose, J C
Stalb, J G
Tusch, W
Wachs, L J
Weissblatt, N
Wilson, A

Buffalo—

Beman, M C
Booth, C A
Cherry, L A
Cheyney, C C
Conaty, B M
Currier, C H
Curtis, W A
Davis, J
Day, H. C
Drake, G M
Eisele, W S
Eutsler, E E, Jr
Farnham, R
Farrar, C W
Gifford, C A
Grieves, T R
Harding, L A
Hawk, J K
Heath, W R
Hedley, P S
Hexamer, H D
Hirschman, W F
Holt, W H
Humphrey, L G Jr
Jackson, M S
Jenkins, F H
Kaiser, F
Kamman, A R
Landers, J J
Lenihan, W O
Lighthart, C H
Lockhart, C W
Long, H P
Love, C H
Mahoney, D J
Messenger, T I
Mosher, C H
Pleuthner, R L
Reif, A F
Reif, C A
Rente, H W
Robinson, E T
Roebuck, W, Jr
Ruff, H A
Schafer, H C
Schmidt, H
Seelbach, H
Seelbach, H, Jr
Shelney, T
Strouse, S W
Voisinet, W E
Walker, E R
Wendt, E F
Yager, J J
Zolitsch, H G

Croton—

Elliott, I

Derby—

Ensign, W A

Dobbs Ferry—

Hewett, J B

Elmira—

Davis, B C

Glens Falls—

Hollister, E W

Hastings-on-Hudson

Reynolds, T W

Hyde Park—

Burr, G C

Irvington—

Bastedo, A E

Ithaca—

Elwood, W H
Fairbanks, F L
Frederick, H W
Sawdon, W M
Williams, J W
Woods, E H

Kendall—

Stangland, B F

Kenmore—

Candee, B C
Criqui, A A
Cross, R A
Mollenberg, H I
Snyder, J S
Sommers, W I

Larchmont—

Downe, E R

Lockport

Johnston, R M
Prudden, O D
Saunders, L P

Long Island—

Adler, J C (Forest Hills)
Alt, H L (St Albans)
Apt, S R (Flushing)
Ballman, W H (Long Island City)
Bastedo, G R (Richmond Hill)
Belsky, G A (Valley Stream)
Blackburn, E C, Jr (Garden City)
Bloom, L (Freeport)
Cameron, R T. (Southampton)
Carbone, J H (St Albans)
Dailey, J A (Astoria)
Eastwood, H. F. (Merrick)
Eskell, J E (Astoria)
Fay, D P (Richmond Hill)
Fritz, C V (Freeport)
Gallaway, J F (Kew Gardens)
Graber, E (Douglaston)
Hiers, C R (Great Neck)
Hinkle, E C (Hempstead)
Hoehl, E R (Garden City)
Kaiser, C W (Woodside)
Keller, G A (Wantagh)
Kenney, T W (Hicksville)
Kern, J F, Jr. (Jamaica)
Klenert, W (Flushing)
Knudsen, W R (Jackson Heights)
Landewit, C J (St Albans)
Lane, D D (Elmhurst)
Lang, J (Richmond Hill)
Lewis, C A (Astoria)
MacWatt, D A (Great Neck)
Magee, K B (Astoria)
Magnusson, N (Jamaica)
Maher, T F (St Albans)
Meyer, C L (Hollis)
Olsen, G F (Arverne)
Pabst, C S (Woodhaven)
Pinto, C B (Lawrence)
Pritchard, W J (Jackson Heights)
Raynis, T (Richmond Hill)
Richfield, N H (Floral Park)
Rudd, D J (Babylon)
Schermer, R (Elmhurst)
Schwartz, M (Far Rockaway)
Skidmore, J G (Long Island City)
Spoerr, F F (Jamaica)
Stellwagen, F G (Woodhaven)
Steine, C M (Long Island City)
Thomson, T N (Huntington)
Tucker, F N (Freeport)
Vervoort, E L (Rockville Centre)
Wade, R H (Laurelton)
Wallace, G J (East Elmhurst)
Whittaker, W K (St Albans)

Mt Vernon—

Cribari, H E
Freitag, F G
Northon, L

New Rochelle—

Abrams, A
Farley, W F
Gaylor, W S
Giannini, M C
Lambert, R D
Rose, H J
Terry, M C

New York City—

Adams, E E
Addams, H
Adler, H
Ashley, E E
Baker, H L, Jr
Baker, W H, Jr
Barbieri, P J
Baum, A L
Bearman, A A
Beebe, F E W
Bennett, E A
Berman, L K
Bianculli, V A
Blackman, A O
Blackmore, J J
Bodinger, J H
Bolton, R P
Bond, H H
Bonthron, R C
Borak, E
Brown, D
Buensod, A C
Carpenter, R H
Charlet, L W
Chase, C L
Citron, D J
Clay, W
Cowell, R J
Cucci, V J
Cullin, W W
Daly, R E
Darts, J A
Dauman, A
Davis, C
Davison, R L
Denny, H R
Deterling, W C
Dodge, H A
Donnelly, R
Dornheim, G A
Downs, C R
Duncan, J R
Eadie, J G
Elliott, L B
Engle, A
Erickson, E A
Etlinger, M J
Everetts, J, Jr
Faile, E H
Fay, F C
Feder, N
Feldman, A M
Fenner, N P
Fiedler, H W
Fischer, I
Fitz, J C
Fleisher, W L
Foster, J G
Friedman, M
Gordon, P B
Greenburg, I
Greenburg, L
Hanburger, F W
Hateau, W M
Hechler, S
Heibel, W E
Heller, J A
Henry, A S, Jr
Hering, A
Herkimer, H
Herske, A R
High, J M
Hildreth, L W
Hinrichsen, A F
Hirsch, M H
Hobbie, E H
Hoffman, C S
Holland, G R
Honerkamp, F
Hosking, H L
Hotchkiss, C H B
Hutcheon, C R
Hyman, W M
d'Issertelle, H G
Jacobus, D S
James, J W
Jarcho, M D
Johnston, W H
Kelly, C J
Kessler, J
Kimball, D D
Koehler, C S
Kuhlmann, R

Kunen, H
Kurth, F J
Lawrence, F D
Lucke, C E
Markush, E U
Martens, E D
Martin, G W
Mayette, C E
McClintock, W
McEwan, E E
McLeish, W S
Meinke, H G
Meyer, H C, Jr
Milener, E D
Miller, C A
Miller, J
Montgomery, O C
Morse, F W
Munier, L L
Munkelt, F H
Murphy, J R
Offner, A J
Oldes, W. E
Olson, R G
Olvany, W J
Ortiz, J V
Patorno, S A S
Pihlman, A A
Place, C R
Pohle, K F
Pollak, R
Purinton, D J
Quirk, C H
Raisler, R K
Ramsay, J W
Rather, M F
Reynolds, W V
Ritter, A
Rodman, R W
Rosenberg, P
Rosenthal, E
Ross, J O
Roth, C I
Salter, E H
Samuels, S
Sanbern, F N
Sawhill, R V
Schoepflin, P H
Schulze, B H
Scott, A F H
Sears, C B
Seelig, A E
Sellman, N T
Senior, R L
Siebs, C I
Simpson, A M
Sklenarik, L
Smith, M S
Sternberg, E
Still, F R
Strock, C
Strumn, J
Syska, A G
Tiltz, B E
Timmis, W W
Torrance, H
Tyler, R D
Vetlesen, G U
Wagner, F H, Jr
Wadsworth, R H
Walker, K
Wallace, G N
Walton, C W, Jr
Waring, J M S
Wasson, R A
Wheeler, J, Jr
Whitelaw, H L
Wilder, H P
Willner, I
Wolf, P
Wrightson, W F

Niagara Falls—
Kessler, M E
Woolcock, E

North Tarrytown
Weiss, A P

North Tonawanda—
Spencer, W E

Oswego –
Mohn, H I

Pelham Manor—
Peacock, J K
White, E G

Port Chester –
Scott, G M

Rochester—
Andresen, G C
Betlem, H I
Cook, R P
Dickason, G D
Eschenbach, S P
Hakes, L M
Hutchins, W H
Lee, R T
Leonhard, I W
Stacy, S C
Treadway, O
Vidale, R
Wilder, F I

Rome -
Lynch, W I

Rye—
Crone, I I

Scarsdale -
Cumming, R W
Ullman, H G

Schenectady –
Welch, L A, Jr

Schodack Landing –
Freas, R B

Scotia -
Hunziker, C E

Snyder—
John, V P
Madison, R D

Staten Island—
Callahan, P J
(Great Kills)
Connell, H
(Mariners Harbor)
Frimet, M
(West Brighton)
Ghose, K N
(New Dorp)
Johnson, E B
(Port Richmond)
Perina, A E
(Port Richmond)
Pfuhler, J L
(W New Brighton)
Pietsch, J A
(New Brighton)
Ruggles, R F
(Randall Manor)
Vivarttas, E A
(West Brighton)
Volkhardt, A N
(Stapleton)
Vroome, A E
(Prince Bay)
Waechter, H P
(Tompkinsville)
Wood, R A
(W New Brighton)

Suffern—
Barnum, M C

Syracuse—
Acheson, A R
Ashley, C M
Cady, E F
Carrier, E G
Carrier, W H
Cherne, R E
Day, V S
Dee, L H
Des Reis, J F
Driscoll, W H
Dunne, R V D
Evans, E C
French, D
Graham, W D
Grant, W A
Hockensmith, F E
Ingels, M
Lewis, L L
Licandro, J P
Lyle, J I
Schulz, E L
Shanklin, A P
Sheldon, N E
Taliaferro, R R
Traynor, H S
Woese, C F

Tonawanda -
Karlsteen, G H

Tuckahoe -
Brabbée, C W

Utica—
Knapp, J H
Sherbrooke, W A
Steinhorst, T F

Valhalla—
Meline, C A

White Plains -
Baker, T
Ruple, P E
Zibold, C E

Williamsville—
Stevens, A I

Yonkers -
Dean, D
Goerg, B
Harmonay, W I
Kelly, J G
Rainger, W F
Walther, F G
Zuhlke, W R

Yorktown Heights—
Gitterman, H

NORTH CAROLINA

Charlotte –
Brandt, E H, Jr
Hill, H H
Hodge, W B
Muirheid, J G
Petty, C E

Durham—
Cooke, T C
McDowell, H L
Wallace, W M, II

Greensboro—
Harding, E R

High Point—
Gray, W E

Raleigh—
Rice, R B
Vaughan, L L

Winston-Salem—
Bahnson, F F
Brown, M D
Cornwall, C C
Kaczenski, C
Page, A
Warren, R M, Jr

OHIO

Akron—
Shields, C D

Ashland—
Rybolt, A I

Chagrin Falls—
Southmayd, R T

Cincinnati—
Bird, C
Boyd, T D
Coombe, J
Cramer, W G
Donelson, W N
Du Chateau, M F
Edwards, A. W
Fenker, C M
Green, W C
Hard, A L
Helburn, I B
Houlis, L D
Houliston, G B
Hudepohl, L F
Hust, C E
Jennings, H K
Junker, W H
Kiefer, C J
Kilday, J A
Kinney, A M
Kramig, R E, Jr
Leupold, G L
Mason, G C
Mathewson, M E
Mayer, R W
Mayne, W L
Mills, C A
Moore, H W
Pillen, H A
Pistler, W C
Powers, L G
Richard, E J
Royer, E B
Ruff, A G
Sigmund, R W
Silberstein, B G
Smith, S
Sproull, H E
Suthn, G V
Thompson, E B
Ward, F J
Washington, L W
Winstel, F E
Winther, A
Wright, K A

Cleveland—
Baggaley, W
Beach, W R
Cary, E B.
Cheeseman, E W
Cohen, H
Cohen, P
Conner, R M
Curtis, H F
Cushing, C F

Eveleth, C F
Ferguson, J H.
Friedman, A
Gayman, P D.
Gottwald, C.
Gray, E W
Harris, J G
Harvey, L C.
Heisterkamp, H. W.
Jones, J P.
Kaercher, C M. H.
Kain, E. M
Kitchen, F A.
Klie, W
Kofoed, V B
Levy, M I
Marotta, J A
Martinka, P D.
McKeeman, C A.
Moore, W R
Nachman, G P.
Pogalies, L H
Quick, B A
Rowe, W M
Schurman, J A.
Slawson, L E
Smith, W D.
Taze, D L.
Tuve, G L.
Vanderhoof, A L.
Walker, J E
Washburn, M J.
Webb, E C
Weldy, L O
Wetzell, H E.
Wilhelm, J E.
Wootan, C
Young, E O.

Cleveland Heights—
Davis, R G
Rhoton, W R
Sogg, A
Sterling, J G, Jr
Wright, D K, Jr

Columbus—
Allonier, H R
Breneman, R B.
Brown, A I.
Cross, R. C
Denise, J R
Myler, W M, Jr.
Oelgoetz, J F.
Peck, H E
Poling, D B
Sandfort, J F.
Sherman, R A.
Smith, J A
Somers, W S
White, E D
Williams, A W.
Wyatt, D H.

Cuyahoga Falls—
Humphrey, D. E.
McElhaney, G. W.

Dayton—
Baker, I. C
Buenger, A
Chapman, W. A., Jr
Fuller, E W
Gibbons, M J.
Godfrey, J E
Gonzalez, R A.
Hansler, J E.
Hauck, E L
Hubbuch, N J.
Hull, H B
Hutchings, R. L.
Jones, H L
Kucher, A A.
Kuempel, L. L.
La Salvia, J J.
Lewis, C E
Livar, A P.
MacMillan, A R
Nessell, C W
Smith, N J
Wood, C F
Woodward, R.
Worsham, H
Wyld, R. G.

East Cleveland—
Geltz, R W.
Nobis, H M.
Stark, W E
Steffner, E. F.

Elyria—
Kalinsky, A G.
Maynard, J. E.

Goshen—
Doyle, W J.

Hamilton—
Thomas, L. G L.

Hudson—
Follett, T L

Lakewood—
Kubasta, R W.
Longcoy, G B
Norrington, W L.
Tanker, G E
Teckmyer, F C, Jr
Weatherby, E P, Jr.
Wilson, R. A.

Lima—
Hawisher, H H

Middletown—
Byrd, T

Newark—
Simison, A L
Slayter, G
Waggoner, J H.

Norwood—
Braun, J J
Motz, O W.

Oberlin—
Ries, L S

Painesville—
Hobbs, J. C.

Parma—
Kajuk, A E.

Piqua—
Lange, R T.

Sandusky—
Rathke, A C

Shaker Heights—
Avery, L. T.
Foley, J L.
Veale, T.

South Park—
Barney, W. E.

Springfield—
Noble, J. P.

Toledo—
Baker, H C
Bergan, J R
Jones, S
Mabley, L. C
McKitrick, W D
Summers, C G
Watkins, G B

Waverly—
Armbruster, F T W

Wilmington—
Marconett, V G.
Sapp, C L.

Worthington—
Slemmons, J D.

Youngstown—
Montgomery, J. R.

OKLAHOMA

Alva
Husky, S T

Houston—
Putnam, N J

Norman—
Dawson, E F
Giles, J C.

Oklahoma City—
Campbell, A O
Carnahan, J H.
Carroll, W M
DeVilbiss, P T.
Dolan, R G
Feldstein, H.
Gray, E W.
Holyfield, E F
Hoppe, A A
Howlett, I G
Johnson, R F
Kidd, C R
Kroeker, S P
Lockwood, G E
Loeffler, F X
Middleton, D K
Mideke, J M
Miller, B R
Morin, A R.
Rolland, S L
Schoenhofen, L H, Jr
Spencer, D
Tiller, L
Woodworth, C. Z.

Tulsa—
Dean, C H
Holmes, A D
Irwin, R R
Jones, E
Meinholtz, H W.
Pauling, R E
Scherrer, K C
Shoemaker, F. F.

OREGON

Corvallis—
Willey, E. C.

Medford—
Hoey, J. K.

Portland—
Enders, C E
Farnes, B W
Freeman, J A.
Heinkel, C E
Kroeker, J D
Taylor, T E

PENNSYLVANIA

Abington—
Bigelow, E S
Park, N W

Aldan—
Mulcey, P A.

Allentown—
Goundie, J K.
Hersh, F C
Korn, C B

Ambler—
McElgin, J W.

Ardmore—
Haynes, C V.

Ardsley—
Tucker, L A

Aspinwall—
Lewis, K C.

Bala-Cynwyd—
Patrick, H M
Whitney, C W.

Beaver Falls—
Zitzman, F T

Beechwood—
Kipe, J M
Murdoch, J P.

Bellevue—
Allen, W. A.

Bethlehem—
Murnin, E A, Jr.
Stuart, M C

Bradford—
Cleveland, C C.
Paterson, F C, Jr.
Presdee, C W.

Brookline—
Donnelly, M A.

Charleroi—
Sutherland, F. A.

Clearfield—
Gault, G W.

Drexel Hill—
Matz, G N
Stokes, A. D.

Dunbar—
Sherwood, L. T.

East Pittsburgh—
Hazlett, T. L.
Penney, G. W.

Elizabeth—
Reed, V A, Jr

Elizabethtown—
Dibble, S E

Erie—
Joyce, H B
Preece, L W
Sahlmann, F L

Glenmoore—
Gant, H P.

Greensburg—
Burkhart, E M

Harrisburg—
Eichler, H C
Geiger, I H
Hedlund, R. A

Haverford—
Black, E N, 3rd

Hershey—
Snavely, A B

Johnstown—
Hunter, L N
Knowles, F R
Novotney, T A

Kingston—
Macdonald, D B.

Lancaster—
Jones, A.
Lloyd, E C.
Shorb, W. A.

Lansdowne—
Hall, M S
James, H R.
Lauer, R F.
Mawby, P
Mayer, R L.

Manheim —
Weitzel, C B
Weitzel, P H

McKeesport—
Dugan, T M

Middletown—
Locke, R A.

Midland—
Crichton, H C.

Narberth—
Dever, H F.
Searle, W. J., Jr

New Castle—
Andrews, G H
Sonneborn, C.

New Kensington—
Edwards, J D

Newtown—
Lewis, T.

Norristown—
Hucker, J H
Mirabile, J J.

Oxford—
Ware, J H, III

Penn Valley—
Smith, W F

Philadelphia—
Ahlff, A A
Arnold, R S
Bachman, F
Barnard, M E
Bartlett, C E
Black, H G
Blankin, M F
Bogaty, H S
Bornemann, W A
Bozeman, R
Bulkeley, C A
Caldwell, A C
Call, J
Cassell, J D
Childs, L A
Cody, H C
Culbert, W P
Dafter, E H
Dambly, A E.
Davidson, L C
Davidson, P L
Dietz, C F
Dome, A G
Donovan, W J
Eastman, C B
Eckart, J H
Elliott, E
Erickson, H H
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Familetti, A R
Flum, J C
Franklin, S D
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Gillett, M C
Gilman, F W
Goff, J A
Guler, G D
Hackett, H B
Hajek, W J
Hedges, H B
Hibbs, F C
Hunger, R F
Hynes, L P
Ickeringill, J C
Jacobsen, K C S
Jakoby, A C
Jardine, W H, Jr.
Kelbie, F R
Killough, R E
Kirkbride, J O
Kriebel, A E
Ladd, D
Landau, M
Leopold, C S
Lyon, P S
Mack, L
Makin, H T, Jr
Martocello, J A
Mather, H H
McClintock, A, Jr
McCullough, H G.
McIlvaine, J H.
Mellon, J T J
Meloney, E J
Mensing, F D
Millham, F B
Moody, L E
Morgan, R C
Morris, A M
Munro, G A
Nesbitt, A J
Nesbitt, J J.
Newcomb, L. B.
Nusbaum, L
Parent, H. M
Pfeiffer, F. F
Plewes, S E
Powell, G. W., Jr.
Powers, E C
Prybil, P L
Rank, A I
Redstone, A L
Reilly, B B
Reilly, C E
Rettew, H F
Roberts, H L
Rugart, K
Sabin, E R
Schneider, C H
Seltzer, P A
Shapiro, C A
Sheffler, M
Speckman, C H
Sullivan, C J
Timmis, P
Touton, R D
Traugott, M
Trenner, K
Trump, C C
Tuckerman, G E
Wagner, E K
Wegmann, A
Weid, H L
Werner, J G
Wiley, D C
Wilmot, C S
Wilson, F J
Wiltberger, C F
Woolston, A H
Yerkes, W L

Pittsburgh—
Allensworth, J E.
Beighel, H A
Blackmore, G C.
Brauer, R
Bushnell, C D
Canon, H A
Carr, M L
Caskey, L H, Jr
Collins, J F S, Jr
Comstock, G M
Cost, G W
Dickinson, R P, Jr.
Dickson, R W, Jr.
Dorfan, M I
Eckstein, J E
Edwards, P. A
Eils, L C
Ellis, G P
Ferderber, M B
Gallagher, F H
Greiner, G E, Jr
Griest, K
Hathaway, C B
Hecht, F H
Heilman, R H
Houghten, F C
Humphreys, C M.
Hyde, E H
Kennedy, O A
Kimmell, P M
Kirkendall, H J
Landes, B D
Lifshitz, H
Loucks, D W
Lowe, W
Maehling, L S
Mahon, F B
Marshall, A W
McGonagle, A
McIntosh, F C
McLean, J E
Miller, R A
Moore, H L
Mueller, J E
Nass, A F
Nicholls, P
Park, H E
Parks, C E
Proie, J
Reed, I G
Reed, W H, III
Riesmeyer, E. H, Jr.
Rockwell, T. F.
Rose, H J.
Scanlon, E S
Schmieler, J B
Selig, E T, Jr.
Shore, D.
Small, B R
Smyers, E C
Speller, F N
Stanger, R B
Steggall, H B
Stevenson, W. W.
Strauch, P C
Tennant, R J J.
Tower, E S
Turk, L G
Waters, G G
Weddell, G O
Zangrilli, A J

Pottsville—
Marty, E O
Smith, J D

Primos—
Johnson, A. J

Reading—
Luck, A W.

Rochester—
Pugh, D C

Scranton—
Mahon, B. B

Sewickley—
Lore, H E.

Springdale—
Lynn, F E.

Springfield—
Grossman, H. E
Payne, R E.

State College—
Queer, E R

Stroudsburg—
Kiefer, E J, Jr

Swarthmore—
Hobbs, W S
Krayenhof, H G
Robinson, A S.
Thom, G. B

Tarentum—
Orr, L

Uniontown—
Marks, A A.

Upper Darby—
Blackmore, J S
Conroy, M J
McClain, C H
Morehouse, J. S

Villa Nova—
Barr, G W.
Carey, J A

Washington—
Frazier, J. E.

Wilkes-Barre—
Stewart, J. P.

Wilkinsburg—
Biber, H A
Campbell, T F

Williamsport—
Axeman, J E

Wormleysburg—
Miller, T G

Wyncote—
Buck, L

York—
Aughenbaugh, H E
Barnum, W E, Jr
Hertzler, J R
Kartorie, V T
Kimmel, W G
Walsh, E R
Zieber, W E

Zelienople—
Grabman, H B

PHILLIPINE ISLANDS

Manila—
Hausman, L M
Macrae, R B

RHODE ISLAND

Pawtucket—
Arden, I L.
Kramer, C.

Providence—
Blanding, R L
Coleman, J B
Gibbs, E W
Hartwell, J C
McCarthy, J J
McLaughlin, J D
Moulder, A W

Warren—
Bowen, H C

SOUTH CAROLINA

Clemson—
Shenk, D H

Columbia—
Hartin, W R, Jr
Kerr, W E
Mercer, C F
Reamer, W S, Jr

SOUTH DAKOTA

Lead—
Pullen, R. R

Sioux Falls—
Monick, F. R

TENNESSEE

Chattanooga—
Campbell, G. S.

Elizabethton—
Torok, E

Knoxville—
Cross, F G
McCain, H K
Oakley, L W

Memphis—
Campbell, A Q, Jr
D'Imor, E J
Flinn, G S
Hoshall, R H

Nashville—
Armistead, W C
Baker, W C
Brown, F
Crane, R S
Wilson, V H

TEXAS

Amarillo—
Burnett, E S
Houska, A D

Austin—
Degler, H E
Gossett, A L

Bryan—
Griesser, C F

College Station—
Badgett, W H
Cook, A L
Giesecke, F E
Hines, G M
Hopper, J S
Long, W E
Smith, E G

Dallas—
Anspacher, T H
Blum, H, Jr
Bock, I I
Brown, M W
Campbell, E K, Jr
Cheatwood, W H
Durkee, M E
Farrow, E E
Gilbert, L S
Huff, J M
Jelinek, F R
Kribs, C L, Jr
Landauer, L L
Martyn, H J
Mehl, O H
Moler, W H
Pines, S
Renouf, E P
Rodgers, F A
Rogers, R C
Schmidt, H I
Schuett, D F
Stokes, A
Taylor, R B
Turner, H S, Jr
Ullrich, A B, Jr
Zumwalt, R

Fort Worth—
Gardner, C. R.
Harris, A M
Harrison, J C
MacEachin, G. C.
Skinner, H W.
Sprekelmeyer, J. M
Werner, R. K.

Galveston—
Ruemmele, A M

Houston—
Banowsky, A B
Barnes, A F
Chase, A M, Jr
Chrone, R E
Closner, J J
Cochran, W B
Constant, E S
Cooper, D S
Earl, W
Drescher, F E
Ibison, J L
Keeland, B W
Kiesling, J A
Koch, A C
Kurtz, R W
McKinney, C A
Mitchell, J
Morrow, J D
Olson, G E
Rollosson, J A
Rowe, I E
Salinger, R J
Spencer, R M
Taylor, R F.
Walsh, J A
Witmer, C N

Kingsville—
Richtmann, W M

Port Arthur—
Shaw, C G

San Antonio—
Billingsley, O F
Diver, M L
Ebert, W A
Kotzebue, R W
Pawkett, L S
Rhine, G R
Rummel, A J
Saunier, W C
Stoltz, G C

Waco—
Benham, F C, Jr

UTAH

Salt Lake City—
Richardson, H G
Young, J T, Jr

VERMONT

Burlington—
Lanou, J E
Raine, J J

North Ferrisburg—
Breckenridge, L. P

VIRGINIA

Arlington—
Ferrarini, J
Marshall, W D.
Nye, L B, Jr.
Rudio, H M
Walz, G R
Welsh, H A

Blacksburg—
Johnston, R McC

Lynchburg—
Doering, F L
Franklin, S H, Jr

Norfolk—
Capps, E L
Huybert, L E, Jr
Nowitzky, H S
Thomas, R C
Webster, W H, Jr

Portsmouth—
Stubbs, W C

Richmond—
Belding, H H
Campbell, F B
Carle, W
Hinnant, C H, Jr
Johnston, J A
Peebles, J K, Jr
Pelouze, H L, II
Schulz, H I

Roanoke—
Bailey, A E, Jr
Nininger, C H

WASHINGTON

Bremerton—
Bysom, L I

Kent—
Boyker, R O

Port Orchard—
Pratt, F J

Seattle—
Beggs, W F
Bouillon, I
Case, R H
Cox, W W
Eastwood, F O
Granston, R O
Griffith, H T
Hauan, M J
Mallis, W
May, C W
Morse, R D
Musgrave, M N
O'Connell, P M
Peterson, S D
Pollard, A L
Sparks, J D
Twist, C F
Veltman, B M
Ward, J J
Watt, R D
Weber, E L
Wesley, R O
Zokelt, C G

Spokane—
Russell, W B

Tacoma—
Foote, E E
Norby, K H.
Spofforth, W.

Yakima—
Leichnitz, R. W
McCune, B. V

WEST VIRGINIA

Charleston—
Ralph, D S
Rosenblatt, A M
Rothmann, S C
Shanklin, J A
Wright, M B

Fairmont—
Lonry, R C
Wright, C F

Huntington—
Johnson, I O

Largent —
Donnelly, T A

Wheeling—
Hitt, J C

WISCONSIN

Appleton—
Eisele, D E

Beloit—
McKinley, C B

Eau Claire—
Grosvold, F F

Clintonville—
Quall, C O

Kohler—
Hvoslef, F W
Kohler, W J, Jr

La Crosse—
Anderegg, R H
Bowen, J C
Miller, M W
Rowe, W A
Ryan, H J
Thomas, N A
Towne, C O
Trane, R N

Madison—
Bledsoe, R P
Dean, C L
Feirn, W H
Hall, G
Larson, G L
Nelson, D W
Seymour, J E
Tuthill, A F
White, J C

Milwaukee—
Alfery, H F
Allan, W
Banks, J B
Bernert, L A
Boden, W F
Bowers, A F
Brown, W H
Buller, C R
Carroll, A F
Cooper, W B
Cutler, J A
Davis, K T
Elliot, N B
Ellis, H W
Errath, E O
Frentzel, H C
Gerstenberger, E J
Goldsmith, F W
Gregg, S H
Griewisch, A H
Haerle, R A
Hamacher, K F
Hanley, E V
Haupt, H F
Haus, I J
Hays, C A
Hessler, L W
Hoffmann, A
Hughey, T M
Jackson, C H
Jones, E A
Jung, J S
Ketter, J W
Knab, E A
Koch, R G
Krenz, A S
Lingen, R A
Lofte, J A
Mack, E H
McKee, J W
Meyer, K A
Miller, C W
Mueller, H P
Noll, W F
Page, H W
Podolske, A R
Randolph, C H
Reinke, I F
Rice, C J
Schmid, J U
Schreiber, H W
Shawlin, W C
Shodron, J G
Spence, M R
Swisher, S G, Jr
Szekely, E
Trostel, O A
Volk, J H
Weil, F H E
Weimer, F G
Winkler, R A

Neenah—
Angermeyer, A H
Eiss, R M
Stiegler, A J

Oconomowoc—
Becker, W A

Racine—
Dixon, A G
Kluge, B M
Menden, P J

South Milwaukee—
Ouweneel, W A

West Allis—
Erickson, M E
Spence, R T

CANADA

Brandon, Man —
Yates, J E
Yates, J E, Jr

Calgary, Alta —
Vissac, G A

Edmonton, Alta —
Mould, D E

Flin Flon, Man —
Foster, P H

Freeman, Ont —
Goodram, W E

Galt, Ont —
Sheldon, W D, Jr

Halifax, Nova Scotia
Meagher, A T

Hamilton, Ont —
Barnes, H
Charters, W A
Dickenson, M E
Moffat, O G

Hampstead, P. Q —
Montgomery, E G

Islington, Ont —
Wilson, G T

Kirkland Lake, Ont
Calver, R W

Kitchener, Ont —
Beavers, G R
Pollock, C A

Lindsay, Ont —
McCrae, G W

London, Ont.—
O'Flaherty, J G

Montreal, P Q —
Allaire, L
Armstrong, W J
Ballantyne, G L
Barnsley, F R
Baxter, W E
Berridge, W W
Bews, J
Boland, R O
Chenevert, J G
Colle, S S
Darling, A B
Dixon, M F
Dufault, F H
Dupuis, J R
Dykes, J B
Elliot, G B
Emerson, J J
Forrester, N J
Freeman, E M
Friedman, F J
Garneau, L
Gendron, H
Gittleson, H
Hamlet, F A
Hamlet, T F
Hughes, H R
Hughes, W U
Johnson, C W
Jungbluth, E N
Keith, J P
Laffoley, L H
Lafontaine, E A
LaMontagne, A F.
Linton, J P
MacLachlan, V D
Madely, F J
Marshall, A G
Martin, L
Martin, R
Milne, A H
Morris, J A
Murray, H G S
Nathan, P V
Nickle, A J
Noyes, R R
Osborne, G H
Peart, A M
Perras, G E
Phipps, F G
Plant, E B.
Pratt, J C
Robertson, J A M
Roche, I F
Ross, J D
Sampson, E T
Shaw, J A
Spark, W
Standring, R A
Ste-Marie, G P
Timmins, W W
Tolhurst, G C
Twizell, E W
Vollmann, C W
Walford, L C A
Watts, A E
Wiggs, G L
Wilkinson, A
Wilson, A
Wormley, R F
Worthington, T H

Ottawa, Ont.—
Allen, A W
Colclough, O T
Gray, G A
McGrail, T E
Pennock, W B

Preston, Ont.—
Everest, R H
Wood, A W

Quebec, P. Q —
LaRocque, P E
Paquet, J M
Roy, L

Sackville, N. B. —
Rand, F R

St Lambert, P. Q —
Lefebvre, E J

Three Rivers, P. Q —
Germain, O

Timmins, Ont.—
Smith, R J

Toronto, Ont —
Abbott, T. J
Alexander, S W
Allcut, E. A
Allsop, R P
Angus, H H
Anthes, L L
Arrowsmith, J O
Baker, G R
Baker, L P
Bishop, J W
Blackhall, W R
Bowerman, E L
Brittain, A, Jr
Cairns, J H
Carter, A W
Chambers, F W
Church, H J
Clifton, J A
Cole, G E
Davis, E J
Daynes, J H.
Dickey, A J
Dion, A M
Dowler, E A
Duncan, W A
Eaton, W G M
Ellis, F E
Ewens, F G
Fear, S L
Fitzsimons, J P

Foley, J J
Forrester, C M.
Fox, E
Fox, J H
Gauley, E R
Givin, A W
Gordon, C W
Gordon, W D
Gurney, E H
Gurney, E R
Harrington, C
Henion, H D
Hill, H G
Hills, A H
Hopper, G H
Hughes, L K
Jenney, H B
Jennings, S A
Jones, A T
Kelly, W C
Lawlor, J J
Ledgett, F D
Leitch, A S
Libby, R S
Lower, H C.

MacDonald, D J
Marriner, J M S
Mathison, R S
Maxwell, R S
McKerlie, J
McLaren, T H
Moore, F C
Moore, H S
Morgan, A S
Nearingburg, A.
Oke, W C
O'Neill, J W
Paul, D I
Philip, W
Playfair, G A
Price, D O
Ritchie, A G
Roth, H R
Shears, M W.
Smith, G E
Stencel, R A
Stott, F W
Sturdy, O C
Tasker, C
Thomas, M F.

Thomsen, N B
Treleaven, H M
Waldon, C D
Wardell, A
Watson, M B
Willer, M D
Woollard, M S
World, H P

Vancouver, B. C.—
Hale, F J
Johnston, R E
Leek, C W
Leek, W
McCreery, H J
Turland, C H

Victoria, B. C.—
Sheret, A

Westmount, P. Q.—
Colford, J

Winnipeg, Man.—
Argue, E. J
Avery, L
Charles, P L
Davis, G C
Eade, H R
Glass, W
Hinton, R P
Jones, B G
Kent, R L
Kipp, T
McDonald, I
Michie, D F
Miller, E R
Munn, E F
Price, E H
Steele, J B
Summers, E T
Thompson, F
Watson, H D
Worton, W

Woodstock, Ont.—
Karges, A

AUSTRALIA

Melbourne—
Atherton, A E
Ross, R

Sydney—
Davey, G I
Henderson, A S
Hunt, N P
Moloney, R R
Picot, J W
Robinson, J A
Sands, C C

BELGIUM

Brussels—
Lebrun, P
Mautsch, R

BERMUDA

Hamilton—
Kitchen, W H J

BRAZIL

Rio de Janeiro—
Botelho, N J
Darby, M H

BRITISH WEST INDIES

Trinidad—
Cox, T M , Jr.

CHINA

Hong Kong—
Bradford, G. G.

Shanghai—
Chen, S T
Doughty, C J
Gange, F B
Hart-Baker, H W
Kwan, I K
Loh, N S
Morrison, C B
Rachal, J M
Waung, T F
Wong, W S B

Tientsin—
Loo, P Y

CUBA

Havana—
Colmenares, G V

DENMARK

Copenhagen—
Reck, W E
Schulein, E H

EGYPT

Alexandria—
Tallianos, P C.

Cairo—
Ezz El Din, K
Henszey, W P

ENGLAND

Birmingham—
Richardson, R D

Cheshire—
Adshead, B

Essex—
Symonds, E S

Kent—
Figgis, T G
Lipscombe, H W J

Lancaster—
Battley, H E

Leeds—
Jennins, H H

Liverpool—
Thomas, A E

London—
Bailey, W M
Benham, C S K
Bird, G L H
Butt, R E W
Chester, T
Faber, O
Fraser, J J
Greenland, S F
Haden, G N
Herring, E
Jackson, G R
Kraminsky, V
Lincbaugh, J E
Nobbs, W W
Pryke, J K M
Troup, J D

Manchester—
Webb, J W
Yates, W

Middlesex—
Case, W G
Gill, E F

St Albans—
Carter, D

Surrey—
Casperd, H W H

Trowbridge—
Haden, W N.

Westminster—
Russell, J. N

Wolverhampton—
Tyson, W H

FRANCE

Dijon—
Bur, J R C

Lille—
Neu, H J E

Lyon—
Goenaga, R C

Paris—
Beaurrienne, A.
Bodmer, E
Modiano, R
Nessi, A
Schmutz, J
Waudby, W

Vanves—
Ghilardi, F

FRENCH INDO CHINA

Tonkin—
Cox, P E

GERMANY

Berlin—
Brust, O
Schmidt, E G

Hamburg—
Brandi, O. H

Stuttgart—
Klein, A. R.

HOLLAND

Amsterdam—
ter Weeme, A

The Hague—
DeWilde, M P

INDIA

Bombay—
Heard, J A E
Wilson, E D.

Burma—
Horner, S D.

Calcutta—
Stromgren, S G.

IRAQ

Alwiyah—
Roos, E B J

IRELAND

Cork—
Barry, P I

Dublin—
Leonard, L C G

ITALY

Milan—
Dell'Orto, L
Gini, A
Hauss, C F
Marzorati, G
Parrilli, R

Torino—
Baldi, G

JAPAN

Osaka—
Fukui, K

Tokyo—
Kitaura, S
Kozu, T
Saito, S
Sekido, K.

JAVA (D. E. I.)

Soeradaja—
Thornburg, H A

MANCHOUKUO

Hsinking—
Kawase, S

MEXICO

Mexico, D. F.—
Gilfrin, G F.
Huber, E
Martinez, J J

NETHERLANDS

Arnhem—
Tanger, O C F

Breukelen—
Teves, H L

NEW ZEALAND

Auckland—
Manning, C E

Christchurch—
Taylor, E M
Vale, H A L

Macandrew Bay—
Davies, G W

NORWAY

Oslo—
Tjersland, A.

Stabekk—
Altsen, N

ROUMANIA

Bucharest—
Wasser, M

SCOTLAND

Angus—
Knox, J R

SOUTH AFRICA

Durban—
Gordon, H H W.
Kothe, F H.

Johannesburg—
Boxall, F
Ehlers, J
Overton, S H
vonChristierson, C A.

SOUTH AMERICA

Venezuela—
Blas, R J

SPAIN

Madrid—
Altageme, B
Jimenez, J. G.

STRAITS SETTLEMENTS

Singapore—
Faxon, H C.
Hill, C. F.

SWEDEN

Lidingo—
Rosell, A. F

Stockholm—
Eklund, K G
Gille, H B
Ostrom, E W.
Theorell, A T
Theorell, H G T.

TURKEY

Istanbul—
Karakash, T J

UNITED STATES AND ISLAND TERRITORIES

Alabama	8
Arizona	4
Arkansas	4
California	117
Colorado	10
Connecticut	37
Delaware	15
Dis. of Columbia	66
Florida	12
Georgia	33
Hawaii	2
Illinois	275
Indiana	32
Iowa	36
Kansas	13
Kentucky	15
Louisiana	12
Maine	6
Maryland	38
Massachusetts	109
Michigan	186
Minnesota	115
Mississippi	1
Missouri	136
Montana	4
Nebraska	42
New Hampshire	1
New Jersey	107
New York	455
North Carolina	18
Ohio	188
Oklahoma	36
Oregon	8
Pennsylvania	288
Phillipine Is	2
Rhode Island	10
South Carolina	5
South Dakota	2
Tennessee	14
Texas	85
Utah	2
Vermont	3
Virginia	25
Washington	32
West Virginia	10
Wisconsin	94
	2713
DOMINION OF CANADA	220

FOREIGN COUNTRIES

Australia	9
Belgium	2
Bermuda	1
Brazil	2
British W. Indies	1
China	12
Cuba	1
Denmark	2
Egypt	3
England	33
France	10
Fr. Indo China	1
Germany	4
Holland	2
India	4
Iraq	1
Ireland	2
Italy	6
Japan	5
Java (D. E. I.)	1
Manchoukuo	1
Mexico	3
Netherlands	2
New Zealand	4
Norway	2
Roumania	1
Scotland	1
South Africa	6
South America	1
Spain	2
Str't. Set'ments.	2
Sweden	6
Turkey	1
	134
Total Membership	3067

PAST OFFICERS

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1894

President	Edward P Bates
1st Vice-President	Wm M Mackay
2nd Vice-President	Wiltsie F Wolfe
3rd Vice-President	Chas S Onderdonk
Treasurer	Judson A Goodrich
Secretary	L H Hart

Board of Managers

Chairman, Fred P Smith

Henry Adams — A A Cary
Hugh J Barron — James A Harding
Edward P Bates *Pres* — L H Hart, *Secy*

Council

Chairman, R C Carpenter

Albert A Cryer — Chas W Newton
F W Foster — Ulysses G Scollay *Secy*

1895

President	Stewart A Jellett
1st Vice-President	Wm M Mackay
2nd Vice-President	Chas S Onderdonk
3rd Vice-President	D. M Quay
Treasurer	Judson A Goodrich
Secretary	L H Hart

Board of Managers

Chairman James A Harding

Geo B Cobb — Ulysses G Scollay
Wm McMannis — B F Stangland
Stewart A Jellett, *Pres* — L H Hart, *Secy*

Council

Chairman, R C Carpenter

Henry Adams — T J Waters
Edward P Bates — Albert A Cryer, *Secy*

1896

President	R C Carpenter
1st Vice-President	D M Quay
2nd Vice-President	Edward P Bates
3rd Vice-President	F W Foster
Treasurer	Judson A Goodrich
Secretary	L H Hart

Board of Managers

Chairman, Wm M Mackay

Hugh J Barron — Stewart A Jellett
W S Hadaway, Jr — Wiltsie F Wolfe
R C Carpenter, *Pres* — L H Hart, *Secy*

Council

Chairman, A A Cary

Albert A Cryer — B. F. Stangland
Wm. McMannis — J J Blackmore, *Secy*

1897

President	Wm M Mackay
1st Vice-President	H D Crane
2nd Vice-President	Henry Adams
3rd Vice-President	A E Kenrick
Treasurer	Judson A Goodrich
Secretary	H M Swetland

Board of Managers

Chairman R C Carpenter

Edward P Bates — Stewart A Jellett
W S Hadaway, Jr — Wiltsie F Wolfe
Wm M Mackay *Pres* — H M Swetland, *Secy*

Council

Chairman Albert A Cryer

John A Fish — James Mackay
Wm McMannis — B F Stangland

1898

President	Wiltsie F Wolfe
1st Vice-President	J H Kinealy
2nd Vice-President	A E Kenrick
3rd Vice-President	John A Fish
Treasurer	Judson A Goodrich
Secretary	Stewart A Jellett

Board of Managers

Chairman, Wm M Mackay

Thomas Barwick — A C Mott
John A Connolly — Francis A Williams
Wiltsie F Wolfe, *Pres* — Stewart A Jellett *Secy*

Council

Chairman, R C Carpenter

Henry Adams — W S Hadaway, Jr
Albert A Cryer — Wm McMannis
Wiltsie F Wolfe, *Pres* — Stewart A Jellett *Secy*

1899

President	Henry Adams
1st Vice-President	D M Quay
2nd Vice-President	A E Kenrick
3rd Vice-President	Francis A Williams
Treasurer	Judson A Goodrich
Secretary	Wm M Mackay

Board of Managers

Chairman, Stewart A Jellett

B H Carpenter — Wm Kent
A A Cary — Wiltsie F Wolfe
Henry Adams, *Pres* — Wm M Mackay, *Secy*

Council

Chairman, R C Carpenter

John Gormly — Wm McMannis
W. S Hadaway, Jr — B F Stangland
Henry Adams, *Pres* — Wm M Mackay, *Secy*

1900

President	D M Quay
1st Vice-President	A E Kenrick
2nd Vice-President	Francis A Williams
Treasurer	Judson A Goodrich
Secretary	Wm M Mackay

Board of Governors

Chairman, D M Quay

Wm Kent, *Vice-Chm* — D M Nesbit
R C Carpenter — C B J Snyder
John Gormly — Wm M Mackay, *Secy*

1901

President	J H Kinealy
1st Vice-President	A E Kenrick
2nd Vice-President	Andrew Harvey
Treasurer	Judson A Goodrich
Secretary	Wm M Mackay

Board of Governors

Chairman, J H Kinealy

Wm Kent, *Vice-Chm* — John Gormly
R C Carpenter — C B J Snyder
R P Bolton — Wm M Mackay, *Secy*

1902

President	A E Kenrick
1st Vice-President	Andrew Harvey
2nd Vice-President	Robert C Clarkson
Treasurer	Judson A Goodrich
Secretary	Wm M Mackay

Board of Governors

Chairman, A E Kenrick

John Gormly, *Vice-Chm* — J H Kinealy
R C Carpenter — C B J Snyder
Wm Kent — Wm M Mackay *Secy*

1903

President	H D Crane
1st Vice-President	Wm Kent
2nd Vice-President	R P Bolton
Treasurer	Judson A Goodrich
Secretary	Wm M Mackay

Board of Governors

Chairman, H D Crane

C B J Snyder, *Vice-Chm* — A E. Kenrick
R C Carpenter — Geo Mehring
John Gormly — Wm M Mackay, *Secy*

1904

President	Andrew Harvey
1st Vice-President	John Gormly
2nd Vice-President	Robert C Clarkson
Treasurer	Ulysses G Scollay
Secretary	Wm M Mackay

Board of Governors

Chairman, Andrew Harvey

John Gormly — H D Crane
Robert C Clarkson — A E Kenrick
J J Blackmore — C B J Snyder
R C Carpenter — Wm M Mackay, *Secy*

1905

President	Wm Kent
1st Vice-President	R P. Bolton
2nd Vice-President	C. B. J. Snyder
Treasurer	Ulysses G Scollay
Secretary	Wm. M Mackay

Board of Governors

Chairman, Wm Kent

R P Bolton — James Mackay
C B J Snyder — B F Stangland
B H Carpenter — J C F Trachsel
A B Franklin — Wm M Mackay, *Secy*

1906

President	John Gormly
1st Vice-President	C B J Snyder
2nd Vice-President	T J Waters
Treasurer	Ulysses G Scollay
Secretary	Wm M Mackay

Board of Governors

Chairman, John Gormly

C B J Snyder, *Vice-Chm* — James Mackay
R C Carpenter — B F Stangland
Frank K Chew — T J Waters
A B Franklin — Wm M Mackay, *Secy*

1907

President	C B J Snyder
1st Vice-President	James Mackay
2nd Vice-President	Wm G Snow
Treasurer	Ulysses G Scollay
Secretary	Wm M Mackay

Board of Governors

Chairman, C B J Snyder

James Mackay, *Vice-Chm* — Frank K Chew
R E Atkinson — A B Franklin
R C Carpenter — Wm G Snow
Edmund F Capron — Wm M Mackay, *Secy*

1908

President	James Mackay
1st Vice-President	Jas D Hoffman
2nd Vice-President	B F Stangland
Treasurer	Ulysses G Scollay
Secretary	Wm M Mackay

Board of Governors

Chairman James Mackay

Jas D Hoffman, *Vice-Chm* — John F Hale
B F Stangland — August Kehm
R C Carpenter — C B J Snyder
Frank K Chew — Wm M Mackay, *Secy*

1909

President	Wm G Snow
1st Vice-President	August Kehm
2nd Vice-President	B S Harrison
Treasurer	Ulysses G Scollay
Secretary	Wm M Mackay

Board of Governors

Chairman, Wm G Snow

August Kehm, *Vice-Chm* — Samuel R Lewis
John R Allen — James Mackay
R C. Carpenter — B F. Stangland
B S Harrison — Wm M Mackay, *Secy*

1910

President ...... Jas D. Hoffman
1st Vice-President ...... R P Bolton
2nd Vice-President ...... Samuel R Lewis
Treasurer ...... Ulysses G Scollay
Secretary ...... Wm M. Mackay

Board of Governors

Chairman, Jas D. Hoffman

R. P. Bolton, *Vice-Chm.*
Geo W Barr
R C Carpenter
Judson A Goodrich
John F Hale
Samuel R Lewis
James Mackay
Wm. M. Mackay, *Secy.*

1911

President ...... R P Bolton
1st Vice-President ...... John R. Allen
2nd Vice-President ...... A B Franklin
Treasurer ...... Ulysses G Scollay
Secretary ...... Wm W. Macon

Board of Governors

Chairman, R. P. Bolton

John R Allen, *Vice-Chm.*
John T Bradley
R. C Carpenter
James H Davis
A B Franklin
Jas D. Hoffman
August Kehm
Wm W. Macon, *Secy.*

1912

President ...... John R Allen
1st Vice-President ...... John F Hale
2nd Vice-President ...... Edmund F Capron
Treasurer ...... James A Donnelly
Secretary ...... Wm W Macon

Board of Governors

Chairman, John R. Allen

John F. Hale, *Vice-Chm*
Edmund F Capron
R. P. Bolton
Jas D Hoffman
Dwight D Kimball
Samuel R Lewis
Wm M Mackay
Wm. W Macon, *Secy*

1913

President ...... John F. Hale
1st Vice-President ...... A B Franklin
2nd Vice-President ...... Edmund F Capron
Treasurer ...... James A Donnelly
Secretary ...... Edwin A Scott

Board of Governors

Chairman, John F. Hale

A B Franklin, *Vice-Chm.*
John R Allen
Edmund F Capron
R P Bolton
Frank T Chapman
Ralph Collamore
James A Donnelly
Dwight D Kimball
Wm W. Macon
James M. Stannard
Theodore Weinshank
Edwin A. Scott, *Secy.*

1914

President ...... Samuel R Lewis
1st Vice-President ...... Edmund F Capron
2nd Vice-President ...... Dwight D Kimball
Treasurer ...... James A Donnelly
Secretary ...... J. J Blackmore

Council

Chairman, Samuel R Lewis

E. F Capron, *Vice-Chm*
Dwight D Kimball
John R. Allen
Frank T. Chapman
Frank I. Cooper
James A Donnelly
John F Hale
Harry M Hart
Frank G. McCann
Wm. W Macon
James M Stannard
J J. Blackmore, *Secy.*

1915

President ...... Dwight D Kimball
1st Vice-President ...... Harry M Hart
2nd Vice-President ...... Frank T. Chapman
Treasurer ...... Homer Addams
Secretary ...... J J Blackmore

Council

Chairman, Dwight D Kimball

Harry M Hart, *Vice-Chm.*
Homer Addams
Frank T Chapman
Frank I Cooper
E Vernon Hill
Wm. M. Kingsbury
Samuel R Lewis
Frank G McCann
J T J Mellon
Henry C Meyer, Jr.
Arthur K Ohmes
J J. Blackmore, *Secy.*

1916

President ...... Harry M Hart
1st Vice-President ...... Frank T Chapman
2nd Vice-President ...... Arthur K Ohmes
Treasurer ...... Homer Addams
Secretary ...... Casin W. Obert

Council

Chairman, Harry M. Hart

F T. Chapman, *Vice-Chm.*
Homer Addams
Charles R Bishop
Frank I. Cooper
Milton W Franklin
E Vernon Hill
Dwight D Kimball
Henry C Meyer, Jr
Arthur K Ohmes
Fred R Still
Walter S Timmis
Casin W Obert, *Secy.*

1917

President ...... J Irvine Lyle
1st Vice-President ...... Arthur K. Ohmes
2nd Vice-President ...... Fred R Still
Treasurer ...... Homer Addams
Secretary ...... Casin W. Obert

Council

Chairman, J Irvine Lyle

A K Ohmes, *Vice-Chm.*
Homer Addams
Davis S Boyden
Bert C Davis
Milton W Franklin
Charles A Fuller
Harry M Hart
E Vernon Hill
James M Stannard
Fred R. Still
Walter S Timmis
Casin W Obert, *Secy*

1918

President ...... Fred R Still
1st Vice-President ...... Walter S Timmis
2nd Vice-President ...... E Vernon Hill
Treasurer ...... Homer Addams
Secretary ...... Casin W Obert

Council

Chairman, Fred R. Still

W S Timmis, *Vice-Chm.*
Homer Addams
William H Driscoll
Howard H Fielding
H. P Gant
C W Kimball
J Irvine Lyle
E Vernon Hill
Frank G Phegley
Fred W. Powers
Champlain L Riley
Casin W Obert, *Secy*

1919

President ...... Walter S Timmis
1st Vice-President ...... E Vernon Hill
2nd Vice-President ...... Milton W Franklin
Treasurer ...... Homer Addams
Secretary ...... Casin W. Obert

Council

Chairman, Walter S Timmis

E. Vernon Hill, *Vice-Chm*
Homer Addams
Howard H. Fielding
Milton W Franklin
Harry E Gerrish
George B. Nichols
Frank G Phegley
Fred. W. Powers
Robt W. Pryor, Jr
Champlain L. Riley
Fred R Still
Casin W Obert, *Secy.*

1920

President .... E Vernon Hill
1st Vice-President .... Champlain L Riley
2nd Vice-President .... Jay R McColl
Treasurer .... Homer Addams
Secretary .... Casin W Obert

Council

Chairman, E Vernon Hill

C L Riley, *Vice-Chm.*
Homer Addams
Jos A Cutler
Wm H Driscoll
A C Edgar
Alfred Kellogg
Jay R McColl
George B Nichols
Robt W Pryor, Jr.
W S Timmis
Perry West
Casin W Obert, *Secy*

1921

President .... Champlain L Riley
1st Vice-President .... Jay R McColl
2nd Vice-President .... H P Gant
Treasurer .... Homer Addams
Secretary .... Casin W Obert

Council

Chairman, Champlain L Riley

Jay R McColl, *Vice-Chm*
Homer Addams
Jos A Cutler
Samuel E Dibble
Wm H Driscoll
H P Gant
E S Hallett
E Vernon Hill
Alfred Kellogg
E E McNair
Perry West
Casin W Obert, *Secy.*

1922

President .... Jay R McColl
1st Vice-President .... H P Gant
2nd Vice-President .... Samuel E Dibble
Treasurer .... Homer Addams
Secretary .... Casin W Obert

Council

Chairman, Jay R McColl

H P Gant, *Vice-Chm*
Homer Addams
Jos A Cutler
Samuel E Dibble
Wm H Driscoll
E S Hallett
L. A Harding
E E McNair
H J Meyer
C L Riley
Perry West
Casin W Obert, *Secy.*

1923

President .... H P Gant
1st Vice-President .... Homer Addams
2nd Vice-President .... E E McNair
Treasurer .... Wm H Driscoll
Secretary .... C W Obert

Council

Chairman, H P Gant

Homer Addams, *Vice-Chm.*
W. H Carrier
J A Cutler
S E Dibble
Wm H Driscoll
E S Hallett
Alfred Kellogg
Thornton Lewis
E E McNair
Perry West
Casin W Obert, *Secy.*

1924

President .... Homer Addams
1st Vice-President .... S E Dibble
2nd Vice-President .... William H Driscoll
Treasurer .... Perry West
Secretary .... F C Houghten

Council

Chairman, Homer Addams

S. E. Dibble, *Vice-Chm.*
F. Paul Anderson
W. H Carrier
J. A Cutler
William H Driscoll
H. P Gant
W. E Gillham
L. A Harding
Alfred Kellogg
Thornton Lewis
Perry West
F. C. Houghten, *Secy.*

1925

President .... S E. Dibble
1st Vice-President .... Wm H. Driscoll
2nd Vice-President .... F. Paul Anderson
Treasurer .... Perry West
Secretary .... F C Houghten

Council

Chairman, S E. Dibble

Wm H Driscoll, *Vice-Chm.*
Homer Addams
F Paul Anderson
W H Carrier
J A Cutler
W E Gillham
W. T Jones
Thornton Lewis
J. H Walker
Perry West
A. C. Willard
F C. Houghten, *Secy*

1926

President .... W. H Driscoll
1st Vice-President .... F Paul Anderson
2nd Vice-President .... A C Willard
Treasurer .... W E Gillham
Secretary .... A V Hutchinson

Council

Chairman, W H Driscoll

F Paul Anderson, *Vice-Chm*
W H Carrier
J. A Cutler
S E Dibble
W E Gillham
C V. Haynes
W T Jones
E B Langenberg
Thornton Lewis
J F McIntire
A C Willard

1927

President .... F. Paul Anderson
1st Vice-President .... A C Willard
2nd Vice-President .... Thornton Lewis
Treasurer .... W E Gillham
Secretary .... A. V Hutchinson

Council

Chairman, F Paul Anderson

A C Willard, *Vice-Chm*
H H. Angus
W H Carrier
W H Driscoll
Roswell Farnham
H H Fielding
W. E Gillham
C V Haynes
John Howatt
W T Jones
J J Kissick
E B Langenberg
Thornton Lewis
J F McIntire
H Lee Moore
F B Rowley

1928

President .... A. C Willard
1st Vice-President .... Thornton Lewis
2nd Vice-President .... L A Harding
Treasurer .... W. E Gillham
Secretary .... A V Hutchinson

Council

Chairman, A C Willard

Thornton Lewis, *Vice-Chm*
F Paul Anderson
H H Angus
W H Carrier
N W Downes
Roswell Farnham
W E Gillham
C V Haynes
John Howatt
W T Jones
J J Kissick
E B Langenberg
J F McIntire
H Lee Moore
F B Rowley

1929

President .... Thornton Lewis
1st Vice-President .... L A Harding
2nd Vice-President .... W. H Carrier
Treasurer .... W E Gillham
Secretary .... A V. Hutchinson
Technical Secretary .... P. D. Close

Council

Chairman, Thornton Lewis

L. A Harding, *Vice-Chm*
H H Angus
W. H. Carrier
N. W Downes
Roswell Farnham
W E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
E. B. Langenberg
G L. Larson
F. C. McIntosh
W. A. Rowe
F B. Rowley
A. C. Willard

1930

President ... L A Harding
1st Vice-President ... W H Carrier
2nd Vice-President ... F B Rowley
Treasurer ... C W Farrar
Secretary ... A V. Hutchinson
Technical Secretary ... P D Close

Council

Chairman, L A Harding

W. H Carrier, *Vice-Chm*
H H Angus
D S Boyden
R H Carpenter
J D Cassell
N. W Downes
Roswell Farnham
C W Farrar
John Howatt
W T Jones
E. B Langenberg
G L Larson
Thornton Lewis
F C McIntosh
W A. Rowe
F B Rowley

1931

President ... W H Carrier
1st Vice-President ... F B Rowley
2nd Vice-President ... W. T Jones
Treasurer ... F D Mensing
Secretary ... A V Hutchinson
Technical Secretary ... P D Close

Council

Chairman, W H Carrier

F B. Rowley, *Vice-Chm.*
D S. Boyden
E K. Campbell
R. H Carpenter
J D Cassell
E. O Eastwood
Roswell Farnham
E H Gurney
L A. Harding
John Howatt
W. T. Jones
E B Langenberg
G L. Larson
F C. McIntosh
F. D Mensing
W A Rowe

1932

President ... F. B. Rowley
1st Vice-President ... W. T. Jones
2nd Vice-President ... C V. Haynes
Treasurer ... F D Mensing
Secretary ... A. V. Hutchinson
Technical Secretary ... P. D. Close

Council

Chairman, F. B. Rowley

W T Jones, *Vice-Chm*
D S Boyden
E. K. Campbell
R H Carpenter
W. H Carrier
John D. Cassell
E. O Eastwood
Roswell Farnham
F. E. Giesecke
E. H Gurney
C. V. Haynes
John Howatt
G. L. Larson
J. F. McIntire
F. D Mensing
W. E. Stark

1933

President ... W. T Jones
1st Vice-President ... C V. Haynes
2nd Vice-President ... John Howatt
Treasurer ... D S. Boyden
Secretary ... A. V Hutchinson

Council

Chairman, W. T. Jones

C. V. Haynes, *Vice-Chm.*
D. S Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O Eastwood
R. Farnham
F E. Giesecke
E. H Gurney
John Howatt
G. L. Larson
J F. McIntire
F. C. McIntosh
L. W. Moon
F. B. Rowley
W. E. Stark

1934

President ... C V Haynes
1st Vice-President ... John Howatt
2nd Vice-President ... G L Larson
Treasurer ... D S Boyden
Secretary ... A V Hutchinson

Council

Chairman, C V Haynes

John Howatt, *Vice-Chm*
M C Beman
D S Boyden
Albert Buenger
R H Carpenter
J D Cassell
F E Giesecke
E H Gurney
W T Jones
G L Larson
J F McIntire
F. C McIntosh
L Walter Moon
O. W Ott
W A Russell
W E Stark

1935

President ... John Howatt
1st Vice-President ... G L Larson
2nd Vice-President ... D S Boyden
Treasurer ... A J Offner
Secretary ... A V Hutchinson

Council

Chairman, John Howatt

G L Larson, *Vice-Chm*
M C Beman
D S Boyden
Albert Buenger
R H Carpenter
J. D Cassell
F E Giesecke
E H Gurney
C V Haynes
J F McIntire
F C McIntosh
L Walter Moon
A J Offner
O W Ott
W. A. Russell
W E Stark

1936

President ... G L Larson
1st Vice-President ... D S. Boyden
2nd Vice-President ... E. H Gurney
Treasurer ... A J Offner
Secretary ... A V Hutchinson

Council

Chairman, G L Larson

D S Boyden, *Vice-Chm.*
M. C Beman
R C. Bolsinger
Albert Buenger
S H Downs
W L Fleisher
F. E Giesecke
E H Gurney
John Howatt
C M Humphreys
L Walter Moon
J F. McIntire
A. J Offner
O W Ott
W A Russell
W E Stark

1937

President ... D S Boyden
1st Vice-President ... E Holt Gurney
2nd Vice-President ... J. F. McIntire
Treasurer ... A J Offner
Secretary ... A V Hutchinson

Council

Chairman, D S Boyden

E. H Gurney, *Vice-Chm*
J J Aeberly
M C Beman
R C Bolsinger
Albert Buenger
S H. Downs
E O. Eastwood
W L Fleisher
F E. Giesecke
C. M. Humphreys
G L Larson
W A Russell
W E Stark

1938

President ... E Holt Gurney
1st Vice-President ... J F McIntire
2nd Vice-President ... F E. Giesecke
Treasurer ... A J Offner
Secretary ... A V Hutchinson
Technical Secretary ... John James

Council

Chairman, E Holt Gurney

J F McIntire, *Vice-Chm*
N. D. Adams
J J Aeberly
M C Beman
R. C Bolsinger
D. S Boyden
S H Downs
E O Eastwood
W L Fleisher
F. E Giesecke
C M Humphreys
A P Kratz
A J Offner
W A Russell
J H. Walker
G L Wiggs